# NBS SPECIAL PUBLICATION 622

U.S. DEPARTMENT OF COMMERCE / National Bureau of Standards

# Detection, Diagnosis and Prognosis: Contribution to the Energy Challenge

MFPG
32nd Meeting

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# MFPG

# Detection, Diagnosis and Prognosis: Contribution to the Energy Challenge

Proceedings of the 32nd Meeting of the
Mechanical Failures Prevention Group,
held at the Inn at Santa Monica,
Santa Monica, California, October 7-9, 1980

Edited by

T. Robert Shives and William A. Willard

Center for Materials Science
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Washington, DC 20234

The 32nd meeting of the MFPG and these proceedings were sponsored by the Center for Materials Science of the National Bureau of Standards, Washington, DC 20234; the Office of Naval Research, Department of the Navy, Arlington, VA 22217; the Naval Air Systems Command, A. J. Koury, Washington, DC 20360; and the National Aeronautics and Space Administration, Goddard Space Flight Center, Greenbelt, MD 20771.

U.S. DEPARTMENT OF COMMERCE, Malcolm Baldrige, Secretary
NATIONAL BUREAU OF STANDARDS, Ernest Ambler, Director

Issued October 1981

Library of Congress Catalog Card Number: 81-600125

National Bureau of Standards Special Publication 622
Nat. Bur. Stand. (U.S.), Spec. Publ. 622, 329 pages (Oct. 1981)
CODEN: XNBSAV

U.S. GOVERNMENT PRINTING OFFICE
WASHINGTON: 1981

For sale by the Superintendent of Documents, U.S. Government Printing Office, Washington, D.C. 20402
Price $8.00
(Add 25 percent for other than U.S. mailing)

# FOREWORD

The 32nd meeting of the Mechanical Failures Prevention Group was held October 7-9, 1980, at the Inn at Santa Monica in Santa Monica, California. The program, which focused on failure prevention in energy related systems, was organized by the MFPG Detection, Diagnosis and Prognosis Committee under the chairmanship of Robert R. Holden of the Hughes Aircraft Company in Los Angeles. Appreciation is extended to the committee, the session chairmen, and especially to the speakers for an exceptional program. Gratitude is expressed to the Hughes Aircraft Company for hosting the meeting.

Appreciation is extended to T. Robert Shives and William A. Willard of the National Bureau of Standards Fracture and Deformation Division for their editing, organization and preparation of these proceedings, and to Leonard C. Smith, also of the NBS Fracture and Deformation Division, for photographic work. Most of the papers in the proceedings are presented as submitted by the authors on camera ready copy. Some moderate editorial changes were required.

Gratitude is expressed to Marian L. Slusser of the NBS Center for Materials Science for handling financial matters.


HARRY C. BURNETT
Executive Secretary, MFPG

Center for Materials Science
National Bureau of Standards

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APPENDIX



*Indicates speaker when a paper had more than one author.

ABSTRACT

These proceedings consist of a group of 21 submitted entries (19 papers and 2 abstracts) from the 32nd meeting of the Mechanical Failures Prevention Group which was held at the Inn at Santa Monica, Santa Monica, California, October 7-9, 1980. The subject of the symposium was the contribution of failure detection, diagnosis and prognosis to the energy challenge. Areas of special emphasis included energy management, techniques for failure detection in energy related systems, improved detection and diagnostic availability for energy related systems, diagnostic and prognostic techniques for energy related systems, and opportunities for detection, diagnosis and prognosis in the energy field.

Key Words: Energy; energy conservation; failure detection; failure diagnosis; failure prevention; monitoring techniques; prognosis; sensors; wear; wear analysis.

UNITS AND SYMBOLS

Customary U.S. units and symbols appear in some of the papers in these proceedings. The participants in the 32nd meeting of the Mechanical Failures Prevention Group have used the established units and symbols commonly employed in their professional fields. However, as an aid to the reader in increasing familiarity with the usage of the metric system of units (SI), the following references are given:

NBS Special Publication, SP330, 1977 Edition, "The International System of Units."

ISO International Standard 1000 (1973 Edition), "SI Units and Recommendations for Use of Their Multiples."

IEEE Standard Metric Practice (Institute of Electrical and Electronics Engineers, Inc., Standard 268-1979).

# SESSION I

## ENERGY MANAGEMENT

Chairmen: A. Whittaker, Honeywell, Inc.

J. Philips, David Taylor Naval

Ship R & D Center

# STATISTICAL ANALYSIS OF CORE BARREL MOTION ORBITS

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Abstract: Excore neutron detectors have been widely used to provide in-service monitoring of the mechanical motion of the reactor core support barrel in pressurized water reactors. The accepted procedure for determining core barrel motion (CBM) involves analyzing small fluctuations in the output signals of the excore neutron detectors and relating part of these fluctuations to the motion of the reactor core support barrel. This relationship between detector signal fluctuation and core barrel movement is non-trivial and statistical descriptors have been previously developed to make determination of CBM root-mean-squared magnitudes possible. This paper presents a new statistical descriptor, herein called Core Barrel Orbits, which provides information about both the magnitude and direction of core barrel motion. This analysis can be used to determine the most probable locations for CBM and to develop confidence limits on the maximum displacements which occur. Use of this detailed analysis can lead to detection of potential problems in the core barrel support or restraint mechanisms which would occur only at certain directions.

Key Words: Core Barrel Motion; Neutron Detectors; Reactor Internals; Surveillance; Vibration.

Introduction: Excore neutron detectors have been widely used to provide in-service monitoring of the mechanical motion of the reactor core support barrel in pressurized water reactors (Ref. 1-4). The accepted procedure for determining core barrel motion (CBM) involves analyzing small fluctuations in the output signals of the excore neutron detectors and relating part of these fluctuations to the motion of the reactor core support barrel. This relationship between detector signal fluctuation and core barrel movement is non-trivial and statistical descriptors have been developed (Ref. 5) to make determination of CBM possible. These statistical descriptors are the cross- and auto-power spectral densities and the amplitude probability distribution functions of these fluctuations. These descriptors, although adequate for determination of CBM magnitude, give little information about the direction in which the core barrel moves. This paper presents a new statistical descriptor, herein called Core Barrel Orbits, which provides this information.

## CBM Determination

The core of a pressurized water reactor is contained within a cylindrical core barrel which is surrounded by a thermal shield, as illustrated in Figures 2.1 and 2.2. The core barrel is supported at its top with the lower portion being free to move in a beam mode, or cantilever fashion. In an operating reactor, movements of the core barrel are induced primarily by random pressure fluctuations due to the turbulent flow of water in the downcomer and up through the core. The frequency of this mechanical vibration is governed primarily by the clamping force of the core barrel support mechanism.

When a reactor is operating at power, the displacements of the core barrel produce small variations in the number of neutrons escaping from the core due to variations in the thickness of the water gap between the core barrel and thermal shield, as illustrated in Figure 2.3. These CBM induced variations are superimposed with signals produced by effects not associated with CBM, such as reactivity fluctuations and statistical variations in the reaction rate within the detector.

The problem of measuring CBM-induced responses reduces to separating signal fluctuations due to CBM from the other effects which are seen by these detectors. This may be accomplished by realizing that diametrically opposite power range detectors will see the same signal from beam mode CBM-induced phenomena, but shifted in phase by 180 degrees. Making use of this realization, a cross power spectral density (CPSD) function may be constructed for a cross-core detector pair with the corresponding phase and coherence information used to establish a frequency band over which CBM is occurring. A band-pass filter may then be employed to reject signal components outside this frequency range. If CBM is the dominant driving function over this frequency band (as determined from the coherence and phase of cross-core detector pairs), the band-limited signal is primarily the result of only core barrel movement (with some statistical noise). This signal can be assumed to be directly proportional to CBM provided the system behaves in a linear fashion or if the signal fluctuations are sufficiently small so that non-linear effects are negligible.

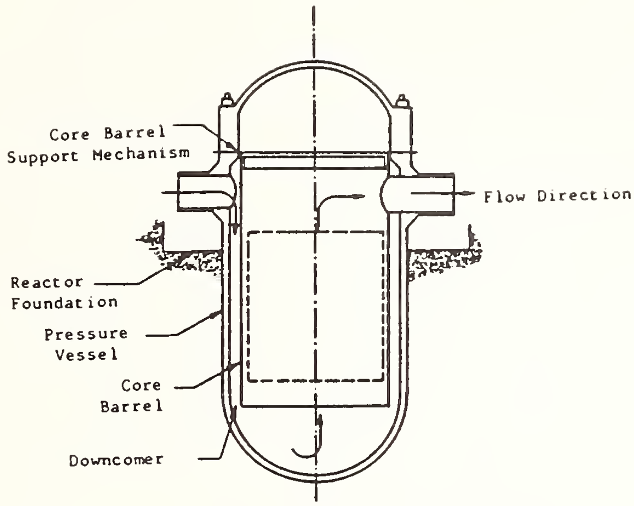


Figure 2.1 A Typical PWR Reactor Core Internals Structure

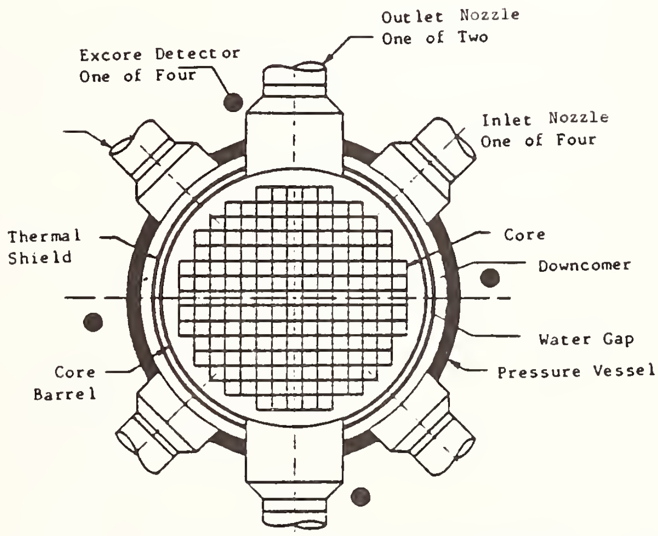


Figure 2.2 General Arrangement of Reactor Vessel Internals, Top View.

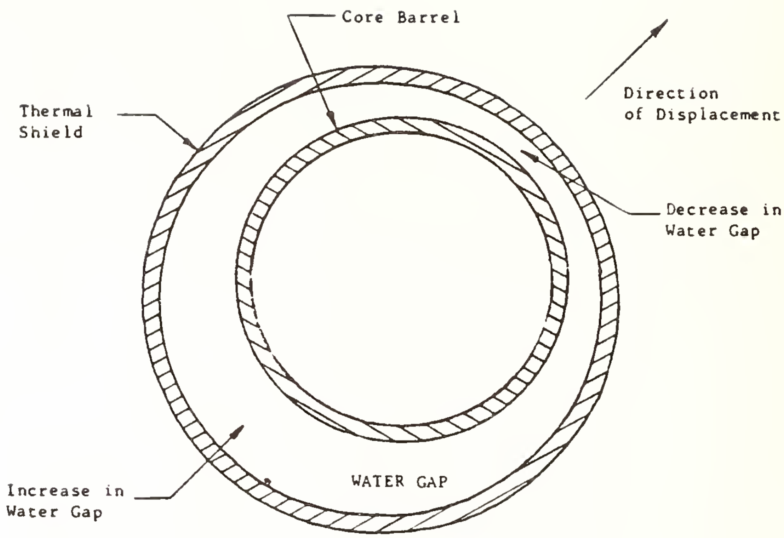


Figure 2.3 Representation of the Change in Water Gap Due to Core Barrel Motion

Experimental evidence has defended this linear assumption and a scale factor converting fractional detector response to mils motion of the core support barrel has been calculated (Ref. 6) as:

$$\eta = 0.02\ \%\ \text{detector change per mil} \qquad (2.1)$$

Theoretical Considerations

The motion of the core barrel can be represented by a displacement vector, D(t), between the center of motion and the time varying position of the core barrel center, as illustrated by Figure 3.1. Given a properly conditioned neutron detector signal, the fluctuation (Xj(t)) caused by this displacement is given by:

$$X_j(t) = \frac{D(t) \cdot Rj}{\eta} \qquad (3.1)$$

where $\eta$ represents a conversion factor converting detector response to mils motion (given by Equation 2.1) and Rj represents a unit vector in the direction of the detector. Given N detectors, Equation 3.1 leads to N equations (one equation for each detector)

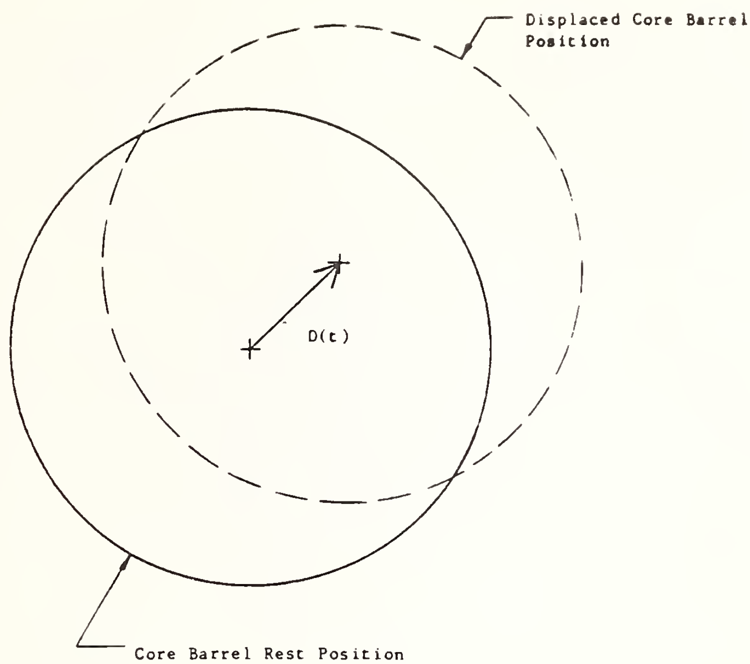


Figure 3.1 Vector Representation of Core Barrel Motion

with only 2 unknown variables - the magnitude and the direction of the displacement vector, D(t). This set of equations is solvable for two or more detectors (provided at least two are linearly independent). Therefore, given at least two neutron detectors which are separated by an angle different than 180 degrees (to insure independence), we may transform their time variant signals into a time variant core barrel displacement (both magnitude and direction). This provides the basis of the core barrel orbit statistical descriptor.

Core Barrel Orbit Implementation

Most PWRs have four or six power range neutron detectors which are located as pairs of diametric (cross core) detectors (See Figure 2.1). To construct a core barrel orbit, all available detector signals are band-limited so that signal components outside the beam mode core barrel motion range are rejected. A time variant core barrel displacement is then computed from the simultaneous output of all detectors by employing a least-squares fit to Equation 3.1.

If a large number of the resulting core barrel displacement vectors are plotted together, a scatter plot similar to Figure 4.1 is

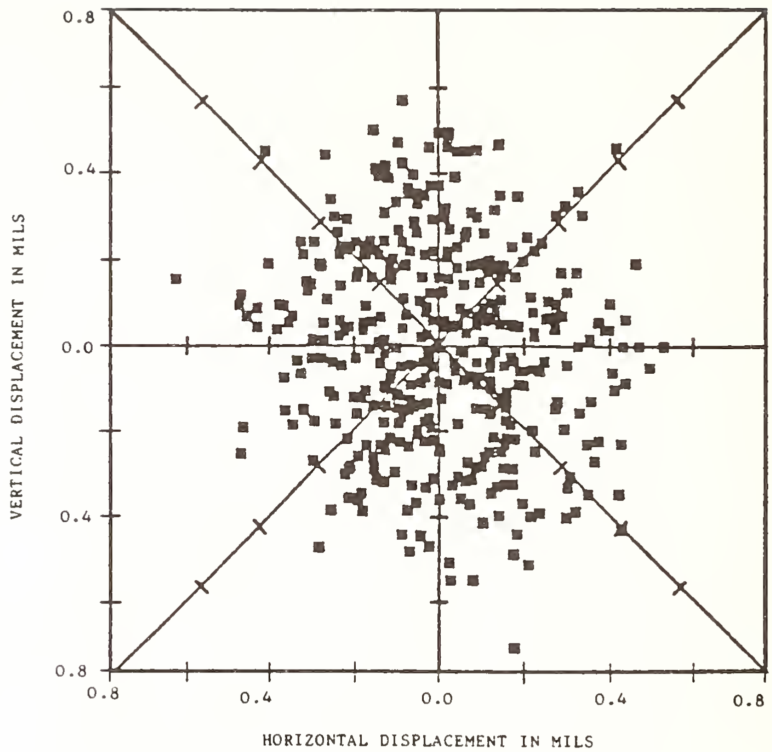


Figure 4.1 Variation in Core Barrel Center Position Over a 5.0 Second Time Interval

produced. This figure clearly indicates the random nature of the core barrel motion since there is not an observable path the core barrel consistently follows. As a result of this random behavior, describing the spatial characteristics of the CBM is a statistical problem. One approach for quantifying CBM that has been used is the construction of a joint amplitude probability distribution function, which describes the two dimensional probability of the core barrel being in any given position. The result of this approach, however, does not lend itself to easy interpretation.

A more direct approach is to average all displacements which occur along any given direction and to compute the variance of these displacements. When this average is calculated for all directions and plotted, a plot similar to Figure 4.2 is obtained. The scattered data points in the figure represent the angular displacement average while the smooth curve is a best fit to an elliptical path.

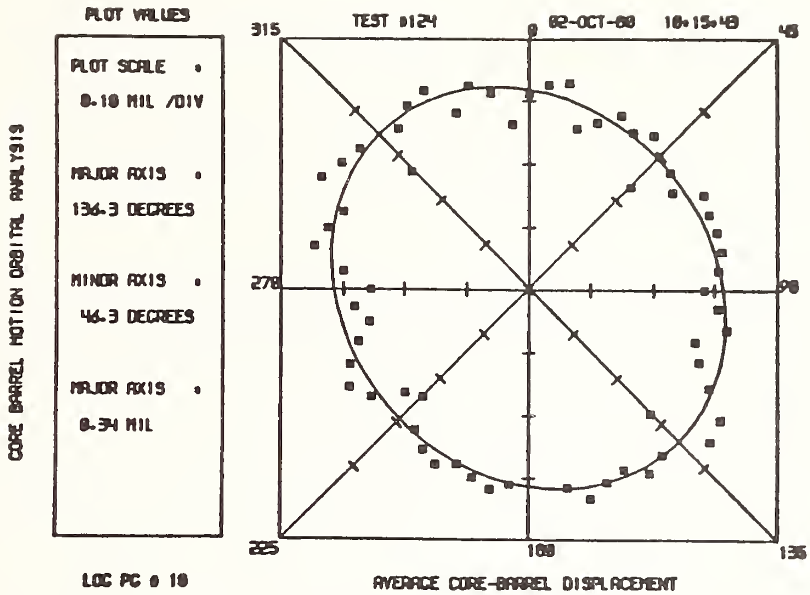


Figure 4.2. Average Core Barrel Position and Best Fit Elliptical Path

If the standard deviation limits are also plotted for each direction, the resulting curves will represent the 63 percent confidence path. This is defined as the path with which there is a 63 percent confidence level that any given core barrel displacement will be contained within. By a similar manner, one may obtain the path at any desired confidence level.

## Summary

Statistical descriptors are used to determine core barrel motion from the signals of the ex-core neutron detectors. The descriptors currently in use give little information about the directional manner in which the core barrel moves. A new statistical descriptor (core barrel motion orbits) is presented which allows this information to be determined. The theoretical considerations of this methodology are discussed as well as its implementation and results. We feel that this new descriptor provides a significant new tool for the analysis and understanding of core barrel motion.

## References


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4. C. W. Mayo and R. L. Currie, "Neutron Noise Monitoring of Pressurized Water Reactor Internal Vibrations" Proc. of the Second Specialists Meeting on Reactor Noise. (Sept. 1977).

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# ACOUSTIC MONITORING OF POWER PLANT VALVES

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Abstract: Hydrodynamic related signals from pressurized water reactor (PWR) feedwater control valves (FCVs) measured by a mini-computer based monitoring system were able to predict and identify vibration levels and cavitation conditions during plant transients. These results came from a program sponsored by the Electric Power Research Institute (EPRI), the overall objective being to increase nuclear power plant availability by monitoring FCVs for failure mechanisms leading to plant shutdowns. Acoustic emission and accelerometer signals, inlet water pressure, feedwater temperature and signals necessary to calculate water velocity were analyzed. The analyses determined that cavitation and high vibration conditions occurred during certain operating conditions. These results can be used by power plant operators to avoid operating conditions where cavitation will occur.

Key words: Acoustic emissions; cavitation; feedwater control valve; pattern recognition; pressurized water reactor; remote surveillance.

## INTRODUCTION

The Electric Power Research Institute is sponsoring research projects to improve monitoring and diagnostic systems for power plant machinery. The primary objective of these projects is to increase plant availability by developing systems that can detect failures early enough to warn operators of impending failures and inform maintenance personnel in advance about the nature of the failure. Warnings will allow compensation in advance for the eventual equipment shutdown, and information on the nature of the failure will allow more effective planning and scheduling of maintenance.

One of the projects in this program is RP1246-1, "Acoustic Monitoring of Power Plant Valves." The overall objective of the project was to improve nuclear power plant availability, productivity and safety through the application of advanced surveillance techniques to power plant valves. The scope of work reported here concerns PWR Feedwater Control Valve Monitoring and includes:

- A survey and classification of PWR feedwater control valve (FCV) induced outages to determine likely failure modes which could be monitored

- The development of analytic models for FCVs

- The design and construction of a remotely located, minicomputer based system to monitor FCVs

- Installation of the monitoring system and data acquisition at three PWR plants

- Evaluation of system performance and a conceptual design for a reduced system

## SURVEY AND CLASSIFICATION OF FCV FAILURES

A survey was conducted with the intent of identifying the prominent failure modes in FCVs. This survey utilized EPRI report NP-241, "Assessment of Industry Valve Problems," in conjunction with License Event Reports (LER) and communications with power plant personnel and FCV vendors. As a result, six classifications of FCV problems and failures were identified. These are:

- Controller Malfunction

  -- Problems in this category are characterized by mechanical failure of components or poor response of the FCV to normal demands such as passivity when movement is expected and motion which does not correspond to the demand given.

- Mechanical Breakdown

  -- This category is characterized by structural degradation of FCV components. This includes such items as stem failure, trim cracking and air line breaking.

- Hydrodynamic Instability

  -- This category is characterized by structural degradation directly attributable to flow and FCV misbehavior due to particular flow instabilities. This includes cavitation phenomena, sudden valve motion not related to controller demand signals and water leakage.

- Electronic/Pneumatic Malfunction

  -- This category is characterized by electrical and pneumatic component problems and includes spurious signals to controller components and breakdowns of components of an electrical and/or pneumatic nature.

- Generic Design

-- This category is characterized by inadequate design of FCVs for the operating conditions encountered. This includes items such as under/oversized actuators or underdamped movement characteristics.

- Human Error

-- This cateogory is characterized by misjudgments of the operator and improper repair.

The problem areas investigated in this report include controller malfunction, mechanical breakdown and hydrodynamic instability.

RESULTS AND CONCLUSIONS

The surveyed problems and failures of FCVs were listed and placed in one of the six classifications discussed above. The plant downtime associated with each failure was also included. The primary source of this data is EPRI Report NP-241. The results are summarized in Table 1.

Table 1

SUMMARY OF SURVEY OF FCV FAILURES

| Failure Category | Incidents | Forced Outages | Downtime (days) |
|---|---|---|---|
| Controller | | | |
| - Mechanical | 7 | 7 | 4.4 |
| - Unknown | 13 | 11 | 5.8 |
| Mechanical | 10 | 7 | 9.5 |
| Hydrodynamic | 5 | 4 | 3.6 |
| Electronic/Pneumatic | 14 | 5 | 4.7 |
| Generic | 2 | 2 | 70.0 |
| Human | 3 | 3 | 1.1 |
| TOTALS | 54 | 39 | 99.1 |

The following conclusions were made based on the survey results shown above and the communications with plant personnel and vendors:

* Most of the mechanical problems in FCVs appear to be a result of excessive vibrations and/or associated physical fatigue. This is reflected by vendors who are now redesigning the FCV internal structure and actuator damping features in order to reduce excessive vibrations.

* Many problems are apparently due to stem packing tightness; i.e., if it is too loose, the FCV leaks but if it is too tight, the stem is unable to move. Determining the proper tightness is still more of an art than a science.

* The "unknown" controller problems could not be attributed directly to water flow phenomena or spurious behavior of the controller components.

* Many of the electronic malfunctions were on equipment mounted directly onto the FCV topworks which are related to the control signals (e.g., valve positioners, actuators, I/P transducers), and were induced by excessive vibrations.

## FCV ANALYTICAL MODELS

In order to assist in the design and development of the surveillance system and aid in the interpretation of data, analytic models of the FCV internal structures were developed and the FCV controller and internal structures interaction with hydrodynamic phenomena were analyzed.

### Controller

The controllers were typical 4-20 mA, three-element controllers using feedwater flow/steam flow mismatch and steam generator level as the controlling parameters. Figure 1 shows the flow of the control signals and the normal components used. These components are usually a combination of (1) a low pass filter to eliminate spurious steam generator level signals, (2) two integral-proportional filters, (3) a current to pressure (I/P) transducer, (4) a stem positioner, and (5) a diaphragm or piston operated actuator.

### Valve Internal Structure

Mechanical models of the FCV plug and trim were derived to determine resonant frequencies that could be excited by hydrodynamic forces. Finite element models were developed for a double globe stem-plug assembly, a cylindrical stem-plug assembly, and an upper and lower trim.

These models determined the resonant frequencies of the dominant mode shapes for the plug-stem assemblies clamped at the bonnet and for the upper and lower trim. These are shown in Table 2.

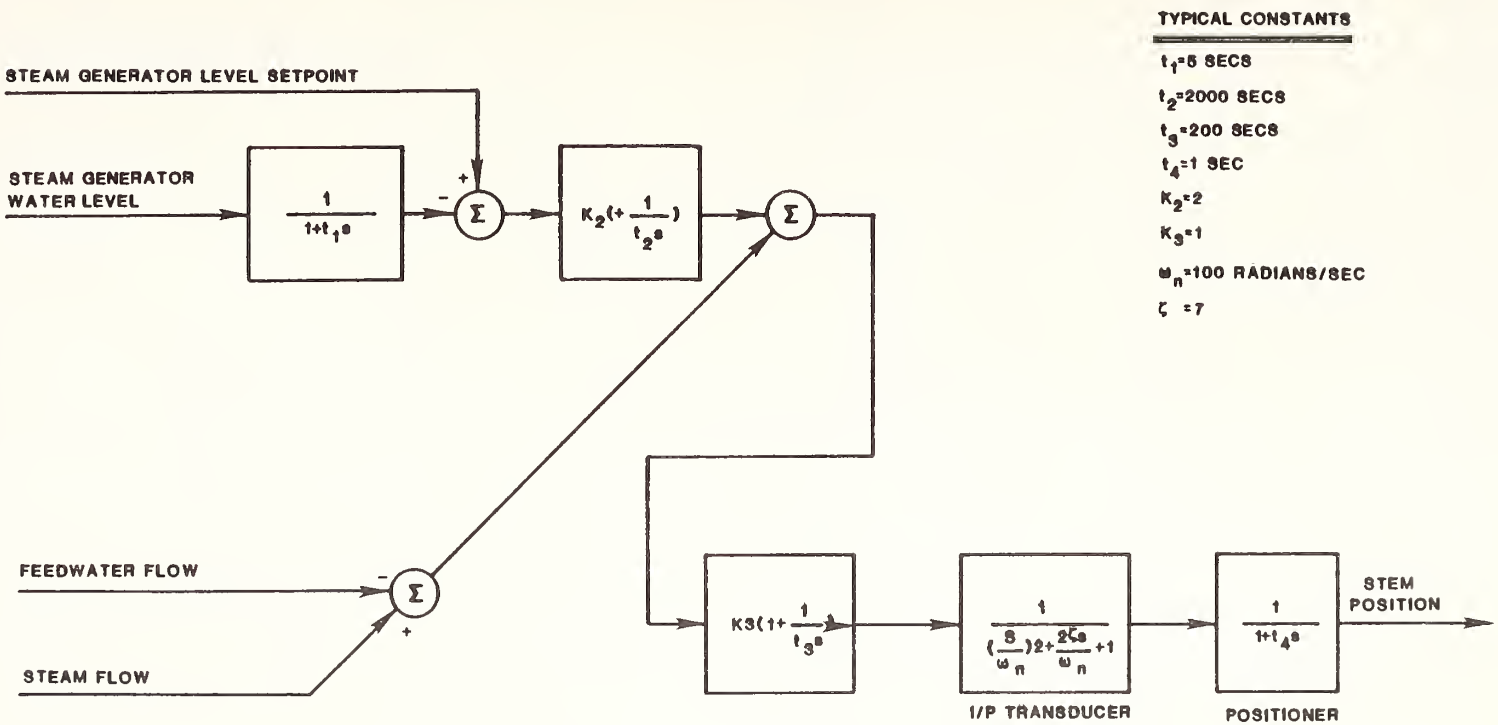


Figure 1. Three Element Feedwater Control Valve Controller

TABLE 2

Natural Frequencies and Mode Shapes for FCV Internal Structures

| Double Globe Plug-Stem Frequency (Hz) | Double Globe Plug-Stem Mode | Cylindrical Stem-Plug Frequency (Hz) | Cylindrical Stem-Plug Mode | Upper Trim[a] Frequency (Hz) | Lower Trim[a] Frequency (Hz) |
|---|---|---|---|---|---|
| 7 | Fundamental bending | 17 | Fundamental bending | 4,046 | 7,977 |
| 99 | Fundamental bending | 20 | Fundamental bending | 4,058 | 9,265 |
| 428 | Longitudinal | 43 | Fundamental torsion | 4,796 | 12,520 |
| 1,233 | Higher order bending | 100 | Higher order bending | 4,831 | 13,650 |
| 2,080 | Higher order bending | 217 | Higher order bending | 5,415 | 14,380 |
| | | 454 | Longitudinal | | |
| | | 1,015 | Higher order bending | | |
| | | 1,129[b] | Ovalizing shell | | |

[a] Mode shapes were bending or ovalizing shell.

[b] Seven ovalizing shell modes were found between, 1,129 and 3,200 Hz.

The notable features of these models are:

- The longitudinal resonant frequencies for the plug-stem assemblies were both around 450 Hz.
- The cylindrical plug-stem assembly has a torsional resonance around 43 Hz that was not seen in the double globe assembly.
- The valve trim structures had resonant frequencies greater than 4 kHz.

A further analysis of the torsional resonance in the cylindrical plug-stem assembly for stem lengths not clamped at the bonnet determined that resonant frequencies could vary from 11 to 19 Hz depending on length and clearance assumptions.

## Hydrodynamic Phenomena

The internal structure of the cylindrical plug modelled above was conducive to vortex shedding phenomena. Vortex shedding frequencies were determined which could create both torsional or longitudinal driving forces for various flow rates through the valve. These are summarized in Table 3.

TABLE 3

Vortex Shedding Frequencies for the Cylindrical Plug-Stem Assembly

| | Shedding Frequency (Hz) | |
|---|---|---|
| Flow Rate (lbs/hr)[a] | Longitudinal | Torsional |
| $3.5 \times 10^6$ | 30.5 | 15.3 |
| $3.2 \times 10^6$ | 28.0 | 14.0 |
| $2.7 \times 10^6$ | 27.5 | 13.8 |
| $2.3 \times 10^6$ | 19.8 | 9.9 |
| $1.9 \times 10^6$ | 16.4 | 8.4 |
| $1.5 \times 10^6$ | 13.4 | 6.7 |

[a] Rated 100% flow rate is $3.5 \times 10^6$ lbs/hr.

Cavitation in valves can occur on the exit side of the ports if the water velocity becomes high enough to cause the vena-contracta pressure to drop below the fluid vapor pressure. The vena-contracta pressure was calculated by assuming the ports act as a sudden contraction in a free steam pipe, i.e.,

$$Pv = Po - \frac{\dot{m}^2}{2\rho Cd^2 A^2}$$

where $\dot{m}$ = Fluid flow rate

$\rho$ = Fluid density

Po = Free stream pressure

A = Area of the port

Pv = Vena contracta pressure

and Cd = Discharge coefficient.

## DESIGN AND CONSTRUCTION OF THE FCV SURVEILLANCE SYSTEM

A mini-computer based pattern recognition system was chosen for the FCV monitoring. The system was designed to be moved to a plant site and set up in a remote area where it would continuously monitor the FCV.

The design and construction of this FCV surveillance system was done in three overlapping stages:

- Determination of key parameters, the signals required to monitor them and the sensors required.
- Procurement and construction of a mini-computer based hardware system.
- Development of data handling and pattern recognition algorithms.

A simplified block diagram of the finished system is shown in Figure 2.

### Determination of Key Parameters, Signals and Sensors

The parameters important for monitoring the failure modes presented in the survey were chosen using input from plant and vendor personnel and engineering judgement. Following the selection of these parameters, the signals or variables best describing them were chosen on the basis of availability and simplicity. Special effort was made to include signals already available in the plants; otherwise sensors were chosen to make the measurement. A total of 19 signals were chosen of which six were already available. Table 4 summarizes the key parameters, sensors and variables measured.

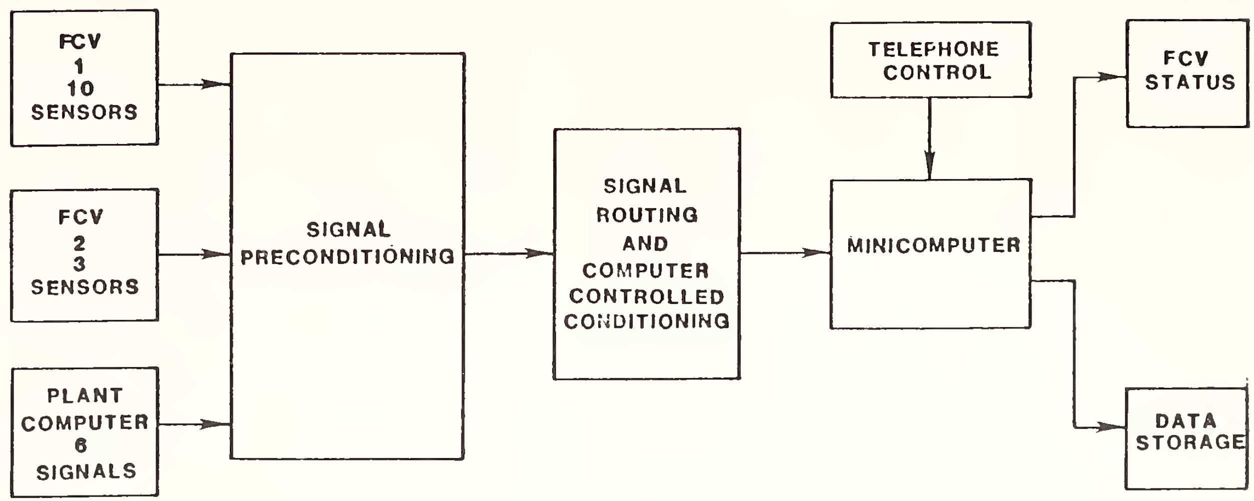


Figure 2. Block Diagram of Feedwater Control Valve Surveillance System

Table 4

KEY PARAMETERS, SENSORS AND MEASURED VARIABLE
USED IN THE FCV SURVEILLANCE SYSTEM

| Key Parameter | Sensor(s) | Measured Variable |
|---|---|---|
| Valve Demand and Response (Controller) | Pressure transducer[a] | Control pressure to positioner |
| | Pressure transducer | Actuator pressure |
| | Linear Variable Differential Transformer (LVDT)[b] | Stem position |
| | Flow transmitter[c] | Feedwater flow |
| | Flow transmitter[c] | Steam flow |
| | Level transmitter[c] | Steam generator level |
| | Pressure transmitter[c] | First stage turbine pressure (nominal power level) |
| Valve and Valve Stem Vibration (Mechanical) | Accelerometer[a] | Valve stem-plug accelerations |
| | Accelerometer | Valve body accelerations |
| | Rosette strain gauge[d] | Dynamic strain in valve stem |
| Valve Hydrodynamic Characteristics | Acoustic emissions | Acoustic emissions |
| | Pressure transmitter[c] | Absolute outlet water pressure |
| | Pressure transmitter[e] | Absolute inlet water pressure |
| | RTD or thermocouple[c] | Feedwater temperature |

Notes: [a] Sensor was mounted on two separate FCVs.
[b] Sensor was mounted on two separate FCVs except for the first plant.
[c] Sensor was available as a plant signal.
[d] Sensor was available for 2 of 3 plants.
[e] Sensor was available for 1 of 3 plants.

System Hardware Components

The hardware components were procured and built to support the desired signal characteristics discussed below.

Development of Algorithms

The data handling algorithms can be placed in the following five categories:

- Signal characterization
- Pattern recognition
- Data storage
- Data display
- Off-line analyses

The signals were characterized by: (1) steady state or absolute (DC) value, (2) root mean square (RMS) value of the noisy component over various frequency bandwidths, (3) power spectral density (PSD), and (4) cross power spectral density (CPSD) with another signal. The CPSD was displayed in terms of coherence and phase.

The pattern recognition algorithms were developed to operate in three primary modes: (1) a learning mode to determine "normal" baseline signal characteristics; (2) a tracking mode to survey for abnormal signal characteristics; and (3) a transient mode for non-steady state operating conditions (e.g., power reductions). Data storage was developed to allow serial storage (for historical inspection) and to store all abnormal measurements. Data display was developed to displav the signal characteristics either graphically or numerically. Off-line algorithms for in-depth analysis were developed as necessary.

INSTALLATION AND OPERATION OF THE MONITORING SYSTEM

The monitoring system was installed in three operating PWRs for about 100 days each.* The first plant had an FCV with a cylindrical plug and diaphragm-operated actuator; the second plant had an FCV with a double globe plug and diaphragm-operated actuator; and the third plant had a cylindrical plug with a piston-operated actuator.

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*At the third plant, only 40 operating days were monitored.

Plant 1 had a chronic history of FCV problems including:

- Breaking stems
- Leakage through the stem packing
- Loss of control due to overtightened stem packing
- Significant stem and topworks torquing resulting in mounted equipment failure.

Plant 2 had few problems with their FCVs and was chosen for comparison purposes.

Plant 3 had moderate problems including:

- Leakage through the stem packing
- Loss of control due to overtightened stem packing
- Sudden, inexplicable rapid movements of the valve plug

Plant 3 was also a source of additional comparative information as a topworks changeout was done during the course of the monitoring measurements.

Significant Results

Six major results were obtained from the data analysis.

- FCV stem and bonnet vibration levels were correlated with the plants' historical problems; i.e., higher vibrations were measured on those FCVs which had a history of trouble, particularly at high frequencies.
- All the FCVs showed strong vibrational peaks at various points in a normal power transient.
- Tightening and loosening of the valve stem packing could be seen in strain gauge and accelerometer data.
- Strain gauges detected significant valve stem torquing at one of the plants.
- Conditions approaching cavitation were calculated, measured and verified.
- Significant differences were seen before and after the valve topworks changeout of Plant 3.

## FCV Vibration Levels

The vibrational data at higher frequencies revealed much higher stem-plug and valve body vibrations for FCVs at Plants 1 and 3 than Plant 2. The stem RMS accelerations at Plant 1 is nearly 47 g's over the frequency range 0-15 kHz (most of it over 3 kHz) whereas at Plant 3, the two stem RMS accelerations were 5 and 10 g's and at Plant 2, they were 0.9 and 1.3 g's. Figures 3 and 4 show the FCV stem RMS accelerations and integrated accelerations for each 1 kHz bandwidth at the three plants.

## FCV Vibrations During Power Transients

In some cases during power transients, several power levels could be identified where vibrations were higher than they were at full power. This phenomenon was particularly pronounced at Plant 2. Figure 5 shows the RMS stem accelerations band-passed from 0-20 and 20-1000 Hz respectively for a startup at Plant 2.

There are evident vibrational peaks at power levels from ~40-50% and ~75-85% particularly in the higher frequency range. This phenomenon was seen repeatedly during all transients and at the other plants.

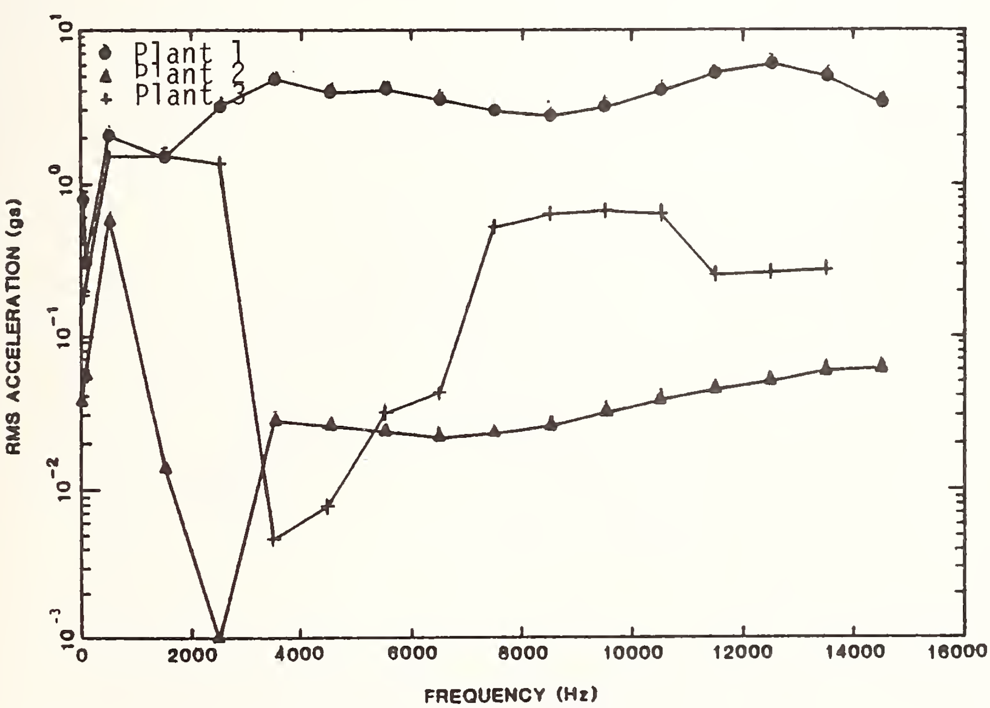


Figure 3. RMS Accelerations for the FCV Stem from 0 to 15 kHz

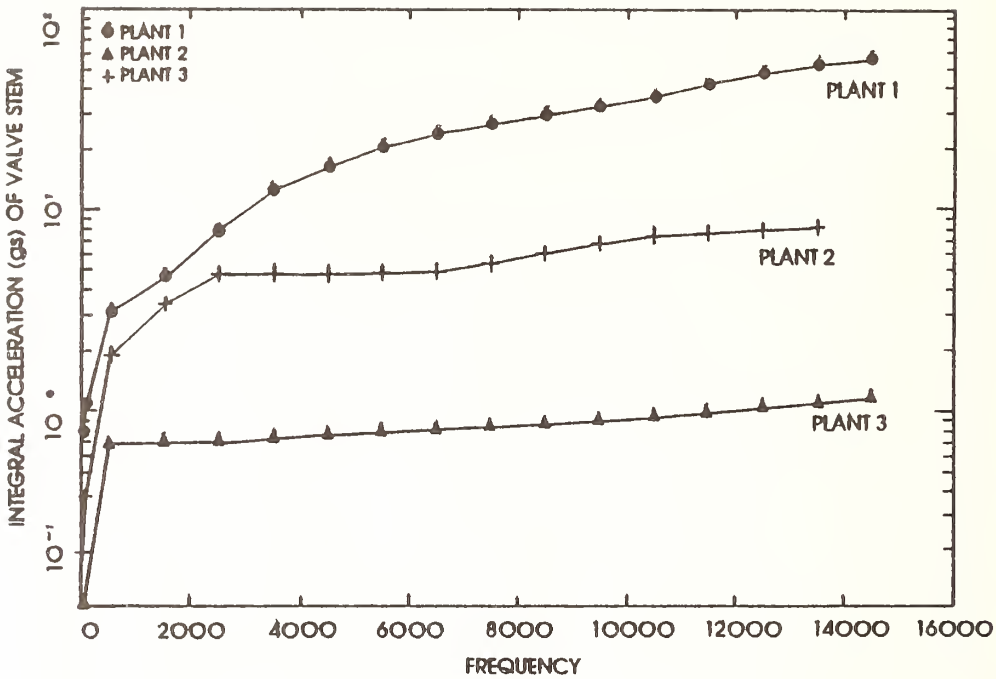


Figure 4. Integrated RMS Accelerations for the FCV Stem from 0 to 15 kHz

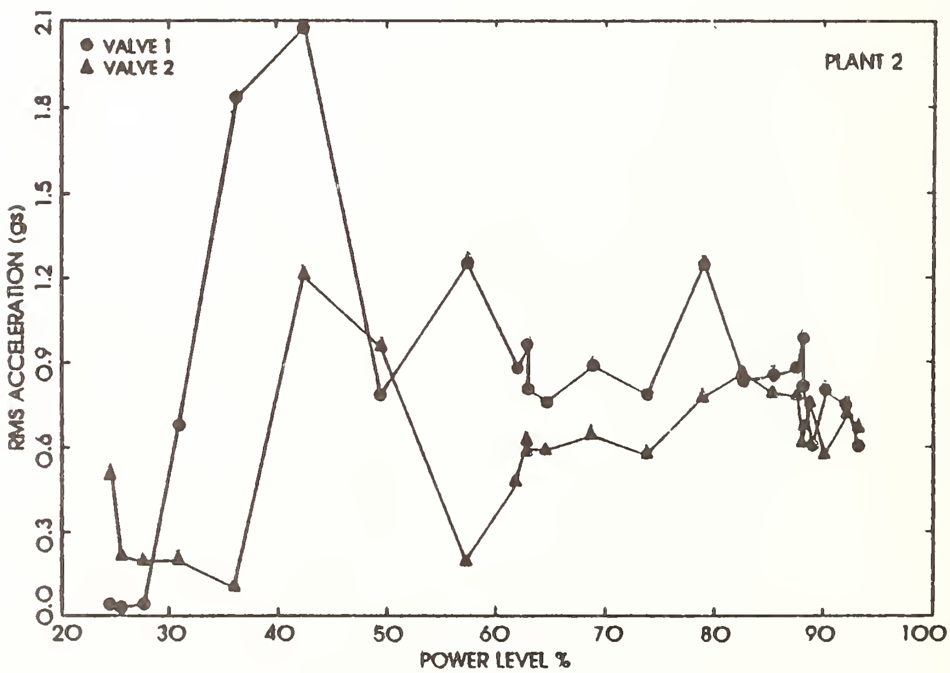


Figure 5. RMS Vibrations of the Valve Stems from 20 to 1000 Hz During a Startup at Plant 2

## Vibrations Related to FCV Stem Packing Tightness and Valve Torquing

At both Plants 1 and 2, stem vibration data was obtained which clearly indicated when the stem packing was tightened. At Plant 1, this vibration was due to an excited torquing resonance of the stem-plug assembly at about 14 Hz. Figure 6 shows the PSD of a strain gauge mounted 45° to the stem axis on four separate days. The peak at 14 Hz increased dramatically until the stem packing was tightened when it reduced to nearly its original level. The RMS value of this signal ranged from 38 μin/in on the 28th day to 123 μin/in on the 48th day to 45 μin/in on the 51st day. The tightening of the stem packing was also observed at Plant 2 by monitoring the stem accelerations.

## Cavitation Parameters

Cavitation can occur whenever the water pressure falls below the liquid vapor pressure. At Plant 3, calculations indicated cavitation conditions. Figure 7 shows the vena contracta pressure and the liquid vapor pressure for a transient to ~ 50% power and then back to 100% power. There were several times when the vena contracta pressure fell below the vapor pressure. This was reflected by the acoustic emissions whose 100 to 400 kHz RMS value increased as the pressure difference between the vena contracta and liquid vapor pressure decreased (Figure 8). This increase was attributed to the increasing water velocity through the trim ports.

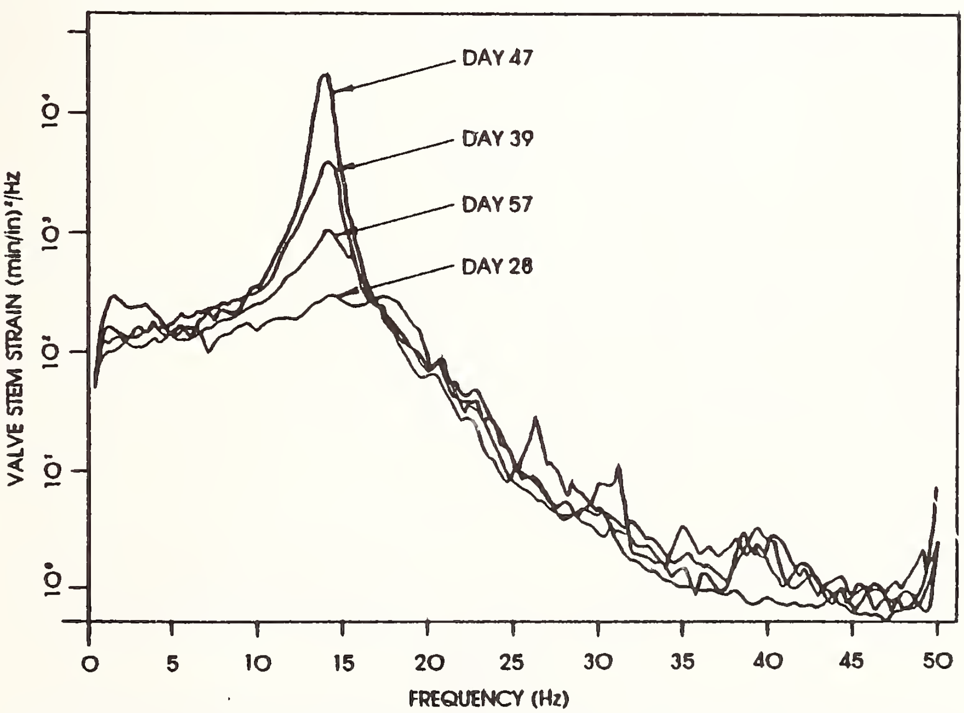


Figure 6. PSD of Stem Strain Gauge Showing Torguing Frequency at 14 Hz and Effect of Stem Packing Tightening on Day 48

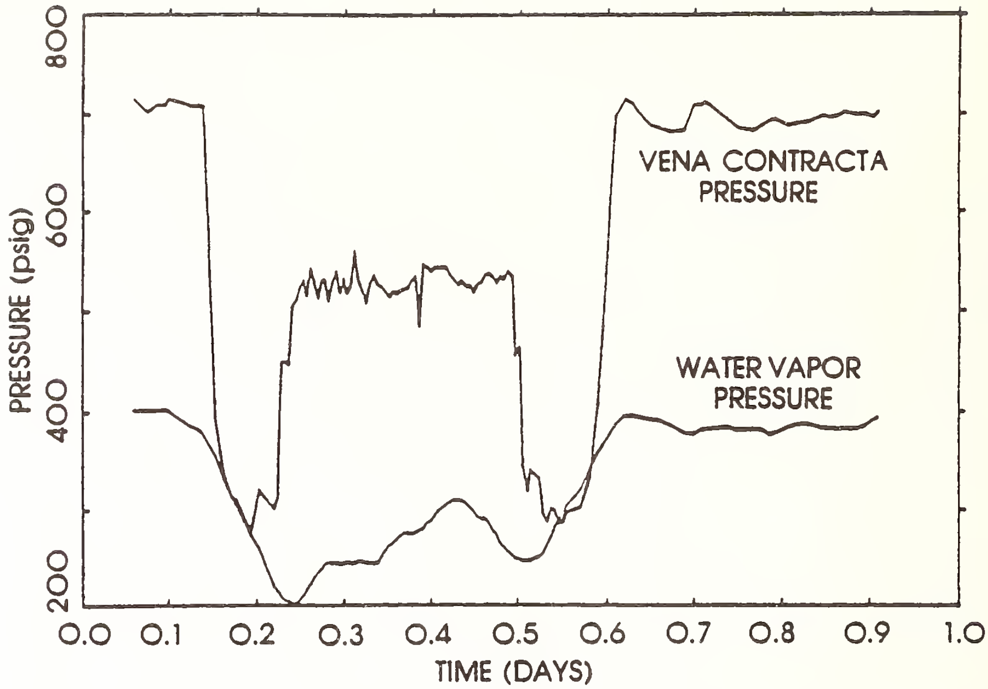


Figure 7. Water Vapor Pressure and Vena Contracta Pressure During a Power Transient

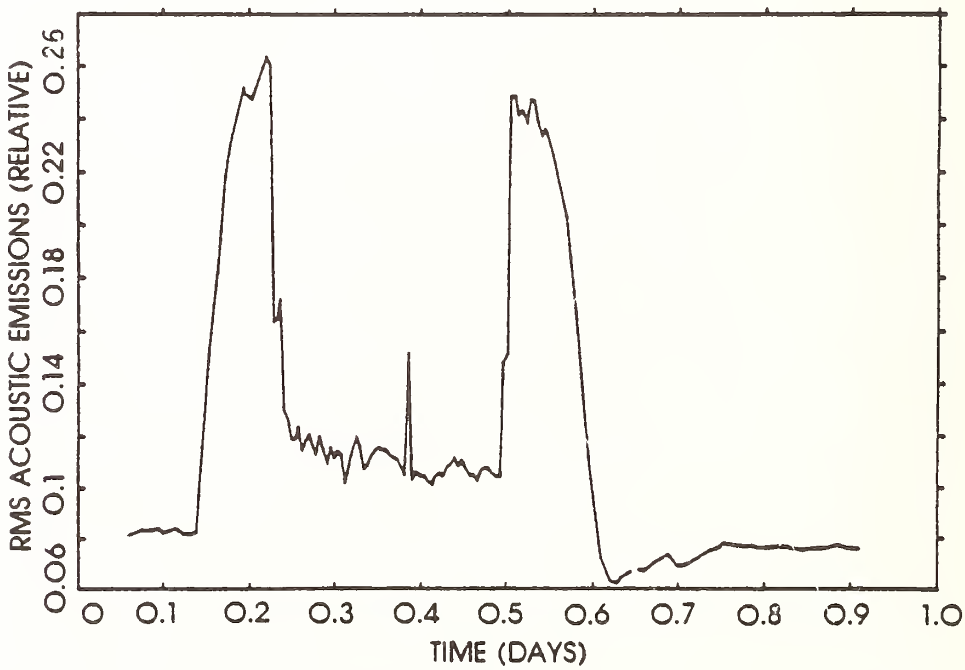


Figure 8. Acoustics Emissions During a Power Transient

## FCV Topworks Changeout at Plant 3

During the monitoring period at the third plant, the piston actuator was replaced with a different model. Four significant results were seen: (1) there was a large increase in the piston pressure; (2) the problem with sudden unexplained stem motion disappeared (as of this writing); (3) there was a corresponding significant increase in a mechanical vibration resonance in the plug and stem assembly at ~900 Hz; and (4) cavitation conditions appeared.

## CONCLUSIONS AND DISCUSSION

Three of the significant results pertained to phenomena which could be continuously monitored to assist plant operators in maintaining the FCV integrity:

- FCV vibration resonances during plant transients

- Vibrations related to the tightness of the stem packing

- Detection of cavitation conditions.

The ability to calibrate and monitor these phenomena would allow operators to avoid those conditions where high vibrations and/or cavitation may exist and to be able to determine when and/or how much to tighten the stem packing. The other two results fell primarily into generic and calibration design categories. Further investigation would need to be done by the specific plants and/or vendors to affect the adverse conditions.

## EVALUATION OF THE FCV MONITORING SYSTEM AND THE CONCEPTUAL DESIGN FOR A REDUCED SYSTEM

The FCV monitoring system was remotely located at three plant sites and successfully measured and evaluated large amounts of data on a continuous basis using customized measurement, storage and pattern recognition techniques. These algorithms formed the bases for determining the significant results discussed above. The FCV surveillance system is a state-of-the-art on-line monitoring system developed to prevent unnecessary plant outages.

The features designed for this system include the ability to:

- Alert the surveillance system operator to measured variables which do not conform to the normal state of affairs

- Use pattern recognition techniques to establish baseline data libraries and the expected ranges of the measured variables under normal conditions

- Measure and store a large number of variables

- Perform in near real time operation

- Provide historical information for long term trends

- Perform when the plant is in both steady state and transient operation modes.

The FCV surveillance system was designed to be general and flexible enough so that adjustments could be made during the course of operation. This was necessary because the nature of the signals chosen for monitoring was largely unknown. Therefore, it was necessary to use an overly large number of representative variables and trim down the number as their effectiveness was evaluated. Among those variables which were most useful in this evaluation were:

- The steady state, DC values of the signals

- Root Mean Square (RMS) values of the AC component of the signals

- The power spectral density (PSD) of the signals.

Several variables were found which were not of significant value. These were:

- Some variables comprising the controller (steam generator level and steam flow)

- Calculated time series variables for controller representation

- Low frequency power spectral densities (CPSDs) displayed as coherence and phase.

Concept of a Reduced System

The significant events which occurred suggest three functions which could be monitored by a reduced system:

- Maintenance of proper stem packing tightness

- Avoidance of vibrational or torquing resonances during transient and normal operations

- Maintenance of an adequate margin from incipient cavitation conditions.

In addition, there was a lack of significant results concerning the valve controller. Since known problems have occurred in the controller, a fourth function should be considered:

- Assurance that positioner and actuator functions are performing as expected.

The required sensors for such a reduced system would be a subset of those used in the general system described in the main report. These sensors are shown in Table 5. In addition to the above sensors, a "status" sensor should be chosen to indicate gross operational conditions, e.g., power level.

Table 5

SENSORS REQUIRED FOR A REDUCED MONITORING SYSTEM

| Sensor | Location | Function Monitored |
|---|---|---|
| Accelerometer | Stem | Packing, vibration |
| Accelerometer | Bonnet | Vibration |
| Acoustic emissions | Bonnet | Packing, vibration, cavitation |
| Flow transmitter | Feedwater line | Cavitation |
| LVDT | Stem | Cavitation, controller |
| Pressure transmitter | Inlet feedwater line | Cavitation |
| Pressure transmitter | Outlet feedwater line | Cavitation |
| Pressure transducer | Positioner pressure line | Controller |
| Pressure transducer | Actuator pressure line | Controller |
| Rosette strain gauge | Stem | Packing, vibration, twisting |
| Thermocouple/RTD | Feedwater Line | Cavitation |

Microprocessors could be designed for the individual functions above. In order to monitor all of the functions, a mini-computer designed for field use in a non-optimal environment may be required. Furthermore, the computer peripherals, sensors and signal conditioning, should also be resistant to such environmental conditions as heat, dirt and non-routine maintenance.

# SESSION II

## TECHNIQUES FOR DETECTION

Chairmen: D. Fairchild, Fram Corporation

R. M. Whittier, Endevco Corporation

# OIL DEBRIS DETECTION PROGRESS

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Abstract: For the last two years, the U.S. Army has been testing an Advanced Oil Debris Monitoring System on a fleet of 36 UH-1 helicopters at Fort Rucker, Alabama. To date, approximately 25,000 flight hours have been logged. The system has successfully detected all failures of oil-wetted engine and transmission components, most of which were missed by the conventional chip detectors and by SOAP. These include transmission input quill and mast bearing failures as well as engine No. 2, 3 and 21 bearing failures. In all cases, the system has given early, unambiguous and repeated failure alert. Component removal in early response to these alerts has resulted in very limited or no secondary damage. In some instances, only the affected component required replacement.

Key words: Chip detectors; condition monitoring; failure detection; helicopter diagnostics; oil debris; wear monitoring.

The benefits of improving existing wear debris monitoring techniques are well-known and easily understood. For instance, the U.S. Army estimates that 86% of all chip lights in its helicopter fleet are erroneous and therefore cause unnecessary mission aborts(1). Despite a logistically complex and expensive Condition Monitoring Program consisting of Spectrometric Oil Analysis, chip lights, filter checks and pilot inputs, unnecessary component removals and failures progressing to considerable debris and secondary damage are a problem.

In 1978, the U.S. Army began testing an advanced debris monitoring system at its Aviation School in Fort Rucker as part of the so-called OLSAFE program. Initially involving 60 helicopters of its instrument flight training center, the program originally had a different main objective. It was designed to demonstrate that ultrafine filtration will permit the extension of oil change intervals in the engine from the current 100 to 1000 hours and in the transmission from 300 to 1000 hours and thereby save the Army a considerable amount of Mil-L-23699 oil.

Previous experience on T-53 engines equipped with ultrafine filters and with the General Electric T-700 engine with its 7-micron filter had indicated that Spectrometric Oil Analysis becomes virtually ineffective. This is due to the fact that these filters remove the recirculating fine debris which gradually builds up as a failure progresses and which otherwise could lead to SOAP indication.

Considering the low reliability of existing chip detectors, the need for an improved oil debris detection system became clear. The system which was eventually developed to meet this need consists of:

- Full-flow chip detectors in engine and transmission
- Pulsed chip detection system ("Zapper" or "burn-off") throughout the aircraft
- Separate cockpit indication parallel to and redundant with existing chip light installation

## FULL-FLOW CHIP DETECTION

In older installations, such as the T-53 engine and the UH-1 transmission, chip detectors are of the splash-type. This means that they are located in the sump wall, usually with no provision to ensure that wear debris is transported to them. The magnetic attraction of the chip detector alone cannot pull the debris from the corners of the sump. That splash-type chip detectors are not very effective is illustrated by the fact that oil filters, coolers and oil lines usually contain much more debris after a failure than the chip detector.

The principle of full-flow debris monitoring places the chip detector into the scavenge oil flow where the entire flow can be intercepted. Figure 1 compares the splash-type chip detector with a full-flow chip detector located in the pump inlet screen. The oil, on its way to the pump, deposits the debris in the screen where it is indicated by the chip detector. A full-flow chip detector installation takes advantage of the oil being the debris transporting agent, in addition to its lubricating and cooling functions.

In the OLSAFE program, the engine and transmission full-flow chip detectors are located on the pressure side of the scavenge pumps. Figure 2 shows the lube system schematic of the T-53 engine with the location of the full-flow debris monitor. The conventional chip detector is located in the accessory gear box and its function is retained. The full-flow debris monitor, shown in Figure 3, is actually a centrifugal debris separator. The oil enters tangentially into a cylindrical chamber and creates a vortex whose centrifugal action separates the heavier wear debris from the oil and deposits it on the chip detector at the bottom of the chamber. The high debris capture efficiency of this device is illustrated in Figure 4.

The transmission full-flow chip detector is shown in Figure 5. It replaces the disc-type filter located inside a transmission cavity. Since the disc filter also has the function to protect an oil jet downstream, the chip detector has an internal screen through which the oil flows back into the transmission.

## PULSED CHIP DETECTION SYSTEM

The majority of erroneous chip lights of conventional chip detectors are caused by "wear fuzz". This fine debris is the product of normal wear processes.

Pulsed chip detection systems are designed to suppress these erroneous chip lights. This is accomplished by a current pulse which discharges through the chip detector, melting the contacts between the fine wear debris but leaving larger, failure-related particles unaffected. This type of chip detector also has come to be known as "Zapper" or "burn-off" chip detector.

The engine and transmission, as well as the tail rotor and intermediate gear box chip detectors in the OLSAFE program are of the pulsed type. The necessary electronics are located in the cockpit display panel which is shown in Figure 6.

For reasons of flight safety, the pulsed chip detection system is completely redundant with the conventional chip detection system of the aircraft. The cockpit display panel originally had a manual pulse switching feature, as indicated in Figure 6. This allowed the pilot to initiate the pulse. However, the program has shown that the failure indication reliability is very high and all panels have since been converted to the automatic mode. This means that chip indications caused by fuzz or fine slivers are suppressed automatically.

STATUS OF OLSAFE PROGRAM

As indicated previously, the major objectives of the program are as follows:

- Reduce no-cause component removals (engines, transmissions, gear boxes)

- Reduce mission aborts due to erroneous chip lights

- Increase oil change intervals from 100 (engine) and 300 hours (transmission) to 1000 hours.

The program is currently configured as follows: 36 UH-1 helicopters belonging to the U.S. Army Flight Training School at Ft. Rucker have been modified with improved debris detection systems and ultrafine filters. This fleet logs approximately 2000 hours per month and has attained a total of about 25,000 flight hours. An additional control group of eleven unmodified helicopters flies under identical conditions. The high-time aircraft in the test group has approximately 1000 hours on engine and transmission. In the control group, there is a high-time engine with 1700 hours and a high-time transmission with 1300 hours.

In both groups, the oil is not changed until component overhaul. The condition of the oil is determined from samples taken every 50 hours which are analyzed at the Naval Air Propulsion Test Center (NAPTC) in Trenton, New Jersey. All aircraft are also within the Spectrometric Oil Analysis Program conducted at Ft. Rucker.

Following a chip light indication, the chip detector is removed and replaced with a clean one. The contaminated chip detector is photographed and the debris analyzed at the Joint Oil Analysis Program (JOAP) Center in Pensacola, Florida. If the component is removed, the tear down inspection results are carefully documented. In this way, the information loop consisting of chip light/oil analysis/debris analysis/tear down result is closed. This is one of the major accomplishments of the program.

Approximately one year into the program it was realized that some chip indications were caused by debris reintroduced into the engine and transmission by inadequate design of the bypass system on the ultrafine filters. This required substantial modification of the filters and of the engine full-flow debris monitor. To date, 22 aircraft have been retrofitted with the improved filters. Accordingly, data on erroneous chip light indications cannot be considered reliable at the present time.

RESULTS

The improved chip detection system has shown a 100% reliability in detecting failures of oil-wetted components. Table 1 lists eight of the thirteen failures which have occurred in the test group and for which tear down inspection records are available. It is interesting to note the wide variety of failures. They include transmission mast bearing, input quill bearing and gear failures, as well as engine shaft bearing and accessory gear box bearing failures. In all cases, the full-flow chip detectors provided early and repeated warnings. Where the component was removed soon after the start of indications, secondary damage was minimal. None of these failures was detected by SOAP; only one was indicated by the conventional chip detector. However, even this indication was preceded by warning from the full-flow chip detection system.

A typical engine failure is illustrated in Figure 7. The failure was caused by a rotating No. 2 shaft bearing. Upon removal, the outer bearing race was found to have.006 inch deep wear ridges. Otherwise, the bearing was in good condition. None of the other shaft bearings or any of the accessory gear box components showed secondary damage. The debris causing the indication is shown in Figure 8. It should be noted that failures of this type which have been thought to produce mostly fine debris have generally been considered not detectable with chip detectors. This opinion is clearly based on splash-type chip detectors which would have had no chance to collect the small amount of debris shown in Figure 8.

A second failure, this one in the transmission, is illustrated in Figure 9. This failure was caused by spalling of the inner bearing in a triplex input quill set. The failure had started long before the conversion of the aircraft. Installation of the full-flow chip detector led to immediate and repeated indications. When the transmission was removed, there was considerable debris damage to bearings and gears, as a result of the extensive operation with the failing bearing prior to aircraft modification.

The debris from a transmission mast bearing failure is shown in Figure 10. The transmission was equipped with the full-flow chip detector for a period of 301 hours. During that time, there were eight chip lights. As a result of the

| Component | Failure | Full-Flow Chip Det. Alert | SOAP Alert | St'd. Chip Det. Alert | Remarks |
|---|---|---|---|---|---|
| Engine | No. 2 bearing turning | Yes | No | No | No secondary damage. |
| Transmission | Input quill bearing spalled | Yes | No | No | Failure started before conversion led to immediate indication. |
| Engine | Torquemeter cylinder scored; loose No. 3 bearing | Yes | No | No | Conversion led to immediate indication. Would have caused serious failure if left undetected. |
| Engine | Torquemeter cylinder scored; loose No. 2, 3 bearings | Yes | No | No | Little secondary damage. |
| Transmission | Mast bearing spalled | Yes | No | No | Progressing failure followed for 301 hours. 8 chip lights. |
| Transmission | Input drive pinion plus bevel gear failure | Yes | No | No | Failure followed for 3 chip indications. |
| Transmission | Mast bearing spalled | Yes | No | No | Failure followed for 190 hours. |
| Engine | No. 21 bearing failure | Yes | No | Yes | Chip indications started 130 hours prior to removal. |

TABLE 1: PULSED CHIP DETECTOR EVALUATION:
UH-1 ENGINE AND TRANSMISSION FAILURES

extended operation with the failure in progress, there was some debris damage. It is worth noting again that in the UH-1 transmission a mast bearing failure is not reliably indicated by the splash-type chip detector. The sun gear and other components of the gear train collect much of the debris on its way down into the sump. With the splash-type chip detector off to the side of the transmission and somewhat above the sump floor, its capture efficiency is much lower than that of the full-flow chip detector.

## CONCLUSION

Despite the fact that the OLSAFE program has not been concluded, it has demonstrated that oil-wetted component failures are indicated reliably and with long warning periods by full-flow chip detectors. This points the way to meeting a major program objective, namely the development of unambiguous component removal criteria and elimination of no-cause overhauls. During the remaining test time, determination of the frequency of erroneous chip lights will be the main objective. This has not been possible due to the filter reconfiguration mentioned earlier. The program has further demonstrated that the oil change interval can be extended safely.

Above all, the program has shown that full-flow debris monitoring represents a significant improvement in Failure Detection.

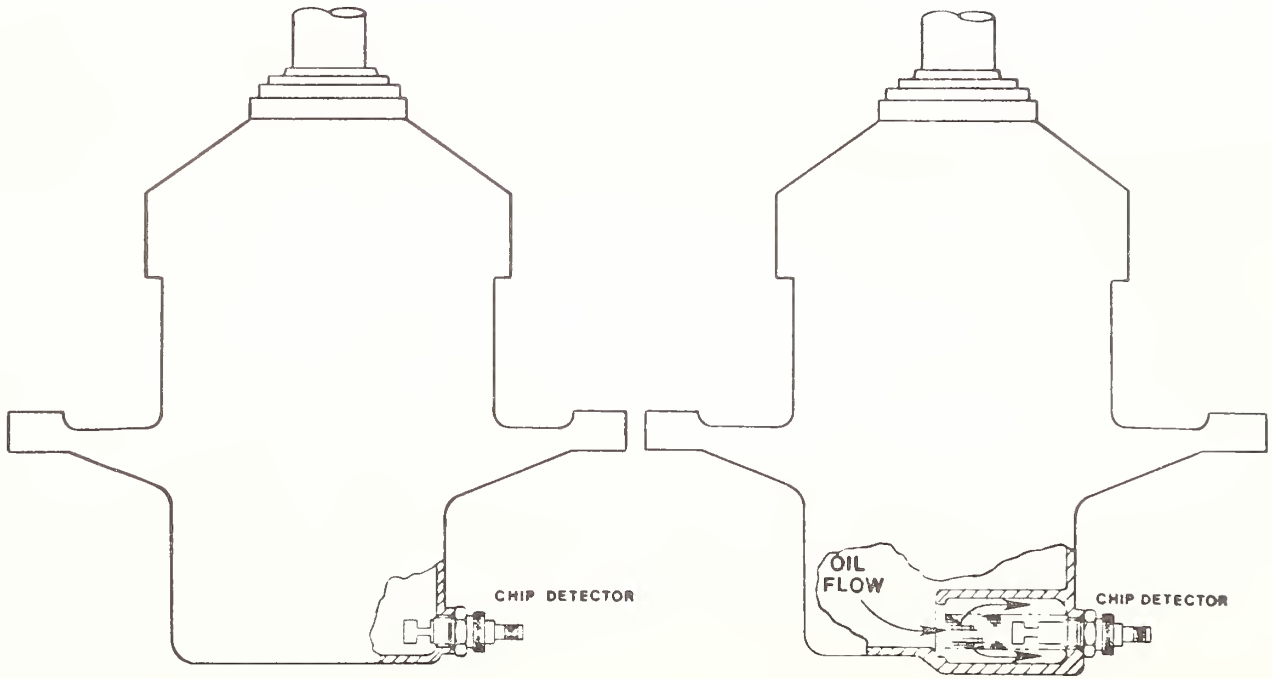
## ACKNOWLEDGMENTS

The author wishes to thank Wayne Hudgins of the Ft. Eustis Applied Research and Technology Laboratories, Rhett Farrior of the U.S. Army Aircraft Development Test Activity, Ft. Rucker, Dick Lee of the Joint Oil Analysis Program, Pensacola and Dominic Lubrano for valuable contributions to this paper.

## REFERENCE

1. D. P. Lubrano: "Field Experience Of Contamination Effects And Diagnostic Effectiveness On Helicopter Transmissions And Engines", 1977 Meeting of ASLE, Atlanta, Ga.

FIGURE 1: **FULL-FLOW DEBRIS MONITORING**




**SPLASH TYPE CHIP DETECTOR** **FULL-FLOW CHIP DETECTOR**

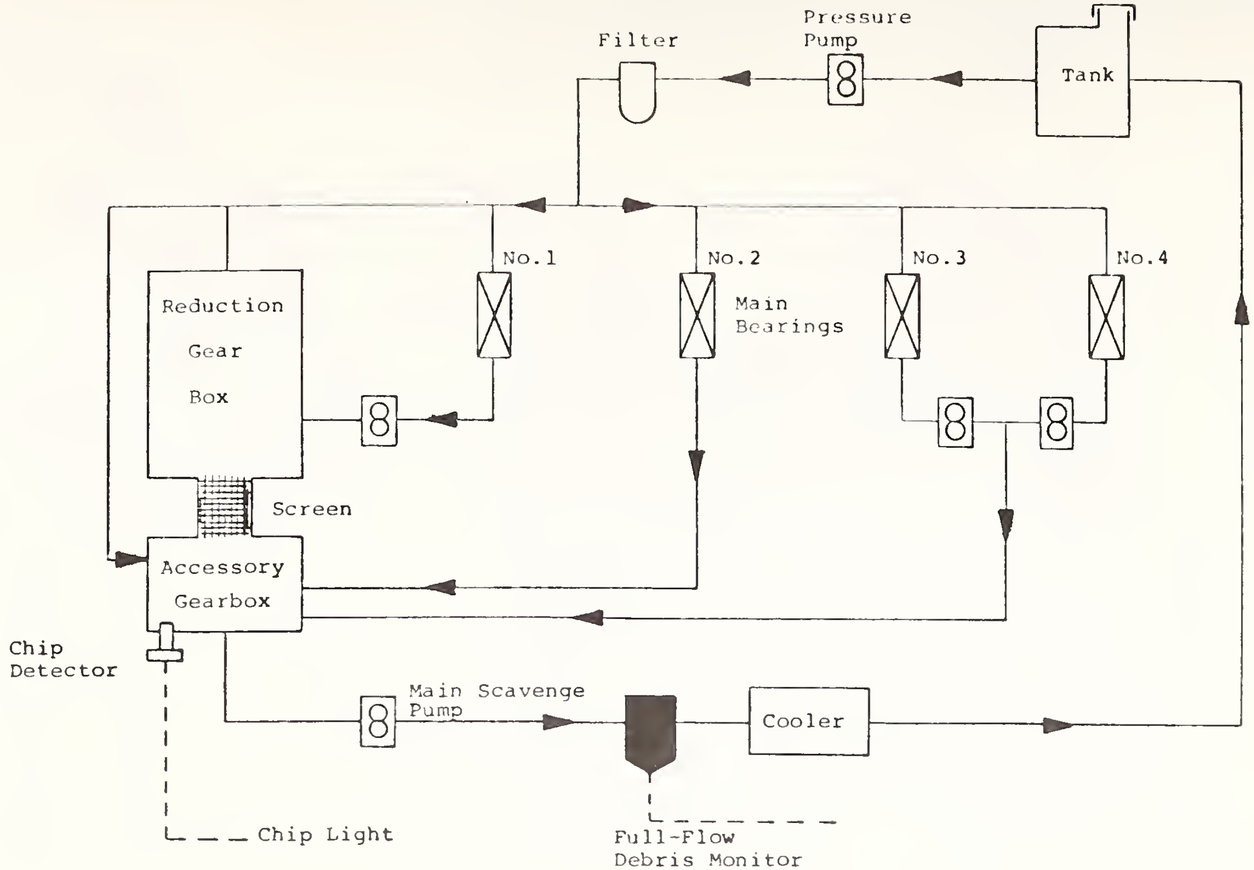



FIGURE 2: T-53 ENGINE WITH FULL-FLOW DEBRIS MONITOR

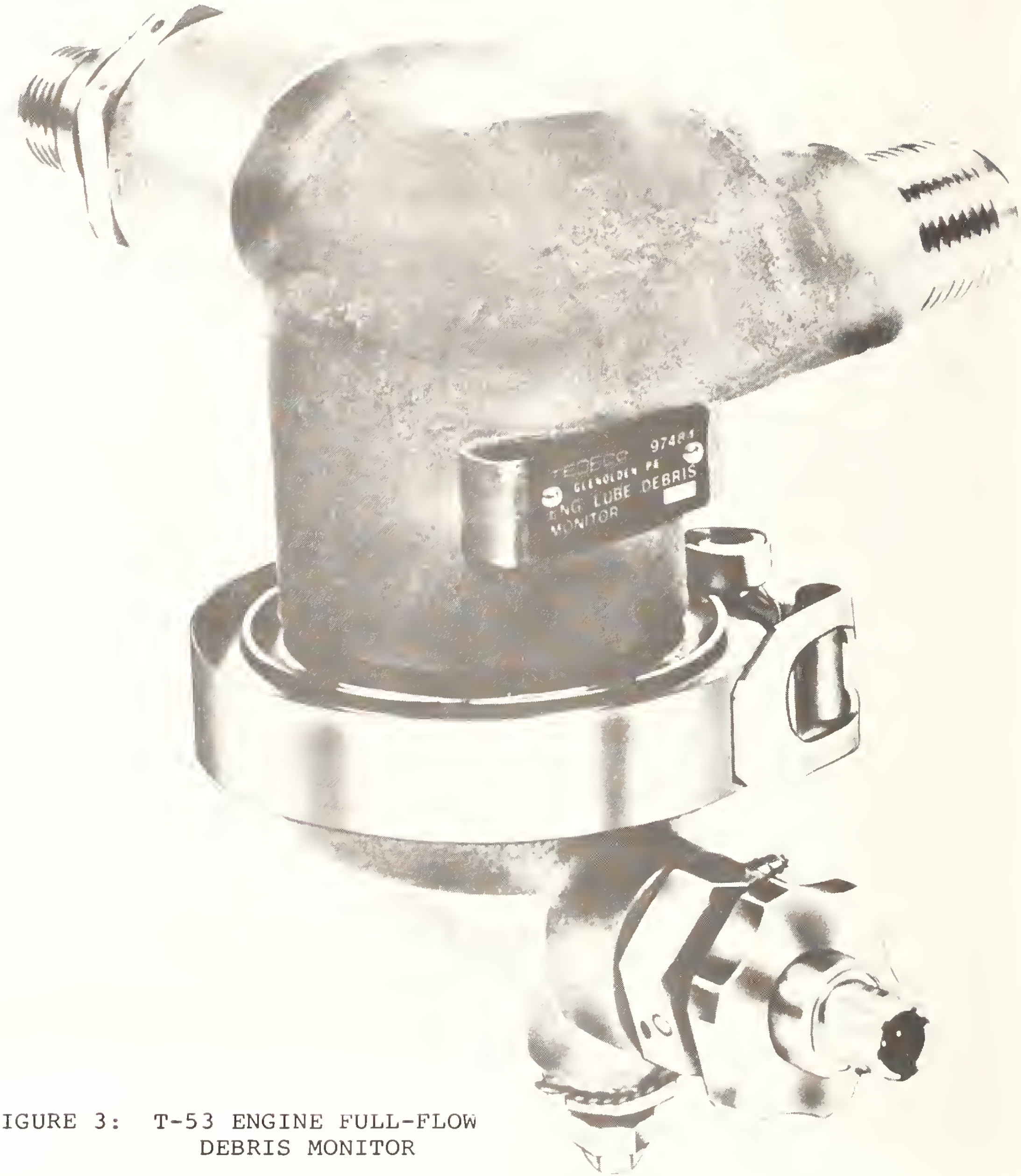



FIGURE 3: T-53 ENGINE FULL-FLOW DEBRIS MONITOR

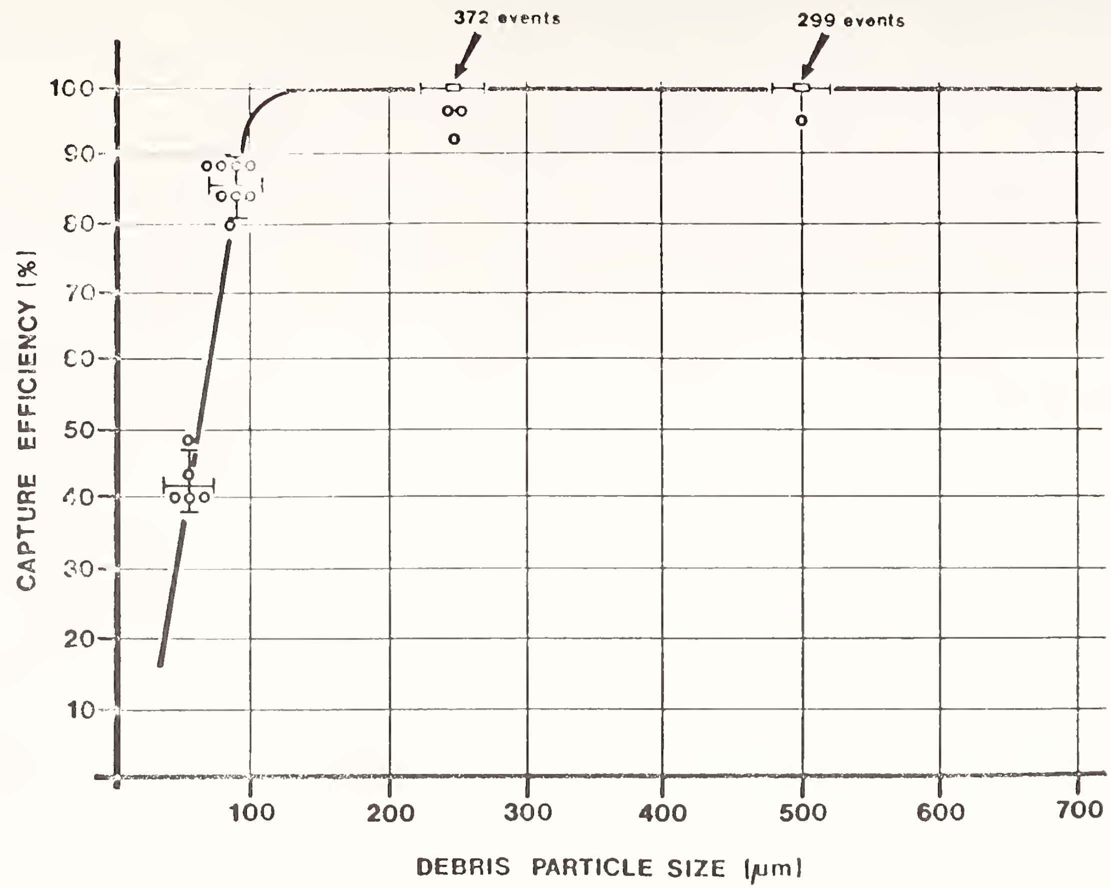



FIGURE 4: DEBRIS CAPTURE EFFICIENCY OF ENGINE DEBRIS MONITOR

FIGURE 5: TRANSMISSION FULL-FLOW CHIP DETECTOR


FIGURE 6: COCKPIT DISPLAY PANEL

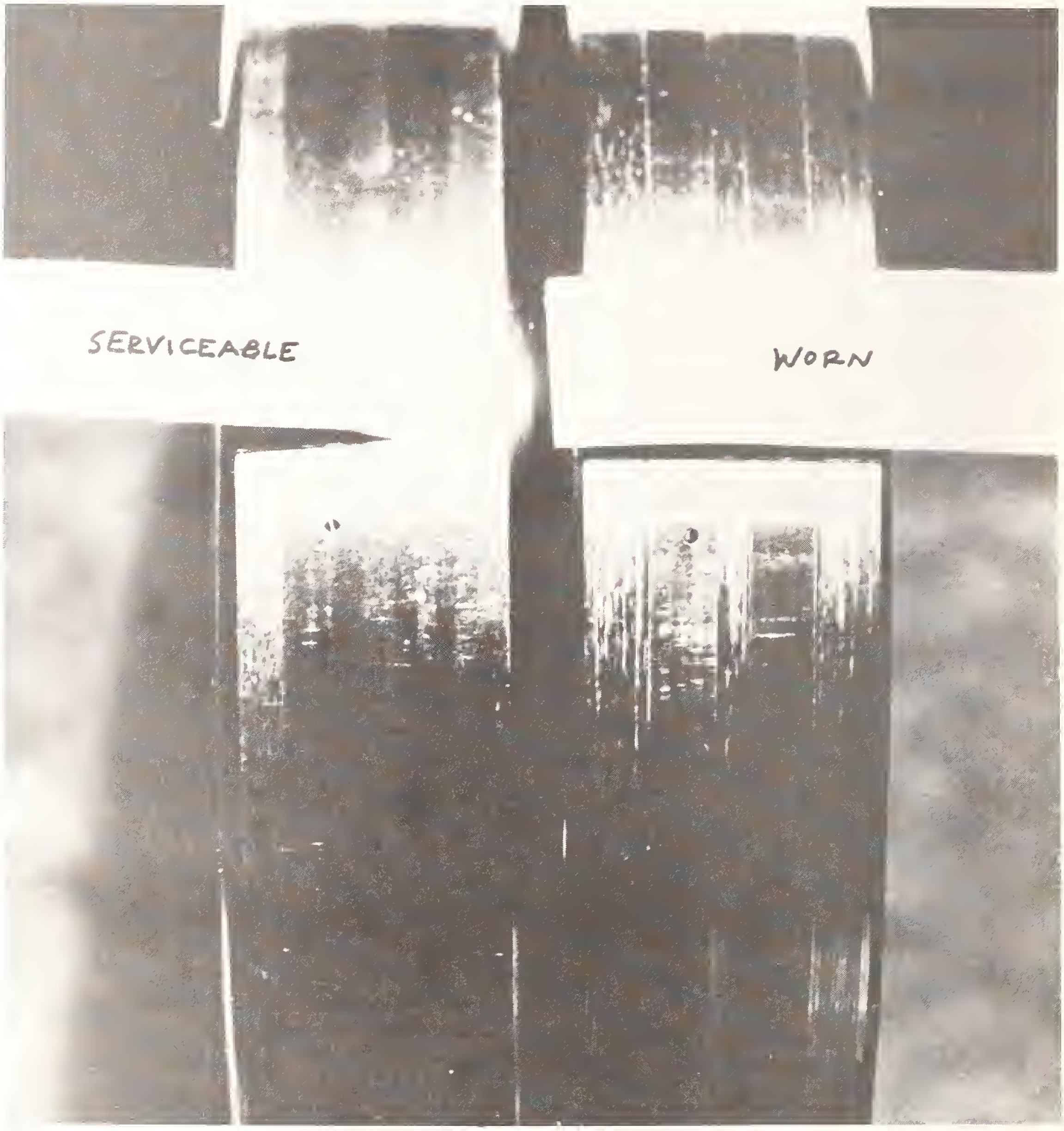



FIGURE 7: ROTATING NO. 2 BEARING, T-53 ENGINE

FIGURE 8: DEBRIS FROM FAILURE ILLUSTRATED IN FIGURE 7

I/B Brg of Brg Set
P/N 205-040-246-3
S/N 3173

FIGURE 9: INPUT QUILL BEARING FAILURE, TRANSMISSION

FIGURE 10: DEBRIS FROM TRANSMISSION MAST BEARING FAILURE

# ADVANCED RADIOACTIVE ENGINE WEAR ANALYSIS FOR FILTER DESIGN AND EVALUATION

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Abstract: Tests were performed to evaluate the effect of filtration on the wear rates and particle size sensitivities of nine components within an engine. The wear measurement method used was a thin layer radioactive activation technique which had been proven in previous engine wear studies. In this test program, several advances were made in the wear analysis technique itself, which permitted measurements of one micrometer of wear on the components. These advances were primarily in the test technique and data analysis areas.

The tests performed yielded results significant to filter design and evaluation. The most important conclusion reached in regard to filtration was that particle wear sensitivity varied for different engine components. This variability included different particle size sensitivities and different effects of filtration on the engine components. This variation was used in the tests to demonstrate the different wear protection values of two filter media for several engine parts.

Key Words: Oil filter testing, oil filter design, radioactive wear testing, engine wear analysis, particle size sensitivity, effect of filtration.

The life blood of an engine is the oil used to lubricate it. As this oil circulates through the engine, it insulates surfaces against wear. In order to do this job effectively, the oil must be kept clean; this is the function of the oil filter. How may we determine whether an oil filter is protecting the oil and the engine? There are many bench test procedures available for oil filters which have been written by manufacturers and technical societies. These procedures are valuable for filter to filter comparisons, but they lack a direct correspondence to actual engine protection in the field. The procedures cannot tell us whether a filter will prevent wear within an engine.

One method to evaluate the wear protection of a filter would be to measure the wear surfaces then measure what wear has occurred on engine parts during a test by disassembling the engine. This would be a difficult and time consuming project. It would also introduce more variables since when the relative positions of wearing components are changed, the wear rate also changes until the parts have broken in again. A better measurement method would be one in which the engine need not be taken apart for the measurements to be made. This is the advantage of radioactivity as a measurement tool; it penetrates the engine walls, permitting measurements of the parts inside.

This paper presents the results of a test program run on a diesel engine to examine the effect of contaminant on its wear rates and the influence of filtration on preventing this wear. The long range goal of the program was to determine methods which would allow more effective filter design to meet the requirements of an engine.

The SPI-Wear™ Method

The radioactive measurement method used in the tests was the SPI-Wear technique devised by the SPIRE Corporation of Bedford, Massachusetts. The method was developed to make measurements of surface wear in areas which are inaccessible or where conditions make physical measurement methods impossible. The original application of the technique was to measure the erosion of missile nosetips during atmospheric re-entry. It permitted wear measurements to be made during the flight itself. The measurement system consists of two parts, an activated volume of the test component, and a gamma ray detector and readout instrument. The detector measures the radiation being emitted by the activated material, which is proportional to the depth of material remaining. This radiation measurement, or count rate, may be related to wear depth by a calibration curve.

The main advantage of this method as a measuring tool is the way in which the radioactivity is used. The use of radioactivity as a wear measurement technique is not new, it has been used in many applications in the past (1)(2). Most previous uses relied on a bulk activation of the test component in which a large volume of the part was made radioactive. This large amount of activated material caused the emission of high radiation. This presented safety problems to the test personnel and, because of the large activated volume, sensitivity to wear was poor. In some techniques, the wear debris was monitored rather than the part itself in order to increase sensitivity. This presumed that no radioactive material was lost between the occasion of wear and the measurement. The SPI-Wear method solves these problems by using a very small amount of activity.

Activation of the wear measurement point is accomplished by bombarding the area with a beam of high energy particles. These particles may be alpha particles, protons, deuterons or helium-3, all of which are generated in a cyclotron. These high energy charged particles can activate most metals in common alloys. The beam can be controlled so that it activates a very selective volume of the test specimen. This is controlled by varying the area and kinetic energy of the beam. The activated spot can be as small as 1.6 mm in diameter and as shallow as 7.6 μm. This allows a very small area to be monitored for wear with a high degree of sensitivity. The limitations of this activation technique are that the material can be activated and that the beam must reach the test spot with no obstructions. The latter requirement necessitates the disassembly of the engine parts for activation. Once the activated parts are installed, however, no further disassembly is required.

Since the technique can carefully control the site of the activation and its intensity, it is possible to test several different wear points within an engine at the same time. This is especially true if the components to be monitored are of different materials, leading to different radiation spectra. In any case, when more than one spot is monitored at a time, cross-talk is an important consideration. This may be minimized through the location of the test elements, shielding of the detector, or through mathematical derivations during the data analysis. This ability to monitor more than one test spot at a time permits several different areas of an engine to be monitored under identical conditions.

Another advantage of this system is that it can produce depth measurements by relating radiation and wear depth through a calibration curve. This curve is generated through the careful wearing and measurement of a sample identical to that under test. These identical parts are irradiated at the same time so that their activation volumes and energies are the same. The calibration part is then manually worn while being measured for depth of wear with a corresponding radiation count. These points generate the calibration curve.

Previous Engine Testing Experience

The test program on a diesel engine described in this paper is the third set of tests done by Fram and SPIRE using this technique. The previous two test programs used an eight cylinder and a four cylinder gasoline engine (3). These tests were done to establish the applicability of the SPI-Wear technique to engine wear measurements and to further refine test methods so that the best possible data would be obtained in further testing. These tests showed that the SPI-Wear method could be used to measure engine wear. They also demonstrated that multiple point monitoring was feasible provided careful fixturing and test controls were used. Most of the problems which had to be resolved in these tests involved how the engine was run and how contaminant was introduced into the engine. With the problems resolved, however, these tests demonstrated conclusively the value of the filter in preventing wear. Conclusions about the filter's role in wear prevention could also be drawn from the results which showed a varying wear response for test components to different contaminant particle sizes. The results of these previous test programs provided an excellent foundation for the diesel engine tests and showed which paths new filter investigations should take.

The Diesel Test Program

This test program was undertaken to gain more information about filtration and its role in preventing engine wear. A primary goal of the program was to better define how a filter should be designed to protect an engine. Based on past testing experience, a test schedule was drawn up which called for monitoring nine different parts within a V-8 direct injection diesel engine for wear responses. The nine parts were:

1. oil ring lands
2. top compression ring surface
3. compression ring groove
4. intake valve sealing surface
5. fuel injection pump plunger
6. rocker arm bearing
7. wrist pin bearing
8. main bearing
9. connecting rod bearing

The tests performed were grouped under three main topics. The first series was used to establish the engine wear rates with contamination in the oil and the effect of filtration on these wear rates. The second series investigated the effect of different particle sizes on engine wear. The third test series investigated the comparative life and efficiency of two filter media types on the engine. In all, twenty different tests were run on the engine, with measurements of all nine points taken after each test.

In previous testing, two main problems had been observed: the method of introducing contaminant into the test engine and the method of maintaining test part to detector relationships. Previous findings had indicated that the best method of introducing contaminant was in a slurry form directly into the lubricating oil system. In these tests, the test contaminant (AC Fine Test Dust or fractions thereof) was mixed with a quantity of oil to form a slurry then pumped with a peristaltic pump to the engine oil pump inlet at a controlled rate during testing. It was found that this method offered the best repeatibility and eliminated problems of contaminant agglomeration.

The problem of maintaining a constant relationship between the test part and the detector is dealt with in two ways. Since all of the test components could move relative to the engine block, all had to be brought to a reference position each time a measurement was made. Since all the parts are mechanically tied to the engine crankshaft position, this was used as the reference to return all parts to their measuring position. Each time measurements were made, the crankshaft was brought to the reference position and pinned there. This insured that each part was in the proper position for measurement.

The mounting of the detector itself is the second part of the problem. The solution used was to prepare a detector mount bolted to the engine block in each of the nine test locations. The detector was mounted in a lead lined sleeve which insured that it would be attached to the engine in the proper position for each measurement. The lead sleeves also helped minimize the problem of cross-talk between test locations by limiting the field of view of the detector to the test component at that particular location.

## Application of SPI-Wear to Diesel Testing

This set of tests was one of the most ambitious engine program applications for the SPI-Wear method. To be successful, a great deal of care had to be taken in all phases of the method. Many new techniques were designed so that errors were further minimized from previous testing, making even finer measurements of wear possible. All of the planning and new technique development of the SPI-Wear method for this program was done by F. L. Milder of the SPIRE Corporation. Through his efforts, the measurements taken during this program were the most accurate possible.

In this program, the SPI-Wear method was tailored and improved to meet the requirements of wear measurements at nine different locations in the engine. The program followed the basic outline of the method as described before, but used different techniques in many areas to improve the data obtained.

The oil ring, compression ring, wrist pin bearing, piston, valve, rocker arm bearing and the fuel injection pump plunger were all irradiated using an alpha particle beam. The depths of activation ranged from 20 micrometers for the fuel injection pump plunger to 100 micrometers for the compression ring. These depths of activation were sized for the projected wear rate for each part. Each of the parts, with the exception of the valve, was irradiated with a 6.4 mm diameter spot in the location projected to be the maximum wear point for that part. The valve was irradiated over the full sealing surface.

A problem was encountered with the main and connecting rod bearings activation. Because of their material, a lead tin alloy, activation by the particle beam method was not possible. The bearing material would not activate sufficiently to overcome the signal produced by the iron impurities in the surface of the bearing, which were also activated. So that data could still be obtained from these two points, a different activation technique was used. This involved the implanting of four tin wires plated with $^{60}Co$ into each bearing. The diameter of these wires was 0.38 mm with the sides being plated for a length of approximately 0.25 mm. The four wires for each bearing were implanted in a 6.4 mm diameter area. These wires provided sufficient activity for measurements to be made of these parts.

One phase of the method which underwent the most scrutiny was the process of curve fitting for both the calibration curves and the radiation spectra. Errors in these phases must be minimized to reduce errors in the conversion from count rate to wear depth. For the calibration curves, the fit was to a polynomial function of arbitrary order. The order of the equation finally chosen was that which had the smallest chi-square per degree of freedom while still being a physically reasonable curve. For the spectra curve fits, a peak or pair of peaks was chosen for each of the parts monitored. These peaks were then fitted with a combination of two functions, one for background and a gaussian for each peak. For both the calibration and spectra curves, errors were determined and used in the final error margin calculations.

With nine irradiated spots within the engine, there was a question of whether crosstalk would be a problem. To account for this effect, each part was put into the engine, one at a time, and count rates were taken at the monitoring points for all nine parts. These count rates gave the amount of interference due to each spot at the monitoring points for the other eight spots. In testing, it was found that there was a significant crosstalk between the main and connecting rod bearings and the wrist pin bearing and compression ring. Because of the interference matrix generated before testing, the actual strength values for these components could be derived during the data analysis.

## Tests and Results

The objectives of the test program were to determine the effectiveness of filters in preventing engine wear and to determine directions which future filter design and testing should take. To meet these objectives, three sequences of tests were performed: wear with and without filtration, wear versus particle size, and wear versus filter media selection. While analyzing the test data, it was found that the most significant data regarding filtration was generated by three parts, the main bearing, the oil ring, and the wrist pin bearing. The rocker arm bearing, piston ring groove, and the valve showed only a minor degree of wear during all the tests. The fuel injection pump plunger showed a significant wear, but this is not associated with lube oil contamination. The compression ring showed a considerable amount of wear, but the test to test scatter prevented the drawing of conclusions from any single test. The data from the connecting rod bearing could not be used because one of the implants broke free and migrated in the bearing surface. Because of these results only the data from the main bearing, oil ring and wrist pin bearing are reported and concluded upon in this paper.

In all the tests performed, an artificial contaminant (AC Fine Test Dust or particle size fractions thereof) was used to accelerate wear rates. These contaminants are representative of road dust which is the abrasive contaminant likely to be present in the oil system. These contaminants are also used in bench evaluation of filters. This provided a link whereby the bench test results and the engine test results could be compared. In all tests, the engine was run on a dynamometer using a load/RPM cycle representative of on-highway service. Before any tests were run, the new parts were allowed to break in during a run-in period of six hours. Following this, initial readings were taken for all parts.

Wear Reduction through Filtration - the first test sequence run was to evaluate filter wear prevention effectiveness. To do this, a comparison was made between the wear generated by injecting two grams of ACFTD (0-80 micrometer) into the engine lube oil system with no filters present, and the wear generated by injecting twenty grams of ACFTD with normal filters present in the system. These amounts of contaminant were arrived at based on previous engine test results and an estimate of filter efficiency at 90%. The contaminant was added at a rate of 0.017 grams/min. for the no filter test and at 0.167 grams/min. for the filter tests. The contaminant was injected in slurry form into the oil pump inlet. Test duration was two hours with the engine run on a simulated highway load/RPM cycle for the full time period.

The wear measurements produced in these tests reveal the effectiveness of filters at preventing wear, but also raise other questions. The greatest reduction in wear occurred with the oil ring. Here the use of filtration reduced the wear rate by 96%. The wear reduction for the main bearing was 86% and for the wrist pin bearing, 57%. These wear reduction figures also account for the difference in dust add rates. The differences in wear rate reduction are indicative of the difference in both the method of lubrication and the clearance between wearing surfaces for each of these parts. The main bearing and wrist pin bearing are pressure lubricated and see a higher oil flow rate than the oil ring. This higher flow rate both subjects the parts to more particles per unit time and tends to wash particles from the clearances. Despite the filtering of particles, any particles present will probably have many opportunities to cause wear. The oil ring sees low oil flow for lubrication and washing, therefore any particles present would probably be trapped allowing them to cause more wear. With filters present, fewer particles will reach the oil ring because of reduced numbers and low flow rates. The clearance between wearing surfaces will also influence wear rates; if particles can pass between the parts, there can be little wear. This is more closely examined in the particle size sensitivity testing.

Particle Size Sensitivity - the next group of tests run were to determine if the engine parts were more sensitive in terms of wear to different contaminant particle size ranges. To do this, two gram samples of particle size fractions of ACFTD were introduced into the lube oil system, under the same conditions as in the previous study, with no filters present. The particle size ranges used were: 0-5 micrometer, 5-10 micrometer, 10-20 micrometer, 20-40 micrometer, 40-80 micrometer and 80-200 micrometer. The 80-200 micrometer dust was derived from AC Coarse Test Dust which is the same chemical composition as ACFTD but with larger particles present (up to 200 micrometer). This type of testing is frequently used in the hydraulic industry to evaluate sensitivities of pumps, motors and seals, most notably in the work of Oklahoma State University's Fluid Power Research Center (4)(5). This testing is done to evaluate system filtration needs, and it seemed most natural to extend it to another oil system, the lube system of an engine.

The test results were surprising. Rather than a uniform sensitivity to all particle sizes, or a common maximum sensitivity for all parts which might be expected, the measurements showed that the main bearing, wrist pin bearing and oil ring were all sensitive to different particle sizes. For the oil ring, its maximum sensitivity occurred with 5-10 micrometer particles, for the main bearing 20-40 micrometer, and for the wrist pin bearing 20-80 micrometer particles. This sensitivity to different particle sizes may be best explained by each of these parts having different clearances between wear surfaces. These clearances allow a particular size range to cause more damage than others, since the smaller particles pass through causing little wear and the large particles cannot enter the clearances. What is not explained by this theory is that the oil ring, which is the best protected by a filter, is the most sensitive to very small particles which would presumably pass through the filter more easily. A possible explanation is that the grinding action of both the oil pump and the oil ring itself gradually converts the larger particles to particles in the sensitive size range for the oil ring. Without the aid of filtration, the large particles become small, increasing the wear rate with time. With filtration, these large particles are removed, lowering the concentration of critical particles over the long run, resulting in lower overall wear.

Filter Media Comparison - in the automotive industry, there has been a trend toward extending service intervals for the engine. One method of extending the life for a filter (life defined to a specified pressure drop) is to use a more "open" filter medium. The more "open" medium is usually of lower efficiency in particle retention capability. For this reason an evaluation was made comparing the wear rates with a "tight" versus "open" medium.

These tests were run differently from the preceeding tests. While previous tests were run using a set time period and contaminant addition, the media comparison tests used the life of the filter as an end point. To achieve this, the test was run and contaminant added until the filter reached a differential pressure of 10 PSI, considered the end of the filters effective life in this engine application. The contaminant used was ACFTD with an add rate of 0.67 grams/min. The injection point and method was the same as for previous testing. The two media chosen had previously been run on this engine type in endurance testing and had exhibited a longer on-engine life for the open medium. These tests used no contaminant addition; only the contaminants formed by the engine itself plugged the filters. Bench testing of these filters to the appropriate specification with ACFTD could not, however, discriminate between the two media.

The first result of the testing was that the open medium took 2.7 times longer to plug than the tight medium. This agreed well with the results of on-engine performance with no contaminant added. The amount of dust added to plug the tight medium also agreed well with the bench test for this medium type. However, there was no correlation between the bench test of the open medium and its on-engine performance with or without contaminant being added.

The wear measurements taken for these tests tell a significant story. After normalizing for the different engine running times, it was determined that the wear rate for the wrist pin bearing increased by a factor of 2.5 by using the open medium in place of the tight medium. For the main bearing, the ratio was better than 3:1. The oil ring results did not differ from one medium to the other, however. This result may seem incongruous until the results of the particle size sensitivity tests are examined. The oil ring is sensitive to a particle size range where neither filter medium has much effect, therefore there is little difference in the wear rates. For the wrist pin and main bearings, their sensitivities lie in a range where the two media perform differently, consequently, there is a difference in wear rates.

These tests demonstrate the trade-off involved in extending service intervals through reduced filter performance. The more open medium does allow a great increase in on-engine life but at the expense of increased wear rates for any components whose sensitivity to particles lies in the area where the media differ.

## Conclusions

This test program was devised to investigate engine wear and the effect of oil filtration on this wear. Three main objectives were part of this investigation: wear reduction through filtration, wear versus particle size and wear versus filter media selection. All tests were performed on a diesel engine, using a radioactive wear measurement technique. To maintain a controlled test and a reasonable test time, contaminants representative of the abrasive dust encountered by the engine in service were used to accelerate the wear rates. Since the primary function of the filter is the removal of abrasive contaminant, their use is consistent with both actual engine conditions and the bench testing usually done on filters.

Several significant conclusions may be drawn based on the results of this testing.

1. The SPI-Wear method of wear measurement is an invaluable tool in the investigation of wear processes. It permits multiple locations within the engine to be measured and tested at the same time without any engine

disassembly other than the initial installations of irradiated parts. Without this method, an investigation of this type would be extremely difficult.

2. Filtration of the engine oil greatly reduces the wear experienced by the engine parts. The reduction of wear due to abrasive action is just one part of the picture, as evidenced by the different wear rate reductions for different parts. If the primary failure mode for a component is through abrasive wear, however, the proper choice of filter should prevent these failures.

3. Each component within an engine is sensitive to different particle sizes. These differences probably stem from lubrication methods and clearance differences. These differences in sensitivity complicate the design of a filter for the entire system, since the increased cost and service life penalty of a finer filter must be weighed against the cost of repairing or replacing a component sensitive to the smaller particles.

4. Selection of a filter for a system on the basis of service interval only may be a poor decision. The cost of increasing on engine filter life is an increased wear rate for many of the engine parts.

In the long run, the useful life of the engine may be reduced as a consequence of trying to extend service intervals.

Future Filter Design and Testing - the results of the testing done here, although they are the products of tests run on only one type of engine, show the directions that filter design and testing should take in the future. These directions will require a rethinking of current design and testing practices to better reflect the true objective of filtration, engine protection.

Design of Filters - because the primary function of the filter is engine protection, the wear characteristics of the engine should be the basis for the design of the filter. Past practices have largely been to mechanically design a filter which will fit a space alloted it by the engine manufacturer. Selection of the specifications for filter performance is left to the engine manufacturer. With increased demands being put on an engine for greater performance in a smaller, more fuel efficient package, the requirement for optimum filter performance will certainly increase. In such a case, the engine itself must be used to determine the proper filtration design.

Using the entire engine for wear evaluations as was done here may not be necessary. If a critical component in the engine could be identified, then only this part need be measured to evaluate filtration needs. By generating a particle size sensitivity spectrum for this critical component, the filter could be sized to meet the particle size retention efficiency required to protect this part. Further filter evaluation could then be done using a multi-pass type of testing, which uses particle size retention as a measure of the filter's efficiency. Further design development would then take the path of gaining maximum filter life in a given envelope size while maintaining this protection level.

Testing of Filters - most current bench tests for filters use life and efficiency measured in terms of volumetric or gravimetric contaminant removal as a measure of the filter's value. These procedures frequently do not reflect either actual engine filter life or whether the filter in fact protects the engine. The results of the filter life testing performed here could show a way to modify bench tests to better reflect on-engine performance. The filter life using the bench contaminant in the engine correlated well with actual engine performance using no contaminant. Yet the results did not correlate well with existing bench tests using the same contaminant. Perhaps the solution may be as simple as using the engine oil pump in the test bench, since its particle modification capability may be causing the change in filter performance observed. Further testing will be necessary to examine this possibility. The ideal bench test would be one that correlated directly with the field performance, but this is not available in current procedures.

In the case of evaluating particle retention efficiency, a procedure is available which measures this parameter. This procedure, if tied to the particle size sensitivities of an engine would allow a good evaluation of the engine protection potential of a filter.

The use of wear evaluation as a tool in the design and testing of filters will lead to the optimization of filters for engines. The end result of this technology will be to reduce filter costs, increase the protection of the engine by the filter, and consequently and most importantly, reduce the cost of operating the engine and increase reliability.

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# OXYGEN SENSOR FOR AUTOMOBILE COMBUSTION CONTROL

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"Three way" catalytic converters are effective in reducing automotive exhaust emissions of carbon monoxide, hydrocarbons, and oxides of nitrogen, but only when the exhaust chemistry is near stoichiometry. The exhaust gas oxygen sensor makes possible a closed loop fuel metering system to provide and maintain the required exhaust gas chemistry entering the three way converter. The oxygen sensor is a solid electrolyte galvanic cell which produces about one volt when it is exposed to a "rich" exhaust mixture and near zero volts when the exhaust is "lean". The sensor can signal the electronic control of the closed loop system whether the mixture is rich or lean, thereby permitting the system to control at or near stoichiometry. The system of three way converter together with closed loop fuel metering permits meeting stringent exhaust emission standards while maintaining better fuel economy than would be provided by known alternatives.

Key words: Air-fuel ratio; Closed loop fuel metering system; Exhaust emissions; Exhaust gas oxygen sensor; Exhaust gas recirculation; Feedback; Oxidation catalyst; Oxygen partial pressure; Stoichiometric; Three way catalytic converters; Zirconium dioxide.

## Introduction

Since the 1968 model year (1966 in California), government limits for exhaust emissions have been in effect for all new cars sold in the United States. Initially, the limits applied to emissions of carbon monoxide (CO) and unburned hydrocarbons (HC). A third exhaust gas constituent, oxides of nitrogen, was recognized as having an adverse impact on air quality, and it was found that the control approaches for CO and HC often caused an increase in emissions of oxides of nitrogen (NOx). Federal limits on oxides of nitrogen were applicable starting with 1973 models. The standards, limiting the maximum amounts of each pollutant to be emitted per vehicle mile, have been made progressively more stringent. The vehicle manufacturers have responded by using many emission control approaches to meet these standards. Generally, the approaches can be classed into four generations.

The first generation, used for 1968 through 1972 models, combined selective spark timing retard with lean (relative to uncontrolled cars) air-fuel ratios. Improvements in the accuracy and distribution of the fuel-air metering systems were needed and were developed during this period. Some manufacturers used an approach in which the air-fuel ratios were rich and an engine driven pump delivered air to the exhaust ports, creating an afterburning effect when the air mixed with the hot exhaust gases.

The standards for NOx spawned the second generation control systems. In the 1973 and 1974 models controlled amounts of exhaust gas were routed back to the engine intake. This exhaust gas served as an inert diluent in the combustion process, lowering peak temperatures and thereby reducing emission of oxides of nitrogen. These exhaust gas recirculation (EGR) systems were used in conjunction with the features of the first generation.

Model year 1975 marked the advent of catalytic converters and the third generation control systems. Catalytic converters, which served to promote oxidation of CO and HC, required the use of unleaded fuel. The converter based systems permitted retuning the engines to improve efficiency and driveability.

The first of the fourth generation (closed loop) control approaches appeared in some 1977 models sold in California. Three key developments led to a concept of a catalyst to deal with all three pollutants together with feedback control of air-fuel mixture. The three developments were: (1) low cost, miniaturized electronics, (2) "three-way" or oxidation/reduction catalysts, and (3) a means to sense oxygen in the exhaust.

## Closed Loop Emission Control Systems

There were significant gains in fuel economy when converter based systems came into use in 1975 because spark timing and combustion chamber design could be more nearly optimized for fuel efficiency. The 1980 standards (see Table 1) made it very difficult to maintain these gains in efficiency. The 1981 standards, especially the lowered limit for NOx, would surely have resulted in a step backwards for engine efficiency, if it had not been for technology that led to fourth generation systems.

The fourth generation systems utilize a catalyst that is known to reduce the oxides of nitrogen in a reducing atmosphere and also to oxidize the HC and CO in an oxidizing atmosphere. Both oxidation and reduction are accomplished in a catalytic converter in atmosphere that is neither net oxidizing nor reducing. This condition is achieved only when the engine's fuel metering system is operated at stoichiometry. That is, the fuel metered must be precisely the amount that will be completely oxidized by the air inducted.

TABLE 1
RECENT FEDERAL EXHAUST EMISSION
AND FUEL ECONOMY STANDARDS

| YEAR | HC (g/mile) | CO (g/mile) | NOx (g/mile) | CAFE* (miles/gal.) |
|---|---|---|---|---|
| 1978 | 1.5 | 15 | 2.0 | 18 |
| 1979 | 1.5 | 15 | 2.0 | 19 |
| 1980 | 0.41 | 7 | 2.0 | 20 |
| 1981 | 0.41 | 3.4 | 1.0 | 22 |
| 1985 | | | | 27.5 |

*Corporate Average Fuel Economy

Figure 1 illustrates the above-described behavior of oxidizing/reducing or "three-way" catalysts. Figure 2 shows how the use of such a catalyst can affect the emissions from an engine at a typical operating point. Clearly, the emissions benefits of the three-way catalytic converter are very significant, provided air-fuel ratio can be maintained near stoichiometry. An important added benefit of operating near stoichiometry is apparent from Figure 3. Fuel consumption approaches its minimum at air-fuel ratios near stoichiometry. The advantages of stoichiometric operation are many if air-fuel ratios are controlled.

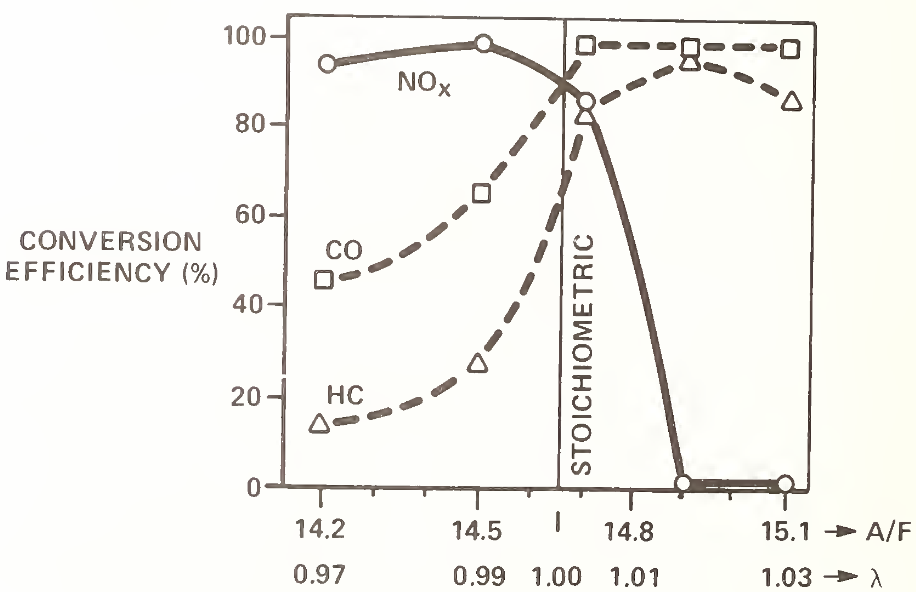


Figure 1. Three-way catalyst characteristics

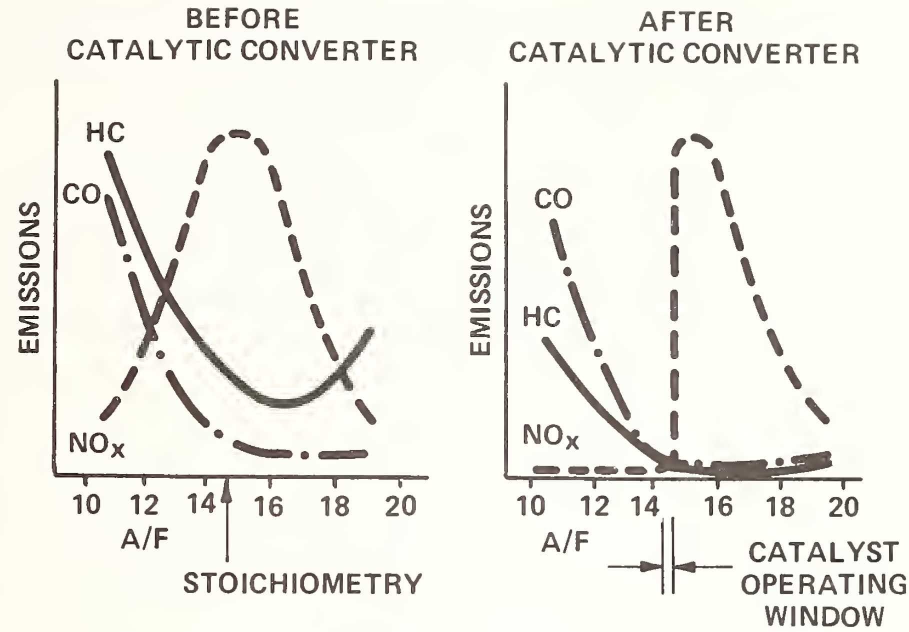


Figure 2. HC CO, and NOx emissions versus air-fuel ratio.

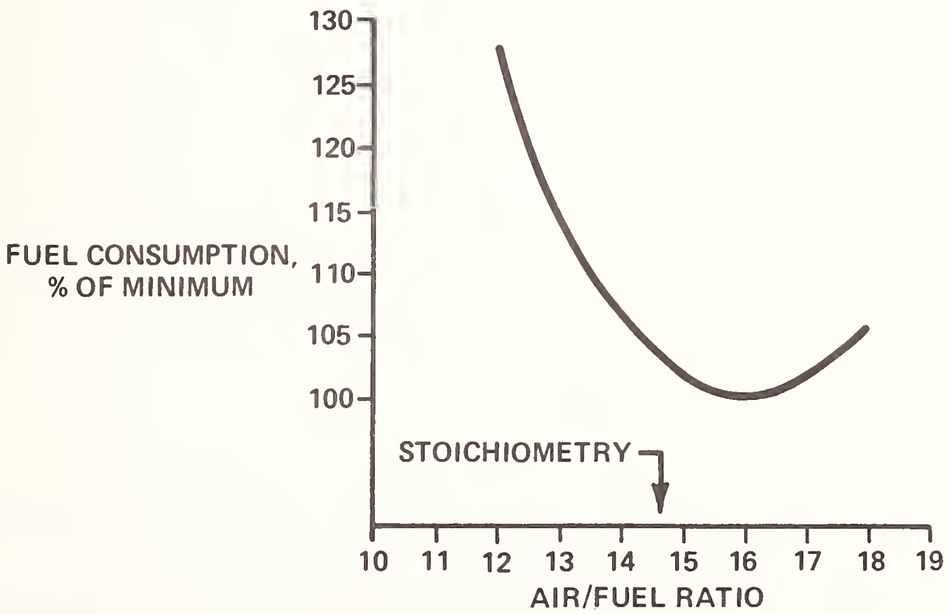


Figure 3. Fuel consumption versus air-fuel ratio

Conventional carburetors, as well as new carburetor concepts and both electronic and mechanical fuel injection, have become highly refined and are capable of accuracy far in excess of what was available in 1968. Nevertheless, engine transients, variations in ambient temperature, humidity and altitude plus wear in service make it impracticable to achieve and maintain the air-fuel ratio control demanded by the three-way catalyst. The solution to this dilemma is the closed loop or feedback fuel metering system.

Figure 4 depicts the essentials of a fuel metering system employing feedback from exhaust gas chemistry. In systems of this type, the oxygen sensor, which will be described in more detail in the following section, behaves much like a switch. Whenever the exhaust chemistry reflects an air-fuel ratio on the rich side of stoichiometry, the sensor produces an output of about 0.9 volt. Lean mixtures cause sensor output to drop to near zero.

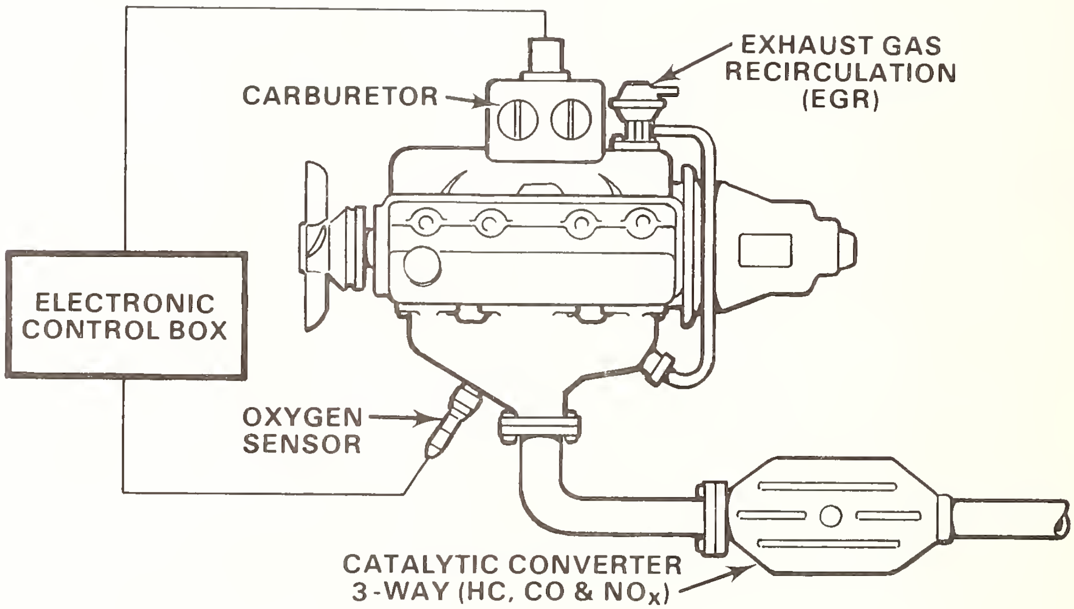


Figure 4. Feedback emission control

The output of the sensor is fed to an electronic control, which interprets sensor outputs above 0.4 volt as indicating a rich mixture and outputs below 0.4 volt as signifying that the mixture is lean from stoichiometry. The electronic control in turn applies a signal to an actuator in the fuel metering system. Actuators may take several forms, but the effect is always that a rich exhaust chemistry gives rise to a correction to the fuel metering system causing a change to a leaner air-fuel ratio and conversely.

Among actuators used with carburetors are solenoids and stepper motors. A solenoid can be incorporated in a carburetor so that when the solenoid is energized, a valve opens an air bleed passage causing the mixture to go to a predetermined lean air-fuel ratio. De-energizing the solenoid allows the mixture to go to a rich calibration. The solenoid may then be pulsed at some frequency, and the duty cycle or percentage of "On" time will determine how rich or lean the mixture will be. The mixture is thus infinitely variable between the calibrated limits. Such systems will respond to a rich signal from the sensor by changing the duty cycle at a predetermined rate toward lean. When the sensor output indicates lean, the direction of duty cycle change is reversed. This concept is somewhat analogous to a temperature control with a thermostatic switch. The sensor is similar to the thermostat, and the rate of change of duty cycle corresponds to the rate of heat input or removal.

All systems based on the switch like behavior of the oxygen sensor necessarily allow mixture to vary somewhat on both sides of stoichiometry. The requirements of the three-way catalyst are nevertheless met very effectively by maintenance of the average mixture at or very close to chemically correct. The three-way catalyst is often supplemented with a downstream oxidation catalyst. In this case air will be pumped in between the two catalyst beds during some modes of operation.

Many variations of feedback systems have been implemented. Electronic fuel injection is particularly well suited to closed loop operation since response is fast and the injectors are the actuators. A number of systems have been developed which combine closed loop fuel metering control with electronic control of other engine parameters such as spark timing, EGR, and even idle speed. In all of these systems the oxygen sensor is an essential component.

## Operation and Principle of the Oxygen Sensor

Certain oxides have the ability to conduct electrical current by means of very mobile oxygen ions. One of these oxides, zirconium dioxide, is used as a solid electrolyte in a galvanic cell to sense the oxygen concentration in automobile exhaust gas. Usually the zirconium dioxide contains a small amount of yttrium oxide in order to reduce its electrical resistance and to provide high temperature structural stability.

The galvanic cell, Figure 5, consists of the solid oxide electrolyte generally in tubular form which is coated on the inner and outer surfaces with porous platinum electrodes. When the platinum surfaces are exposed to gases with different oxygen partial pressures, oxygen ions are formed on one electrode and migrate through the oxide electrolyte from the region of higher to the region of lower oxygen concentration. This establishes a voltage potential difference across the electrolyte, the magnitude of which is defined by the Nernst equation.

Atmospheric oxygen is normally in contact with the inner electrode in an automotive oxygen sensor. Ion flow is therefore in the direction toward the outer electrode and the exhaust gas. Since the equilibrium oxygen partial pressure in exhaust gas varies with normalized air-fuel ratio as shown in Figure 6, calculations of theoretical voltage output using the Nernst equation are possible. The family of curves shown in Figure 7 result. The large abrupt change in the oxygen partial pressure in the exhaust at stoichiometry and the resultant abrupt change in voltage produced by the voltaic cell make it usable as an oxygen sensor.

## NERNST EQUATION

$$EMF = \frac{RT}{4F} \cdot LOG \frac{PO_2 \text{ (ATMOSPHERE)}}{PO_2 \text{ (EXHAUST)}}$$

$$= .0496\,T \cdot LOG \frac{PO_2 \text{ (ATMOSPHERE)}}{PO_2 \text{ (EXHAUST)}}$$

R = UNIVERSAL GAS CONSTANT

T = ABSOLUTE TEMPERATURE

F = FARADAY CONSTANT

$PO_2$ = PARTIAL PRESSURE OF OXYGEN

ASSUMES ELECTRON OR HOLE CONDUCTIVITY CAN BE NEGLECTED AS COMPARED TO THE OXYGEN ION CONDUCTIVITY

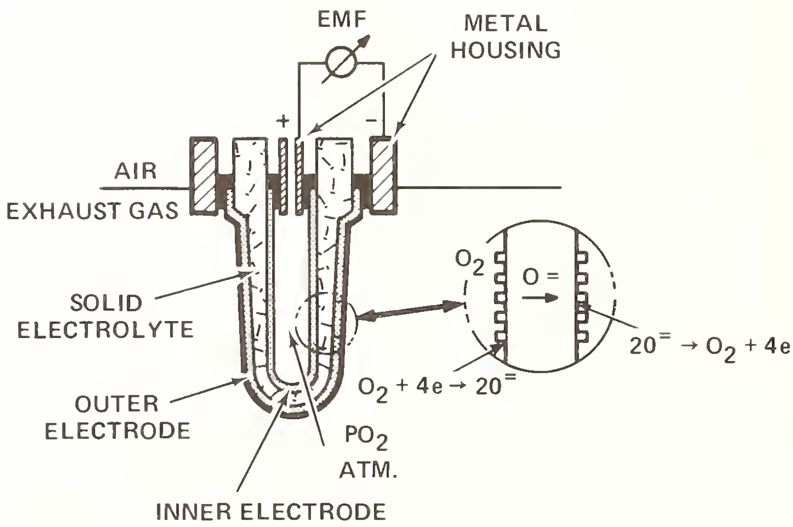


Figure 5. Oxygen sensor schematic

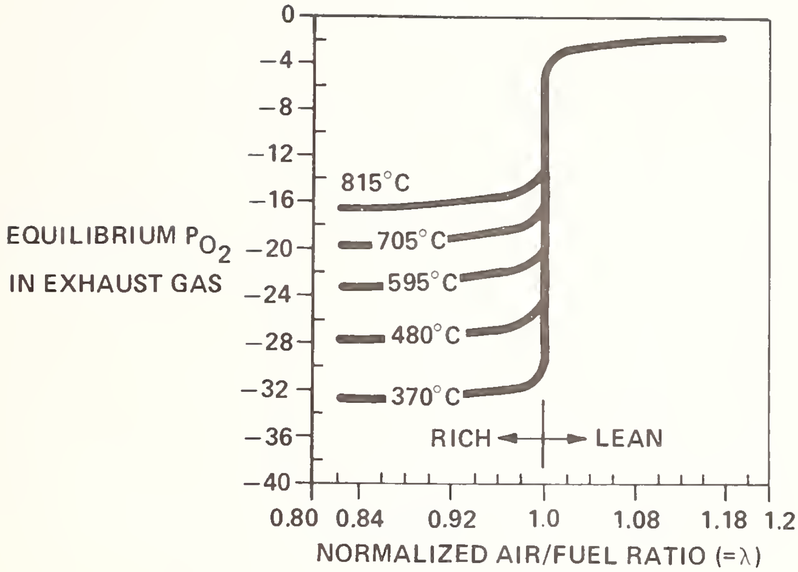


Figure 6. Equilibrium oxygen partial pressure (log of psi) versus air-fuel ratios

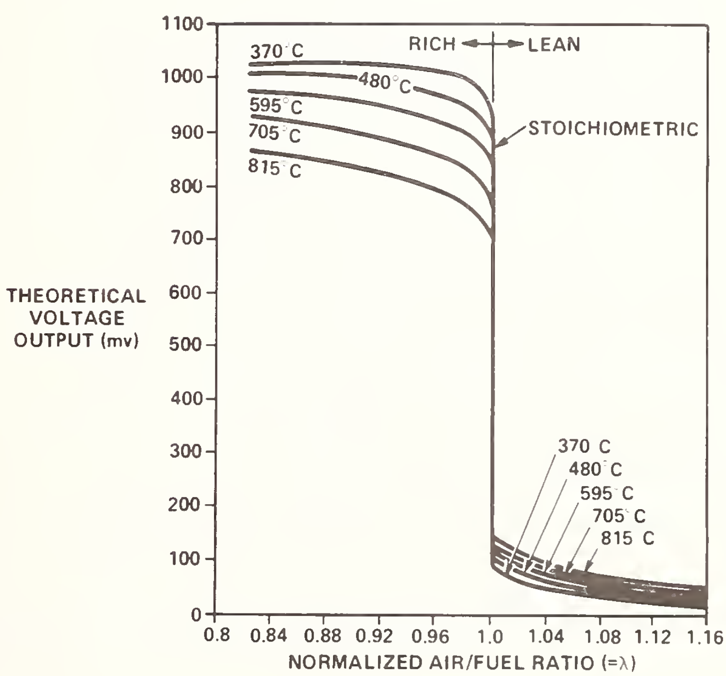


Figure 7. Theoretical voltage output versus air-fuel ratio

When the exhaust gas is rich, the oxygen partial pressure is small and the ion flow from the inner to outer electrode is large, resulting in development of approximately 0.9 volt by the cell. A lean exhaust gas with a large oxygen partial pressure inhibits the ion flow and results in development of less than 50 millivolts by the cell. The electronic control unit uses the voltage output as a signal to the carburetor or fuel injection system to decrease or increase the air-fuel ratios.

Examples of an oxygen sensor voltage trace on an oscilloscope are shown in Figure 8. The traces were obtained when the sensor electrolyte tip was heated on a natural gas burner to temperatures of 350°C and 800°C and the burner exhaust gas was switched through stoichiometry from rich to lean and then back to rich. At 350°C the 970 millivolt rich and 35 millivolt lean outputs are typical. Also typical are the 800 millivolt rich and 50 millivolt lean outputs at 800°C. Apparent on the traces are the rapid voltage changes generally occurring in less than 100 milliseconds at 350°C and less than 50 milliseconds at 800°C. These fast switch times assure the fuel control system is constantly updated as to whether the exhaust gas is rich or lean.

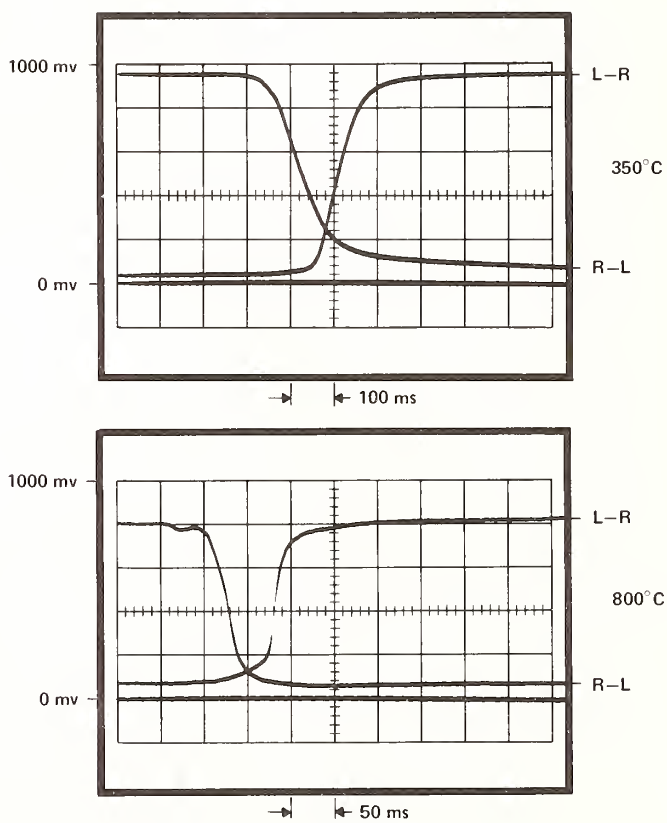


Figure 8. Transient voltage switching curves of an oxygen sensor

The design features of a functional automotive oxygen sensor are shown in Figure 9. The zirconium oxide electrolyte coated on the inside and outside with platinum is protected from the exhaust gas by a layer of magnesium aluminate spinel and an open fluted stainless steel shield. Exhaust gas passes through the flutes in a swirling motion preventing direct impingement on the electrolyte tip of the gas and solid particles contained therein. The outer electrode is connected electrically through the protective shield and a metal gasket to the electrically grounded shell. Talc powder compressed between the electrolyte and the shell provides a gas seal.

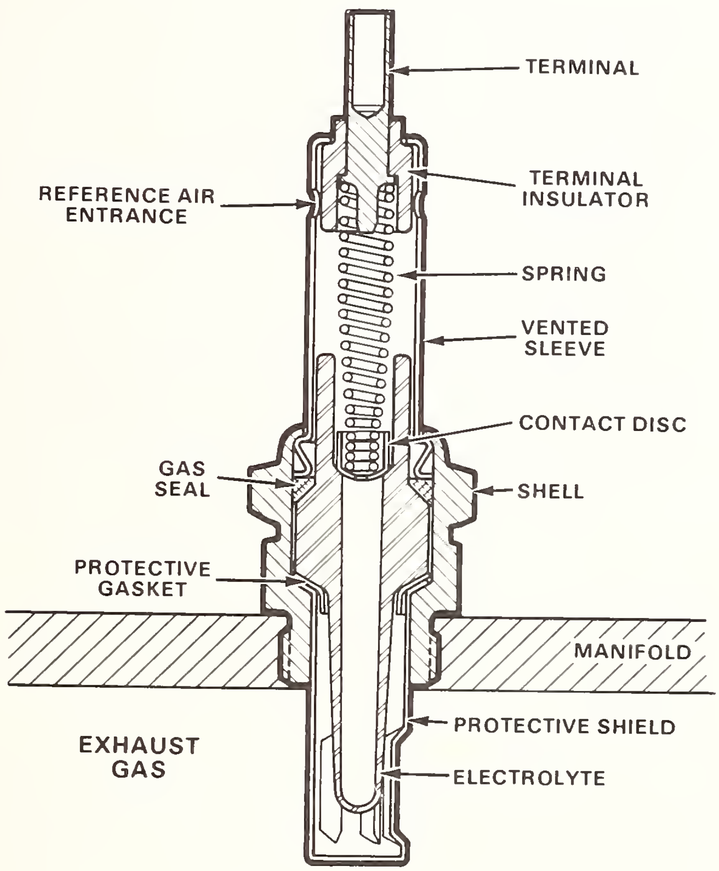


Figure 9. Oxygen sensor cross section

Electrical contact with the inner platinum electrode is through a contact disc and spring to the terminal. A ceramic terminal insulator prevents electrical shunts between the inner and outer electrodes. The vented sleeve protects the inner components from mechanical shock and contamination while allowing the entrance of air for sensor operation.

Automotive oxygen sensors of this design have proven themselves to be durable under extreme operating conditions and are certified with the Environmental Protection Agency for 50,000 miles.

## Energy and Other Implications

A feedback controlled engine, operating near stoichiometry will run very close to the best efficiency point as shown previously. Further, though EGR is still required in most applications, the amount of recirculated exhaust gas can be kept below that which has much adverse effect on efficiency. Similarly, spark timing can be maintained at or near the best efficiency setting for most operating conditions. Thus, the three-way catalyst and closed loop, oxygen sensor based approaches offer better fuel economy than currently practicable alternatives.

Another major benefit from the feedback approach is that the fuel metering system becomes, in effect, self-tuning. Also, there is little or no adverse effect from changes in ambient temperature or operation over a wide range of altitude. Many systems incorporate self diagnosis which detects malfunction and alerts the driver to have his vehicle serviced.

This fourth generation of emission control systems is seen to provide not only improved fuel economy when a car is new, but the economy will inherently be preserved over the life of the vehicle.

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# DETECTION, DIAGNOSIS AND PROGNOSIS OF GAS TURBINE ENGINE HEALTH WITH THE USE OF FIBRESCOPES

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Abstract: The use of fibrescopes in determining the overall health condition of a turbine engine has experienced tremendous growth in the past three years. The maintenance technician can now monitor the performance of each engine, and if the trend analysis signifies structural degradation of gas path hardware, he no longer need remove the engine from service for expensive disassembly to verify suspect damage. With the new developments in the use of fibrescopes, he may now insert a fibrescope into the suspect area and verify that damage does or does not exist and to what extent. Man-hour expenditure ratio for the two methods averages about 160:1, and fuel savings per incident is normally 1000 gallons. This paper describes the growth of fibrescopes over the past five years and their practically unlimited application in transportation, marine, energy production and agricultural equipment in the future.

Key words: Fibrescope; health condition; trend analysis; man-hour expenditure ratio; unlimited application.

## Background

Disassembly to inspect gas turbine engines has been one of the biggest drivers in aircraft maintenance support cost. At this time a majority of commercial engines and some military applications have converted from the fixed time between overhaul (TBO) to the on-condition maintenance (OCM) concept. But many critical parts, buried within the engine, still require major teardown at certain prescribed maintenance intervals. This teardown ensures these "buried" parts are capable of making the next inspection interval. Normally, a pre-maintenance operational check and a post-maintenance operational test run of the engine is performed which consumes an average of 3000 gallons of jet fuel for each maintenance inspection. During operational use of these engines, FOD, DOD and general parts deterioration occur on these "impossible" to inspect parts. Operational incidents, which could have been detected if internal inspection techniques were available, are extremely costly on a multi-engine aircraft where an inflight abort is declared and tons of fuel are dumped to achieve a safe landing, but to the single engine aircraft operation the result could be disastrous.

Having had first-hand experience with both fixed-cycle and on-condition maintenance for twelve years as a U.S. Air Force Propulsion Branch Chief, the author supports the OCM concept. Personal interviews with experienced logisticians within Pratt & Whitney Aircraft indicate a similar attitude in support of the OCM concept. Planning, scheduling and committing resources to maintain combat-ready aircraft around the world can be a logistics management nightmare if the manager elects to employ the fixed-cycle concept. Sudden changes in engine performance require immediate grounding of the aircraft and troubleshooting. Granted that sudden failures, in some cases, cannot be predicted, the majority of those sudden failures resulted from subtle performance changes that could have been detected, diagnosed, and corrected had a diagnostics trend analysis system, coupled with newly developed nondestructive inspection equipment, been used. Spares provisioning, transportation and manpower commitments can be predicted by the manager when he knows the exact health condition of his operational assets regardless of global location.

### Detection

One of the unique features of turbine engines is that their performance parameters, once accurately established, vary only slightly from their initial values. The parameters that are indicative of the gas path performance of a turbine engine include engine pressure ratio (EPR), compressor speed (N1 and N2 rpm), fuel flow (Wf), and exhaust gas temperature (EGT). There are numerous other terms used for these parameters, depending on the type engine and manufacturer, but basically, these four or five signals will provide the input information necessary for most engine condition monitoring programs. Normally, these parameters will not vary on a specific turbine engine unless forced by an internal or external change. If one value varies, variance in others can be predicted, and provided that the associated instrumentation system or electronic engine monitoring system is not in error, this feature permits immediate detection of the shift in performance. By plotting this information on a flight-to-flight basis (Figure 1), subtle changes in the tracked performance parameters can be detected over a period of time. This shift in baseline performance gives the maintenance technician the opportunity to remove the aircraft from the flying schedule before the failure occurs and perform the necessary inspection and troubleshooting to pinpoint the deteriorated component. Figures 2 and 3 identify major shifts in exhaust gas temperature and speed that is indicative of a damaged turbine. It is significant to note that in the majority of cases where potentially serious damage was evident upon inspection, the engine was still performing within the prescribed operational limits and would have failed before being pulled for performance loss.

Using a manual engine condition monitoring system to collect inflight engine data, the Strategic Air Command of the U.S. Air Force has accrued an estimated annual cost avoidance of $20 million in reduced hardware cost for the past 5 years.

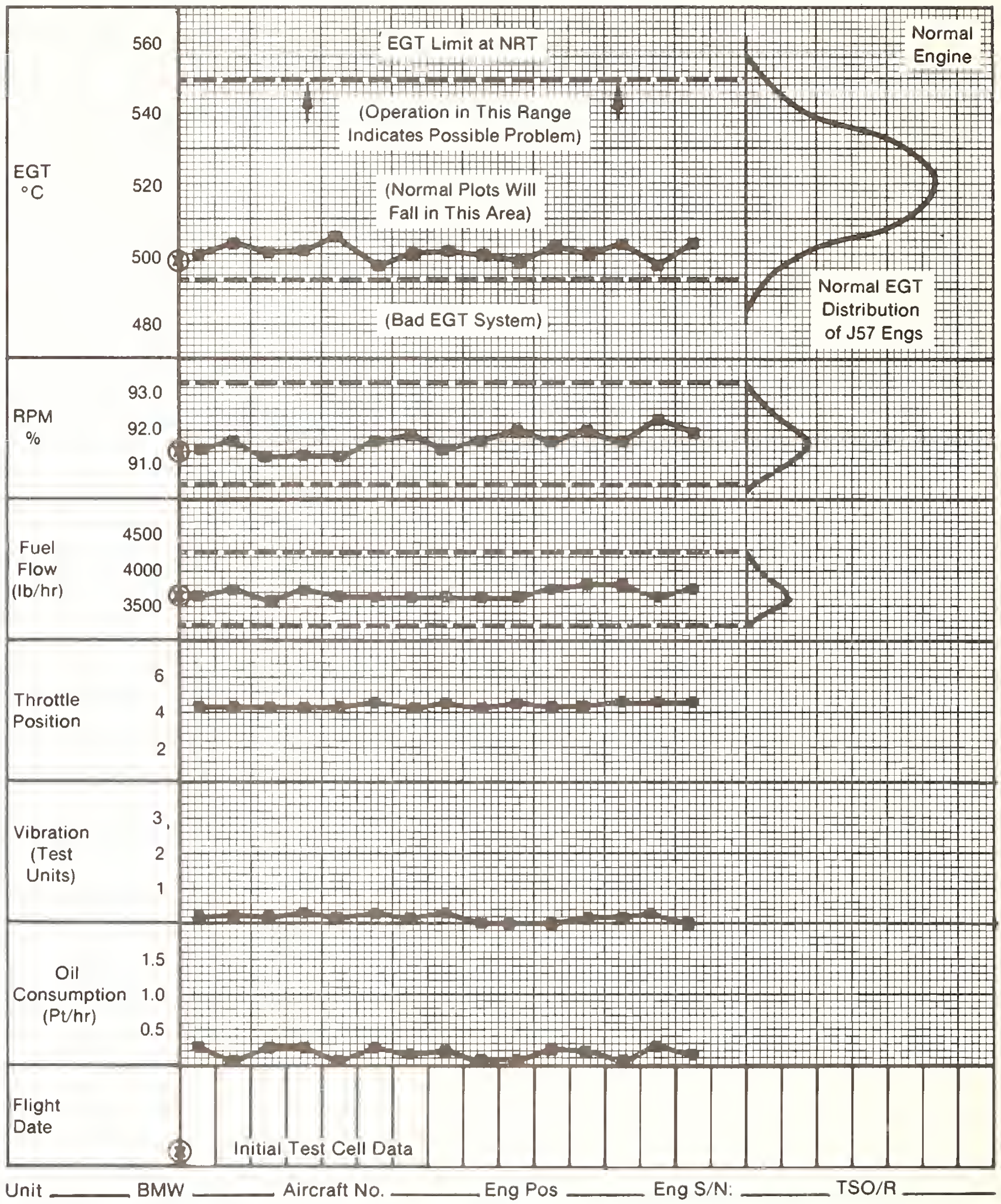


*Figure 1. Sample Engine Performance Evaluation Worksheet*

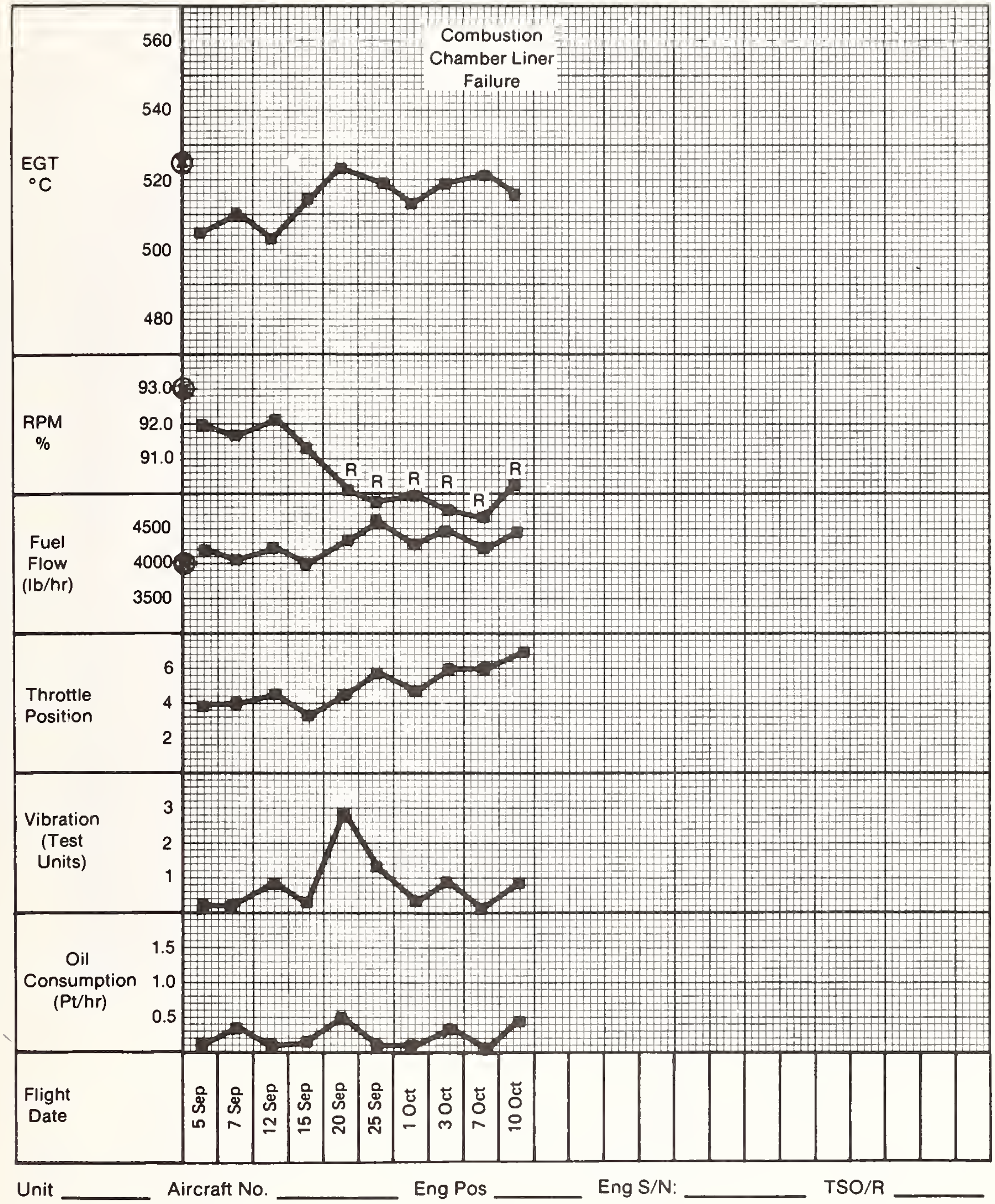


*Figure 2. Sample Engine Performance Evaluation Worksheet*

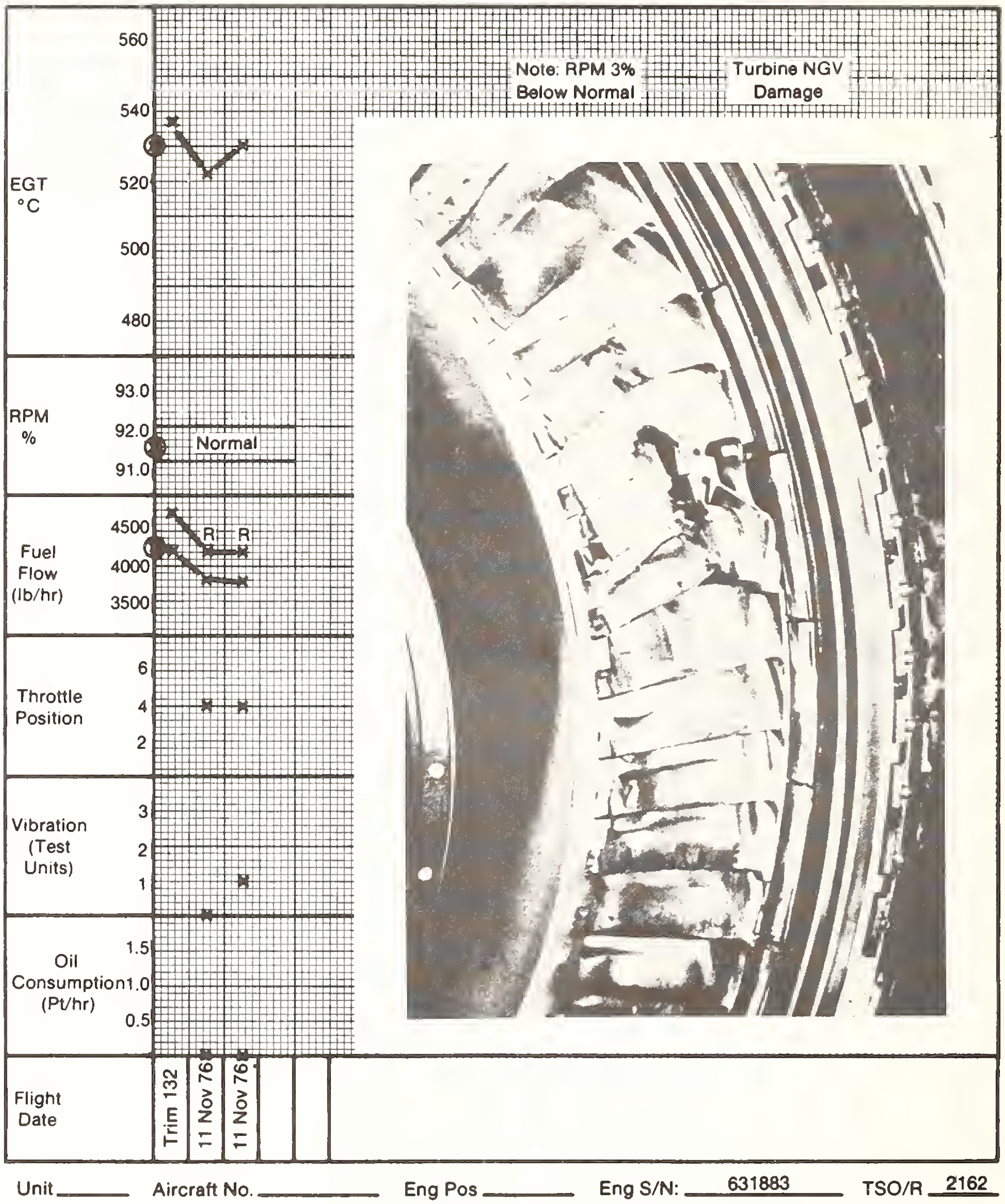


*Figure 3. Sample Engine Performance Evaluation Worksheet*

### Current Applications

In conjunction with the trending program described in the previous paragraph, internal engine inspection to detect deteriorating hardware is required to avoid inflight shutdown and excessive secondary damage. Visual inspection is required since the trending plots do not respond to these deteriorating components. The fibrescope (see Figure 4) is most widely used on the F100 engine which powers the F-15 and F-16 fighter aircraft. In addition to a regularly scheduled inspection to observe turbine health (see Figures 5 and 6), the fibrescope is used to inspect many other engine components, thereby reducing the need for costly engine removal and disassembly. In the past two years, the U.S. Air Force has realized an estimated cost avoidance of $50 million in reduced secondary hardware damage as a result of these inspections. In addition, 200 F100 engines have been removed from the aircraft prior to a major component failure.

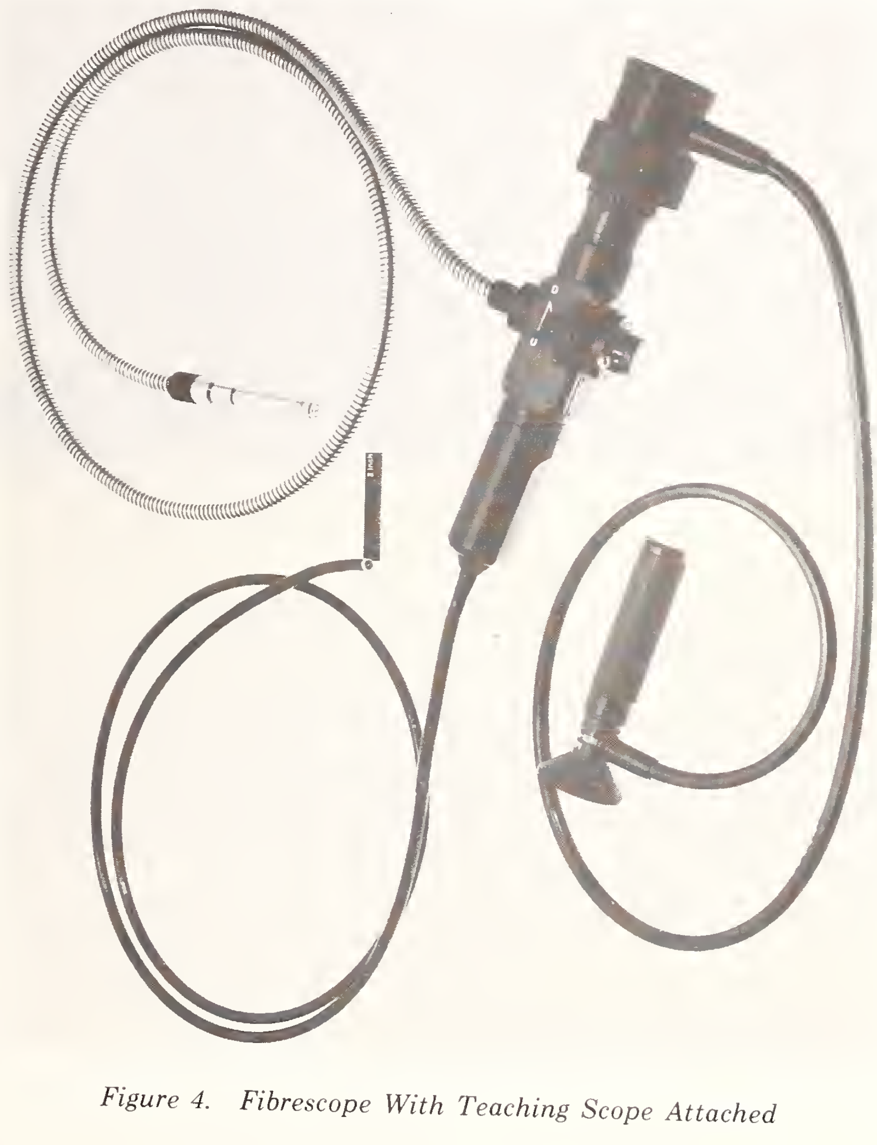

*Figure 4. Fibrescope With Teaching Scope Attached*

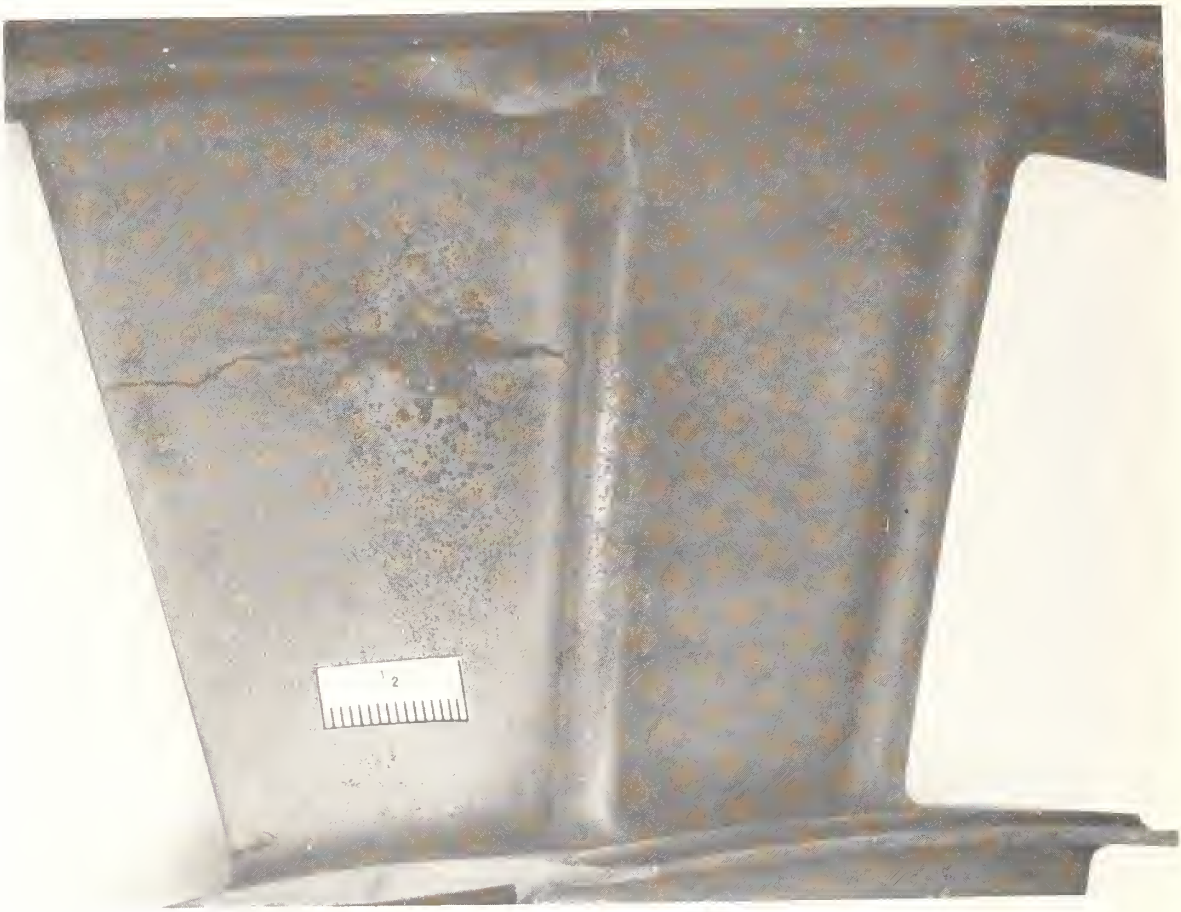

*Figure 5. Actual Turbine Vane Damage*

Other applications include combustion and turbine section inspection on the B-52 bomber family, KC-135 tanker, C-141 cargo and F-111 fighter aircraft. On the commercial side, the method used to inspect the F100 turbine was successfully adapted to the JT9D engine on the 747 aircraft and the PT6 engine installed in so many general aviation aircraft. With each new idea, another application follows, and at the present time the growth potential is immeasurable.

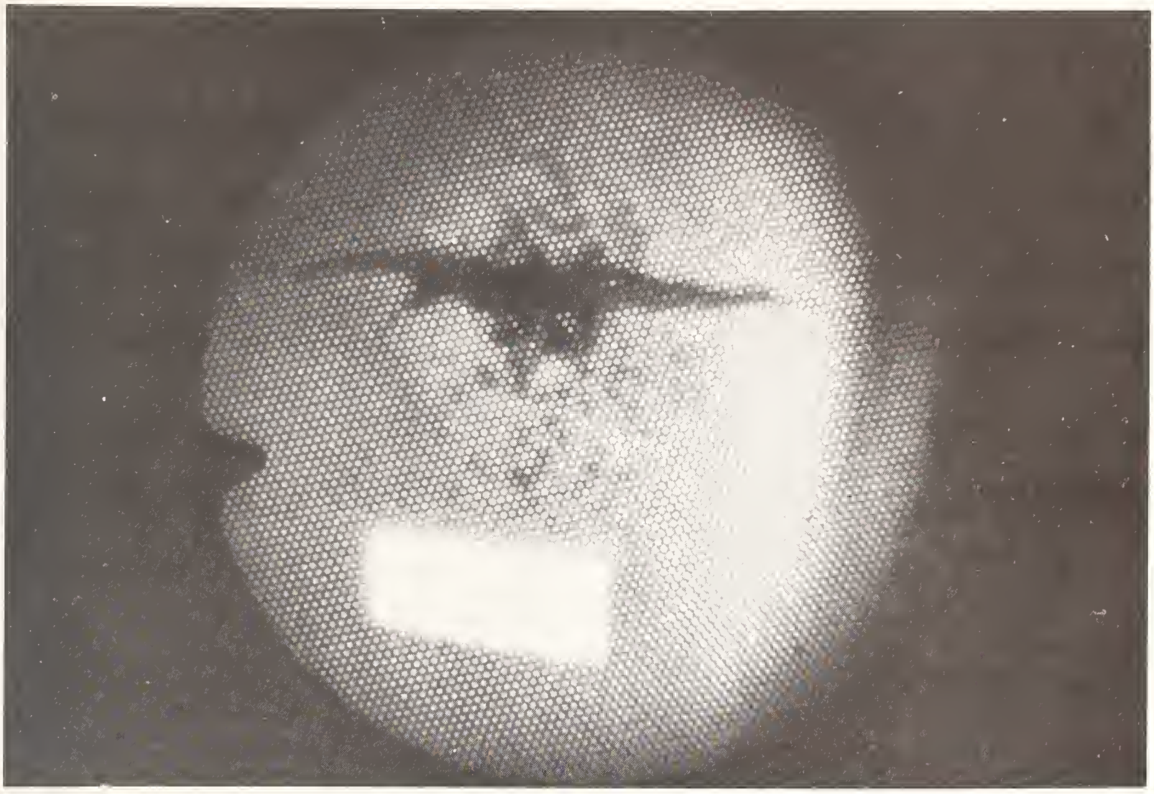

*Figure 6. Actual Turbine Vane Damage As Seen Through Fibrescope*

## Conclusion

As detection and diagnostic techniques become more advanced in the propulsion world, inflight shutdowns should decrease proportionately — if the maintenance manager grows with the technology and responds to the warning signals issued.

The challenge now must be that if this new equipment works so well on inspecting the internal components of turbine engines, what action is being taken to inspect engine attaching mounts; those inaccessible areas in the aircraft where corrosion and cracks occur; or in the fuel cells where holes are so difficult to find. In addition to aviation maintenance, an unlimited potential application exists for inspecting the valve seats, cylinder walls and pistons of internal combustion engines; inspecting weld joints inside of pipes; checking for corrosion and/or obstructions inside of fluid or gas bearing tubing, etc.

We, as engineers and managers, in one of the largest energy consuming industries, must concentrate on reducing the number of ground operational engine runs for troubleshooting and test. We must never sacrifice safety, but we must develop and use new technology and equipment to do the job without consuming fuel.

# SESSION III

## BETTER AVAILABILITY

Chairmen: R. A. Coulombe, Naval Ship
Engineering Center
J. A. George, Parks College

# INTEGRATED ON-BOARD DETECTION DIAGNOSTIC AND PROGNOSTIC SYSTEM FOR MILITARY APPLICATION

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Abstract: Advanced electronic techniques are being applied by the US Army to integrate on-board detection diagnosis and prognosis as an integral part of a total combat vehicle subsystem integration effort. The electronic system manages, controls and distributes electrical power in combat vehicles and provides the system integration functions of communications, data processing and integrated controls and displays to the subsystems. The subsystems include engine, electrical, fire control, stabilization, intercoms and safety. The system enhances the RAM-D of the combat vehicle by the inclusion of on-board diagnostics/prognostics, built-in-test and selected redundancy features. Electrical system design and modification flexibility is improved.

The system integration techniques are described which include a shared transmission medium called multiplex data bus, microcomputer control and solid state power control. The core element hardware and software design are discussed along with installation procedure for a baseline Army combat vehicle. The interfacing of off-board test equipment with an on-board multiplex bus system is discussed.

Key words: Electronic components; multiplexing techniques; on-board diagnostics and prognostics; electrical power management; control and distribution; total tank subsystem integration; MIL-STD-1553B multiplex data bus; microcomputers; bus controllers; remote terminals; crew station terminals.

## INTRODUCTION

The advanced technology systems approach for integrating all electric power and information transfer within combat vehicles includes integrated on-board detection diagnosis and prognosis systems as a design goal. The program is being performed under the title of <u>Advanced Techniques for Electrical Power Management, Control and Distribution Systems (ATEPS)</u>.

ATEPS is a technology thrust to introduce advanced electronic techniques to meet the increasing combat vehicle power requirements impacts which have increased electrical system complexity and reduced the reliability, availability, maintainability and durability (RAM-D) and survivability aspects. Concurrently, this trend has increased the information flow in combat vehicles.

Five basic technical objectives were established for the ATEPS program: reduction in vehicle wiring complexity; improved reliability, maintainability and survivability; reduction or elimination of combat vehicle depot rebuild for the electrical/electronic system; improved crew effectiveness; and improved electrical system design and modification flexibility. The ATEPS system requirements, analysis and conceptual design of a tank-automotive vehicle information system was accomplished in April 1977 by Systems Consultants, Inc. An ATEPS breadboard hardware system was designed and developed in FY78 by the Chrysler Huntsville Electronics Division.

The brassboard hardware system was demonstrated sufficiently to warrant advancement into an accelerated ATEPS prototype hardware development program for vehicle installation. The vehicle selected for the ATEPS prototype hardware program was a baseline XM1 tank.

## SYSTEM DESCRIPTION

Figure 1 illustrates the total combat system integration potential of the ATEPS concept. This system integration is achieved by the use of digital multiplexing with microcomputer control, integrated controls and displays and solid state power control/switching. Communication among the combat vehicle subsystems is accomplished by means of a shared transmission medium called a Multiplex Data Bus (MDB). Other key elements of the system are Bus Controllers (BC), Remote Terminals (RT) and Crew Station Terminals (CST).

Figure 2 is an example showing the multiplex bus architecture. The interconnection of the BC, RT and CST via the MDB is made through a slip ring which interfaces the hull and the turret data/power systems. All data are transferred between the bus controller and various remote and crew station terminals via the data bus section. The subsystems or Line-Replaceable Units (LRU) are interfaced with the ATEPS system through RTs which are placed in zones of high wiring/component density.

Figure 3 shows how a typical ATEPS system might be installed in a tank. The MDB is installed in a "ring" configuration and operates bidirectionally. If the MDB system is severed in one location, it continues to operate in a normal manner by continuing the flow from both sides to the point of rupture rather than in a continuous loop fashion. The MDB and power leads are combined to form a hybrid cable assembly.

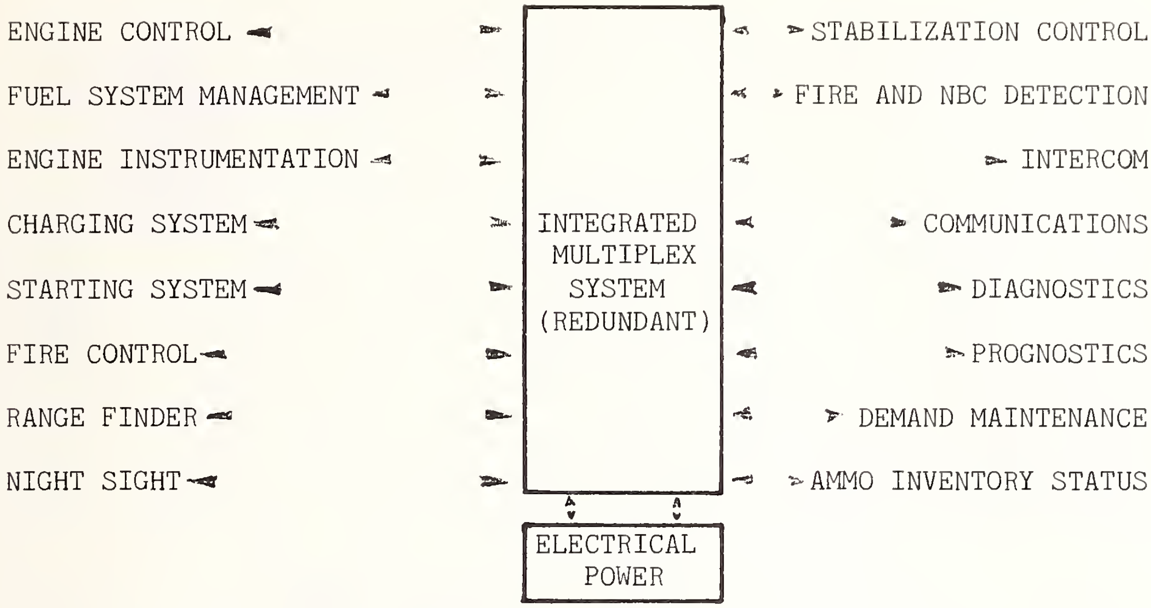



FIGURE 1. TOTAL TANK SUBSYSTEM INTEGRATION POTENTIAL

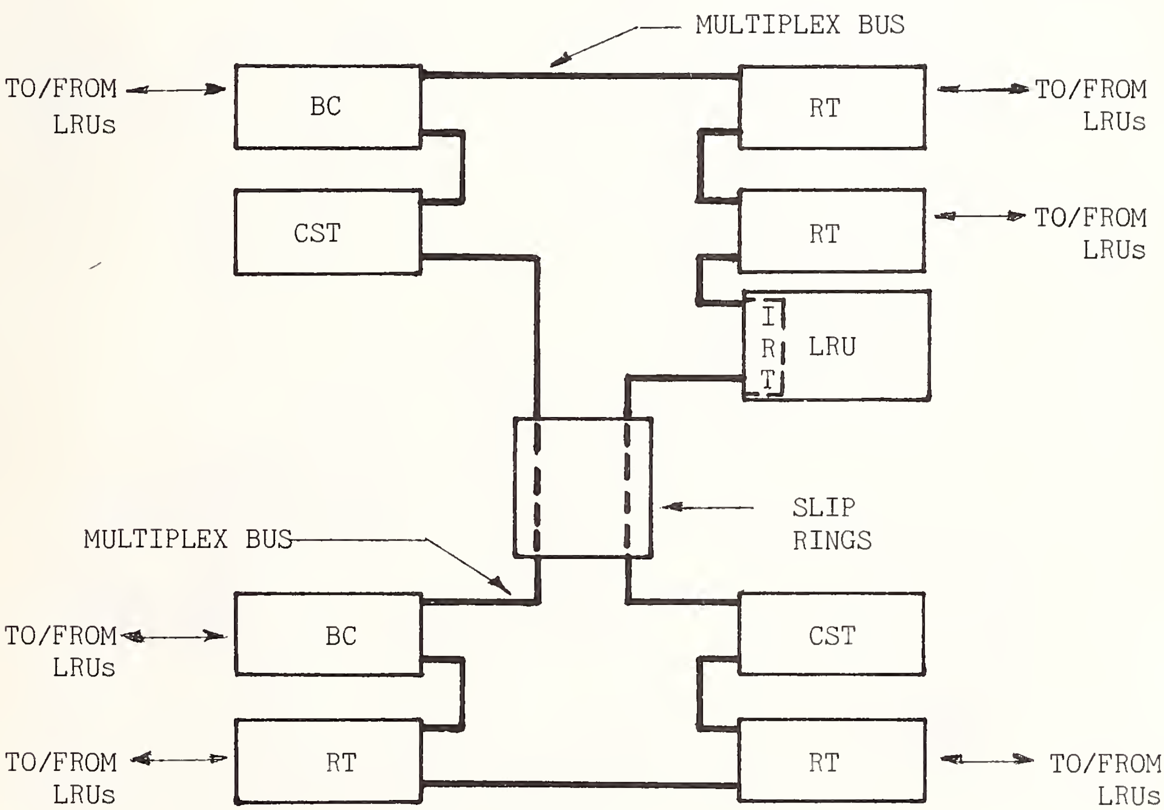
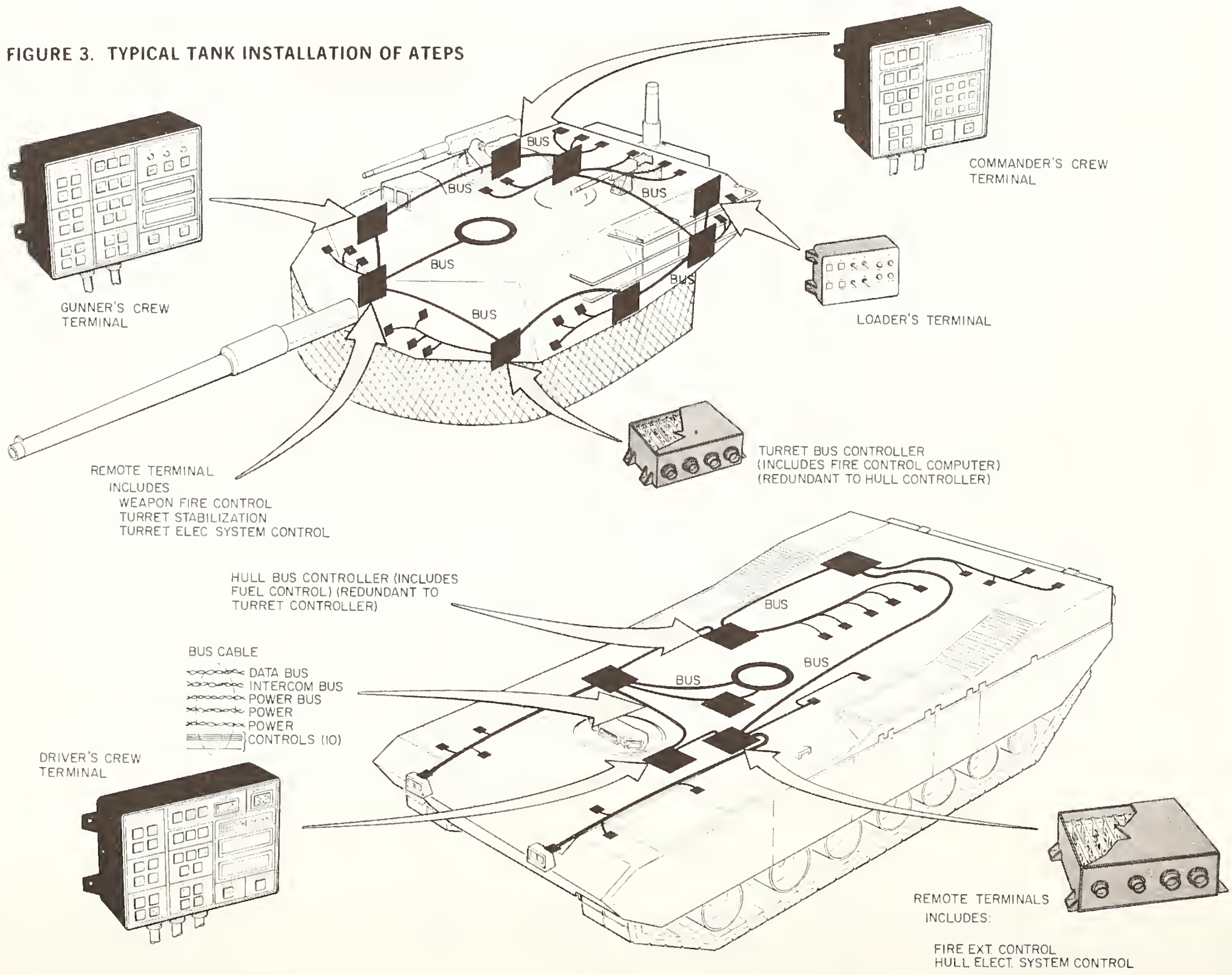



FIGURE 2. TYPICAL MULTIPLEX BUS ARCHITECTURE

**FIGURE 3. TYPICAL TANK INSTALLATION OF ATEPS**



The MDB was designed in accordance with the requirements of the triservice standard, MIL-STD-1553B. Some of the bus features are listed below:

| | |
|---|---|
| Bus Operation | - Asynchronous<br>- Command/Response<br>- Half Duplex |
| Word Types | - Command<br>- Status<br>- Data |
| Modulation | - Serial Digital Pulse Code Modulation |
| Data Code | - Manchester II Bi-Phase Level<br>- Bi-Polar |
| Transmission Bit Rate | - 1.0 Megabit per Second |
| Word Size | - 20 Bits<br>- 16 Bits plus Syn and Parity Bits |

The bus controller controls the transfer of data over the MBD. It consists of a central processor, bus interface logic and memory. The central processor is a programable microcomputer which executes stored programs such as operating programs, maintenance programs and self-test/diagnostic routines. The prognostic routines could be added in future ATEPS systems. The BC should be capable of performing all the functions performed by a RT, if required. The overall BC functions are summarized below:

- Maintains Bus Control
- Performs all Computations
- Performs all Systems Logic Decisions
- Fault Diagnostics and Malfunction Detection
- Displays Formatting and Processing
- Provides Status Information
- Verifies Data Transfer to/from Terminals
  and ...
- Contains Systems Operational Programs
- Contains System Test Programs
- Contains Crew Advisory Service (Checkout Lists, etc.)

The remote terminals contain the electronics necessary to interface the bus cable with the LRUs. It converts signals for transmission to the BC and also decodes data from the BC. The solid state switches required to connect power to the components are also located in the remote terminals. The output of the generating unit is distributed directly to the RTs. This power is distributed as required by short cable lengths to the LRUs. The overall RT functions are as follows:

- Conditions and Converts Input Data
- Multiplexes Input Data for Transfer to Data Bus
- Demultiplexes Output Data from Data Bus
- Drives and Protects Output Signals
- Recycles all Outputs Back to Data Bus for Verification
- Verifies Data Transmission to/from Bus Controller
- Performs Self-Test and Maintains Terminal Status
- Processes Data Utilizing Remote Microprocessor

The crew station terminals are the interface between the ATEPS system and the crew. The following CST functions have been identified:

- Accepts Inputs from Crew Members (Illuminated Push Button Switches)
  Display Control
  Subsystem Control
  Status Inputs

- Audio Input/Output
  Converts Input to Digital
  Converts Output to Audio

- Displays Discrete Status Lamps
  Critical Status Information
  Switch Functional Status

- Presents Status and Warning Information on Alpha-numeric Display (120 Characters, 6 Lines with 20 Characters Each)
  All Subsystem Status
  Maintenance Status
  Test and Checkout Instructions
  Subsystem Failure Indication
  Crew Safety Warning
  Displayed Data Available to all Terminals

The design utilizes an interactive or paging technique controlled by the bus controller. Each crew member is provided with a crew station terminal. However, in current four-man crew applications, the loader only has a panel interfaced with the multiplex data bus via a remote terminal. Crew station terminals are designed to be internally identical, but have unique custom front panels tailored to the specified requirement of its operator. The integration of controls and displays in the crew station terminals reduces the number of line replaceable unit components in the conventional system. For example, the driver's CST replaces driver's instrument panel, driver's master panel, master warning panel and intercom box.

BUILT-IN TEST AND DIAGNOSTICS

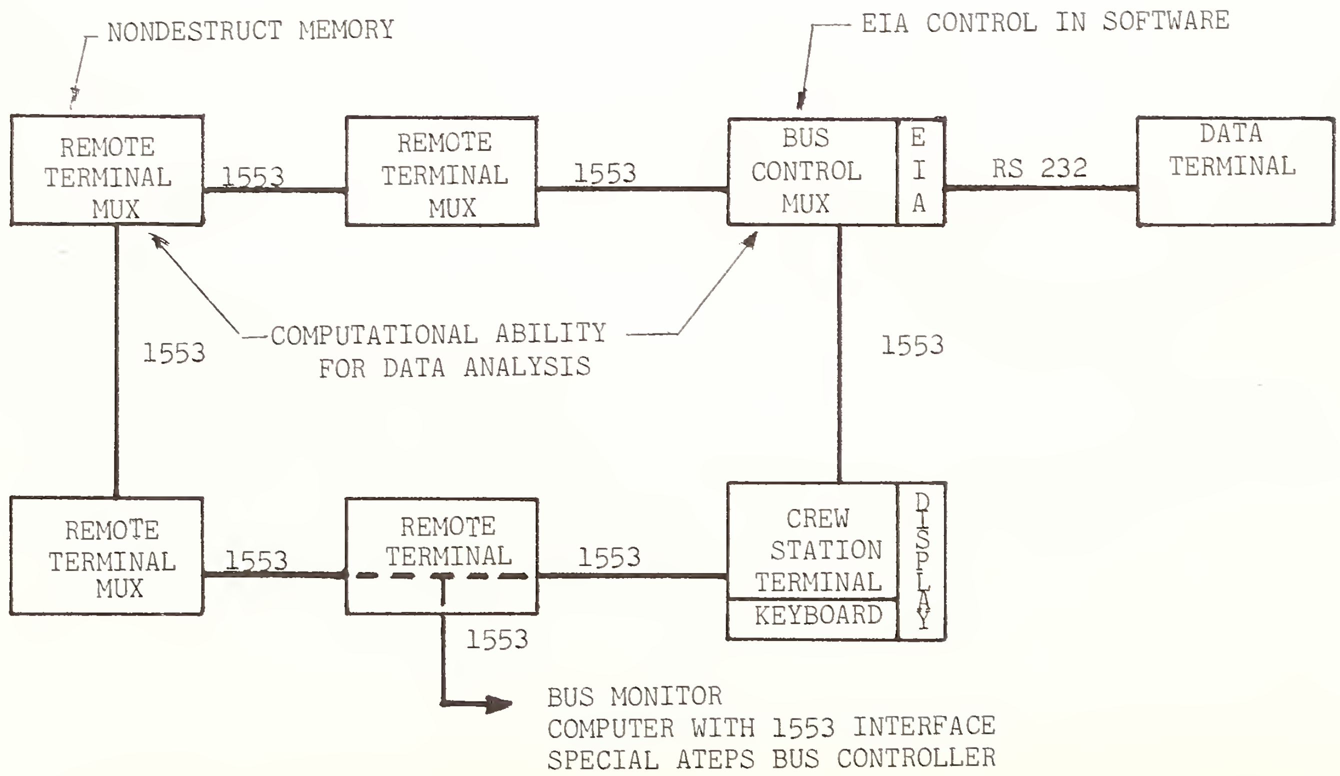
ATEPS will contain all the measurement and stimulus capability to perform self-diagnosis and fault isolation to the line replaceable unit level on all the vehicle subsystems and to the printed circuit board level on some of the subsystems. The effectiveness of the self-diagnostics and fault isolation capabilities of ATEPS is dependent upon the depth of incorporation of the multiplex system into the vehicle subsystems.

The response of the crew station terminal when interrogated by the bus controller is checked for validity. This feature is used to verify operation or isolate faults in the MDB cable assembly or the bus control circuits in the terminals. The terminals contain built-in-test features which allow measurement of inputs and outputs which are transmitted to the BC microcomputer for verification or diagnosis. The CST functions are tested by special diagnostic routines using the CST display to guide the repairman.

The ATEPS concept lends itself extremely well to on-board diagnostics. The major components, both hardware and software, are already in the system except for addition of some sensors. Inputs from the sensors would be transmitted through the remote terminals, via the multiplex data bus to the bus controller where they would be analyzed. Sensor's output data would be checked as a result of either routine operation commands from the BC or a symptom-oriented test selection from a CST. After the BC performs all the necessary computations to determine whether a malfunction has occurred, the resultant decision would be sent to the appropriate CST for display and corrective action.

Diagnostics rationale, software and algorithms for most of the systems have already been developed or will be developed prior to fielding of ATEPS. The software and algorithms for diagnosing of spark and compression engines and associated accessories were developed under the Simplified Test Equipment for Internal Combustion Engines (STE/ICE) program by RCA, Burlington, MA for TARADCOM. The STE/ICE system has been fielded and has proven to be extremely successful. Diagnosis capabilities for the AGT-1500 turbine engine and the electrical, stabilization, fire control and fire suppression systems in the turret were developed under the Simplified Test Equipment-Tracked Vehicle (STE-T) by RCA, Burlington, MA for TARADCOM. STE-T is currently supporting the XM1 tank during OT/DT III. Diagnostics for turret hydraulics, communication systems and on-board missile systems will be developed under the Simplified Test Equipment-Expandable (STE-X) program and will be available in time to incorporate into ATEPS before it is fielded.

FIGURE 4. ATEPS PROGNOSTIC SYSTEM BLOCK DIAGRAM




PROTOCOL ALLOWS ANY TERMINAL ACCESS TO BUS
MUX-CONDITIONS AND MEASURES ALL DATA FROM 20mv-40v <60CPS

PROGNOSTICS

The ATEPS concept provides for the incorporation of prognostics hardware in accordance with the block diagram of Figure 4. The principal features are:

- EIA RS 232 Communication Port with Software
- MIL-STD-1553B Port for External Interface
- Extensive Data Logging and Analysis Capability
- Built-In Failure Analysis-Prediction-Detection-Correction
- Memory and Computing Capability for Trend Analysis
- Protocol to Interface External Computer, Bus Tester, Special Crew Terminal for Data Acquisition

ATEPS is designed to access all the data base for any applicable vehicle. Low level multiplexing is available to condition, sample, convert and filter a wide range of signals between 20 millivolts and 40 volts with a response as great as 60 cps. The standard interface for the digital and multiplex systems requires a minimum of circuitry to provide special excitation and conditioning for special sensors that may be required.

Nondestruct memory is provided to store indefinitely any measurements or resultant calculations to provide a historical data base for any trend analysis that may be requested. There is sufficient computing capability available to do all but the most sophisticated algorithms required for data analysis.

Any of these results may be processed for presentation either directly to a crew station terminal, as accessed by bus monitor equipment, or using an EIA standard interface and to communicate with any data terminal.

The interface of prognostic monitoring with the ATEPS system will initially be limited to engine functions. The establishment of a prognostic test capability on combat vehicles will require definitive cost effectiveness investigation and assessment of its contribution to improved readiness. Some of the combat vehicle systems to be considered are as follows:

- Transmission and Final Drive (Transmission, Tracks, Road Wheels, Sprockets (Drive Wheels), Brakes)
- Power Plant
- Gun Tube and Breech
- Recoil Mount
- NBC
- Fire Control and Electronic Suites
- Autoloader

The TACOM prognostics program is developing around the hardware assets of the Vehicle Monitoring System (VMS) project which was started in 1976 as a joint TARADCOM/DARPA project. Analysis of test data generated by this project has indicated considerable prognostic potential.

## ATEPS BREADBOARD HARDWARE SYSTEM

The ATEPS breadboard hardware system provided capabilities for the demonstration and evaluation of the multiplexing concept with integrated controls and displays for the hull electrical system, turret electrical systems and engine monitor circuits of a combat vehicle. The following items were demonstrated:

- Displays
- Engine (Sensors, Start-Up, Stop)
- Lighting (Service, IR, Combat Mode, Dome Lights)
- Stabilization (Power, Zero Balance, Moving/Stationary)
- Ballistics (Gun and Ammo Selection, Manual Entry, Firing/Safety)
- Crew Safety (Fire Detection, NBC Detection, Smoke Grenades)
- Hardware Failures (Bilge Pump, Driver's Dome Light, Removal of Terminal)

The concept for the Bus Controller was purely a breadboard function to allow the use of an off-the-shelf microcomputer for the Central Processing Unit functions. The unit did perform the bus interface logic for the data bus. The computer prototyping system (990/4) sent messages to the Bus Controller rather than to the terminals directly. The hybrid Multiplex Bus Cable assembly ran in a continuous loop between the six terminals and the Bus Controller. This arrangement provided increased reliability and a means to demonstrate that a single break in the cable assembly would not disconnect any terminals from the Bus Controller and that the entire system would still function.

## ATEPS PROTOTYPE HARDWARE SYSTEM

The ATEPS Prototype Hardware system is being developed using the XM1 tank as a baseline for design, installation and test. The systems concept for the XM1 baseline tank implementation includes:

| | |
|---|---|
| Dual Bus Controllers | - Hull Controller<br>- Turret Controller |
| Bus Control Computer | - Imbedded SBP 9900 Microcomputer Within Each Bus Controller |
| Crew Terminals | - Vacuum Fluorescent Driver Gauges<br>- Vacuum Fluorescent Alpha-Numeric Displays<br>- New LED Illuminated Control Switches |
| Intercom | - Separate Digital Data Bus |

A hull LRU comparison between the ATEPS hardware and the XM1 hardware that would be removed is as follows:

| LRUs Added By ATEPS | LRUs Removed By ATEPS |
|---|---|
| Driver's Crew Terminal | Driver's Instrument Panel |
| Right Front Remote Terminal | Driver's Master Panel |
| Left Front Remote Terminal | Intercom Box |
| Power Pack Remote Terminal | Hull Networks Box |
| Rear Remote Terminal | Engine Control Unit |
| | Fire Extinguisher Amplifier |

The intercom box functions have been integrated into the crew terminals. The engine control unit function of the fuel management system has been incorporated into the bus controller function. The reduction in wiring has been initially estimated at 40%. The fire extinguisher system receives power from ATEPS, however, the fire sensor signals are independent of the multiplex data bus. The fire extinguisher second shot is bus controlled.

The turret prototype hardware system development includes a study and analysis to define the system for design. The system will then be designed to the number of terminals and functions of each terminal. The preliminary work indicates there would be four remote terminals and two crew terminals. The design goal is to perform all electrical functions of the turret, the fire control functions and the stabilization requirements.

The gun/turret drive unit, line-of-sight unit, computer electronics unit, commander's weapon station amplifier, gunner's control panel, turret networks box and commander's panel will probably be replaced. The gunner's primary sight may also be changed as the system is defined.

## BENEFITS

The ATEPS system design provides significant benefits in the areas of improved reliability, maintainability and flexibility. Some of the improvements in each area are listed:

### Reliability

- Protected Solid State Switches
- Less Cabling and Connectors
- Automatic Transmission Verification
- Redundancy Features
- Reduction in Electrical/Electronic LRUs
- Low Current Switches
- Less Relay Logic

Maintainability

- Built-In Test Equipment (BITE)
- Automatic Failure Diagnosis
- Increased Status Visibility
- Reduction of Custom Cables and Improved Cable Accessibility
- Common Input/Output Circuit Boards
- Fewer Test Connectors
- Reduced Mean Time to Repair at Organizational and Direct Support Levels

Flexibility

- Ease of Adding Subsystems
- Interface Compatibility
- Reprogrammable Microcomputer for Operational Changes

It is also anticipated that ATEPS will significantly lower the life cycle cost and possibly lower the acquisition cost of vehicles.

## SUMMARY

The ATEPS technology offers a significant improvement potential for the design of electrical/electronic systems in combat vehicles. This potential includes Product Improvement Programs (PIPs) and future Army vehicle designs. Conceptual designs have been developed for expanding the ATEPS technology to tactical and other combat vehicle weight classes--XM2 (IFV) and an advanced self-propelled howitzer. Exploratory development tasks have been initiated to reduce the size of the ATEPS Core Element electronics through the utilization of recent advances in solid state technology and packaging techniques. Similar developmental tasks are being formulated for a second generation MDB using fiber optic technology with an extension of the technology to include slip rings.

PROGRESS WITH PIELSTICK ENGINE
DIAGNOSTICS AND EXPERIMENTAL RESULTS

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Abstract: In order to improve engine reliability and reduce maintenence expenses on ships, S.E.M.T. has developed a monitoring and diagnostic system for medium speed diesel engines used in marine propulsion or in power plants. The Pielstick Engine Diagnostic system has been presented at the 28th MFPG meeting. It automatically monitors wear of the piston top ring, main bearing alignment and shell wear, pressure charging circuit fouling condition and exhaust gas temperatures. A prototype system was fitted onto an 18 cylinder propulsion engine of the container ship M/V "RENOIR" which was commissioned in January 1978. This engine has now operated for 12,000 hours. The present paper reports the results obtained in the course of two years of operation of the system. The information given by the P. E. D. system is compared with that supplied by the servicing staff's routine inspections. This comparison has led S. E. M. T. to undertake new laboratory research which will concentrate on the problem of piston ring wear. The future improvements contemplated for the P. E. D. system as a consequence of this initial experience, as well as the necessity to have a better understanding of and a better monitoring system for the energy balance of the power stations, are explained.

Key words: Medium speed diesel engine; piston ring wear; crankshaft bearing safety; turbocharging monitoring.

INTRODUCTION

The Pielstick Engine Diagnostic system presented at the 28th meeting of the MFPG has been developed to improve the reliability and reduce the maintenance costs of the medium speed diesel engines with outputs ranging from 3000 to 20,000 kW.

P. E. D. monitors wear of piston top rings, main bearing alignment and shell wear, fouling conditions of the pressure charging system, and exhaust gas temperature.

The prototype fitted aboard the container ship M/V "RENOIR" has been operated for some 12,000 hours since January 1978.

In the course of these two years of operation, the P. E. D.'s indications have been compared to those obtained by conventional means.

The information and experience gained from this first prototype will be used to develop an advanced system.

## CRANKSHAFT MONITORING

The purpose of crankshaft monitoring is to prevent eventual failures in the shafting, the main causes of which are the abnormal wear of a bearing or the misalignment of the driven machine (reducing gear or alternator) with respect to the engine.

In the absence of a monitoring system, the distance between the webs of each crank are periodically measured for five angular positions of the shaft with the engine stopped.

A safety system based on the reading of the bearing shell back temperature will permit the engine to be stopped before the crankshaft cracks from overheating.

By using the P. E. D. system and measuring the vertical displacement of the shaft in its bearings, we eliminate both the periodic inspections and the heat alarm system.

A contact free displacement transducer is fitted in each main bearing on the engine Vee axis (figure 1). The signals received are analyzed in terms of three characteristics: (a) peak-to-peak amplitude, (b) mean value, and (c) minimum lift of the crankshaft journal in the bearings.

Any change in amplitude (a) with respect to the value stored during sea trials triggers an alarm. In the same way, a deviation of (b) or (c) values indicates a misalignment.

We have ascertained that sea roughness and the ship's cargo have little influence on crankshaft movement (figure 2).

The amplitude (a) of the crankshaft vertical displacement increases in a linear way, for instance, from 0.2 to 0.34 mm for main-bearing no. 5, when the rotation speed increases from 350 to 500 rpm.

The influence of crankshaft bearing wear on crankshaft displacement could be measured only in a main bearing of a 12 cylinder engine.

Aboard M/V "RENOIR" we observed that, except for two false alarms due to pick-up connections, the bearings behaved normally. This was confirmed by the continuous reading of the bearing temperature as well as by the periodic measurement of crankshaft deflection; all the parameters measured were stable.

The half bearings removed in the course of the semi-annual inspections were in perfect condition.

For now, we have not enough theoretical or experimental data about the relationship between the measured values (a), (b) and (c) and the defects to be detected.

Should it be compared with bearing temperature monitoring, the displacement reading made with the P. E. D. system offers the advantage of giving information on the causes of anomalies and higher sensitivity. It should allow the elimination of the periodic deflection readings, with, however the disadvantage of being more onerous and more difficult to keep in perfect working order.

PISTON RING WEAR MONITORING

The piston contains five to six rings of cast iron. They are coated with a protective layer of chromium or copper. In some engines, one or two segments are of pure cast iron, without coating.

When the chromium layer protecting the top ring is worn off, the ring deteriorates quickly. The piston must then be taken out and a new set of piston rings fitted.

The purpose of the piston ring wear monitoring is to determine the moment when the chromium layer of the top ring is worn off.

Inductive sensors are fitted in the engine cylinder liners, flush with the cylinder wall and in such places that all piston rings successively pass in front of them (figure 4). This allows the sensor to scan each ring and to emit an output signal whose amplitude varies depending on both thickness and magnetic properties of the coating material.

Should one of the piston rings be of pure cast iron, it then is possible to measure the thickness of the coatings of other rings by comparing the amplitude of the signals emitted at each ring passage.

Unfortunately, there is no non-coated ring in the PC2-5 engine and we therefore had to use the coppered first compression ring as reference. Figure 5 shows such a ring after 1500 hours of operation. It may be observed that no copper remains on the ring face, only in the grooves. Such a ring produces, when passing in front of the sensor, a weaker signal than that obtainable from a pure cast iron ring.

Figure 6 shows a chromium plated ring that is worn unevenly. This ring had no chromium remaining along 2/3 of the face length, but the remaining third and the grooves were still coated.

In order to be able to use the values recorded on the engine, we measured, in the laboratory, the signals from various compression and top rings from various cylinders on M/V "RENOIR". It has been observed that, from the sensor standpoint, the wear on the second compression ring may be considered to be evenly distributed.

The curve of figure 7 gives the mean chromium thickness in terms of amplitude ratio between the signals obtained from both rings.

The dispersion around the mean value may reach $\pm$ 30% which is explained by the fact that the active surface of the sensor overlaps the ring face and that, due to its wear, the chromium is no longer evenly distributed on the ring. Larger signals are obtained from severely worn chromium plated rings than from compression rings.

The "no more chromium" alarm shall therefore be tripped for an amplitude ratio of 1.14 and not 1.

The points plotted in figure 8 represent, for a four cylinder engine, the mean value of 50 measurements made during a 100 hour period. It may be observed that these values which are used to obtain the mean chromium thickness vary over a wide range because of a rotation of the ring in its groove.

Figure 9 indicates the averaged thickness for about 1000 hours.

Figure 10 shows chromium distribution at the periphery of three piston rings removed after 4200, 7200 and 10,000 hours of operation.

A comparison of figures reveals that in two out of three cases, there are large differences between the mean thicknesses measured on the removed piston rings and those read by the P. E. D. system.

We believe that this large difference results from an uneven rotation of the ring on the piston. From our observation, the piston rotation amplitude is short when the engine runs at constant speed and larger when the engine is slowing down.

It's also possible that the ring is subject to an angular alternative displacement so that the pick-up scans only a narrow band of the ring for a longer or shorter time.

For the time being, the monitoring of piston ring chromium wear does not work satisfactorily. We must go on with our research with regard to piston ring rotation and decide if it would be possible to determine the actual chromium wear by using two or three pick-ups per liner. We also must try other types of smaller diameter, cheaper pick-ups. The cost of the system becomes onerous when the number of pick-ups per cylinder is increased.

## TURBOCHARGING AND EXHAUST GAS TEMPERATURE

The constant cleanliness of the turbocharging system is important to thermal efficiency as well as the lifetime of a number of engine components such as the exhaust valves.

We have therefore incorporated into the P. E. D. system the means to automatically make the measurements normally made by the conscious users, i. e., exhaust gas temperatures, air pressures and temperatures, and turbocharger speeds.

Figure 3 shows the variation of the temperature average read at each cylinder outlet and charge air pressure in terms of turbine speed or power rating.

These curves show the correlation of the various parameters. Washings of turbine, compressor, and air cooler were more or less efficient, since their characteristics remain stable sometimes, whereas they deteriorate at other times.

The two mechanical interventions were more efficient, especially the turbocharger inspection.

This stage of the P. E. D. shall be completed and improved in order to determine the efficiency of the turbocharging system and, as far as possible, a more detailed analysis of the exhaust gas temperature.

## HARDWARE PROBLEMS

We had to replace three pick-ups and one signal conditioning card out of 46 measuring channels. On the data processing side, it is the printer which caused us trouble as it was not adapted to work in the engine room ambiency.

Also, we observed three faulty indications because of defective contacts: two in microprocessor cards and another due to a faulty refitting of a pick-up connection after a main bearing inspection.

## CONCLUSION

The P. E. D. system fitted on M/V "RENOIR" was only a prototype. It shall be re-designed and improved not only because it does not work satisfactorily with respect to its original purposes, but also because the ship's users demand other information regarding fuel savings.

S.E.M.T. are going to pursue their research on piston ring wear and complete turbocharging monitoring in order to compute the efficiencies of the system.

The monitoring of crankshaft displacement in bearings will be super seded by 1/2 bearing temperature readings, but we shall pursue our research on crankshaft displacement using theoretical as well as experimental analyses.

We shall add to the P. E. D. the measurement of the output and of the instantaneous consumption of fuel to help the captain determine the ship's speed from an economical criterion.

## CRANKSHAFT SENSOR

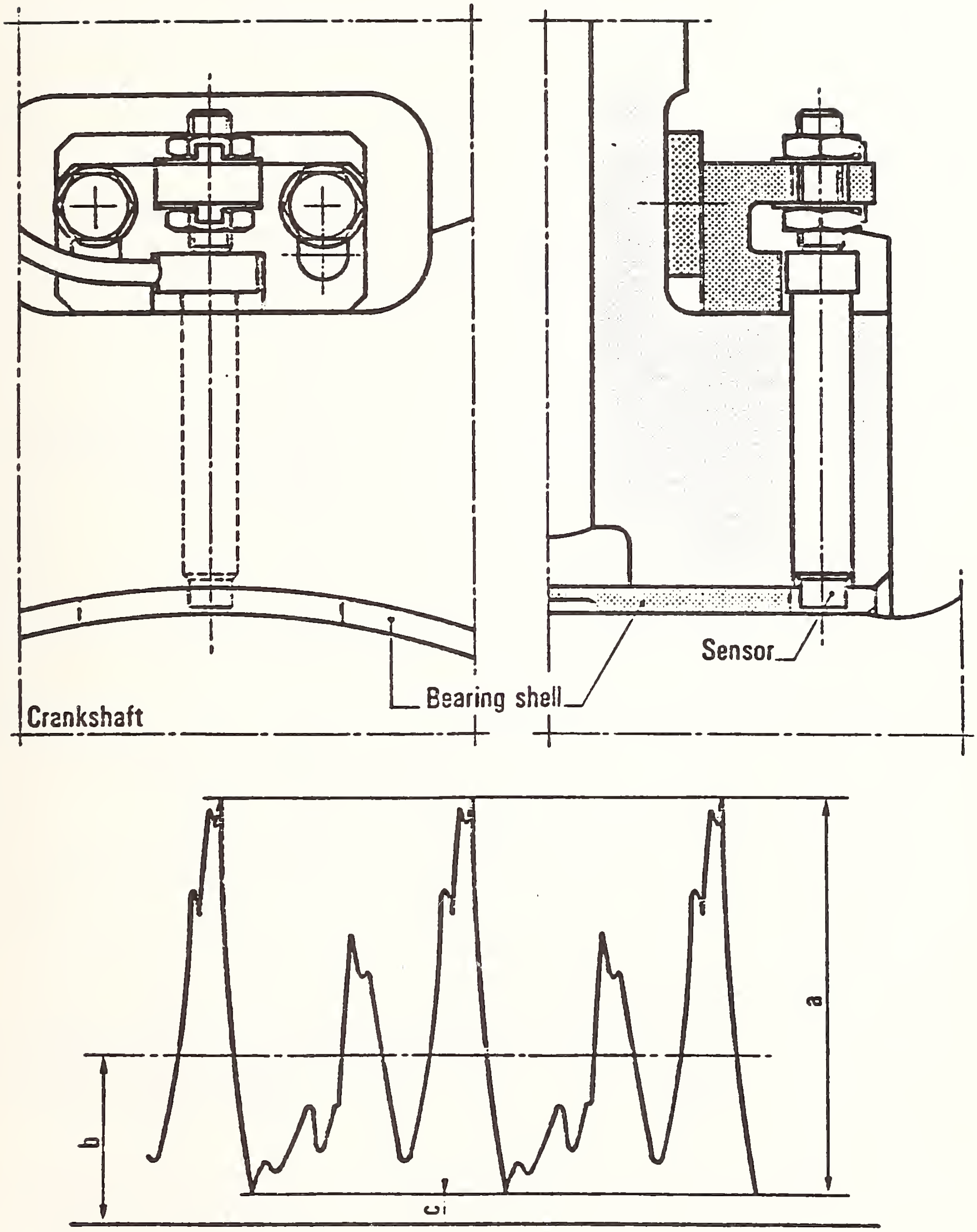



Figure 1.

RENOIR: Influence of deplacement

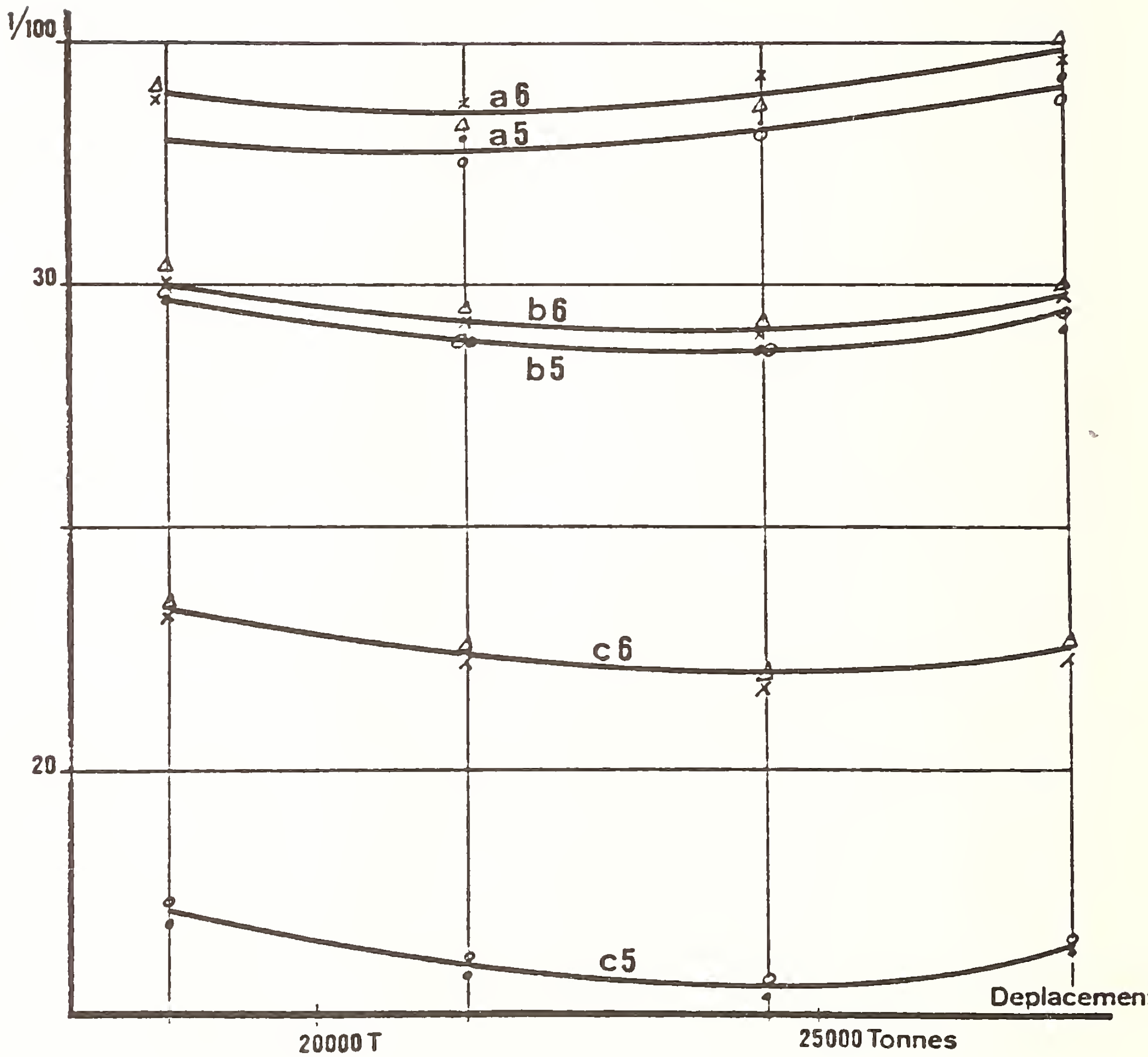



Figure 2.

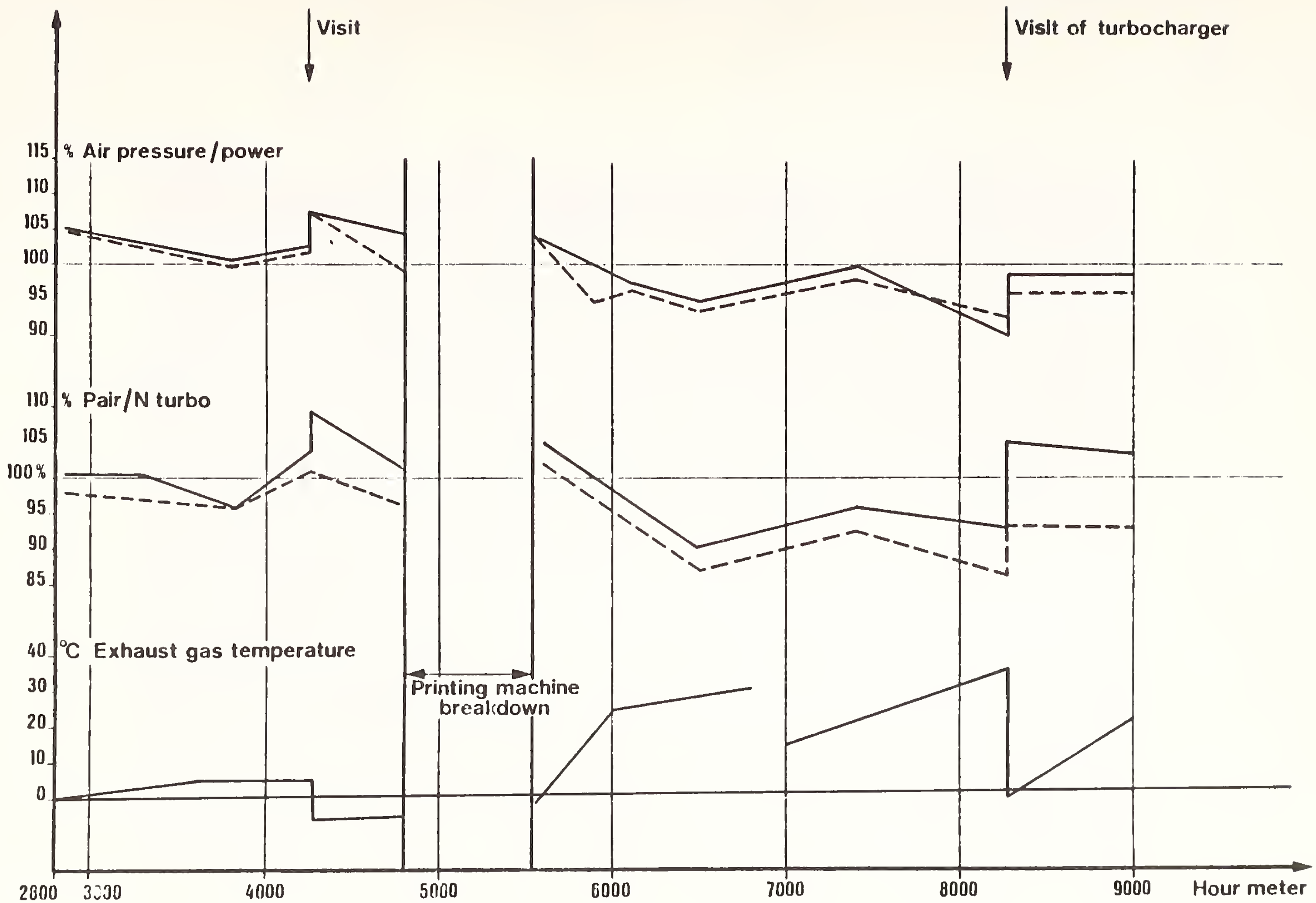

Visit
Visit of turbocharger
% Air pressure / power
115
110
105
100
95
90
% Pair/N turbo
110
105
100 %
95
90
85
°C Exhaust gas temperature
40
30
20
10
0
Printing machine breakdown
2800
3000
4000
5000
6000
7000
8000
9000
Hour meter

Figure 3.

## PISTON RINGS SENSOR

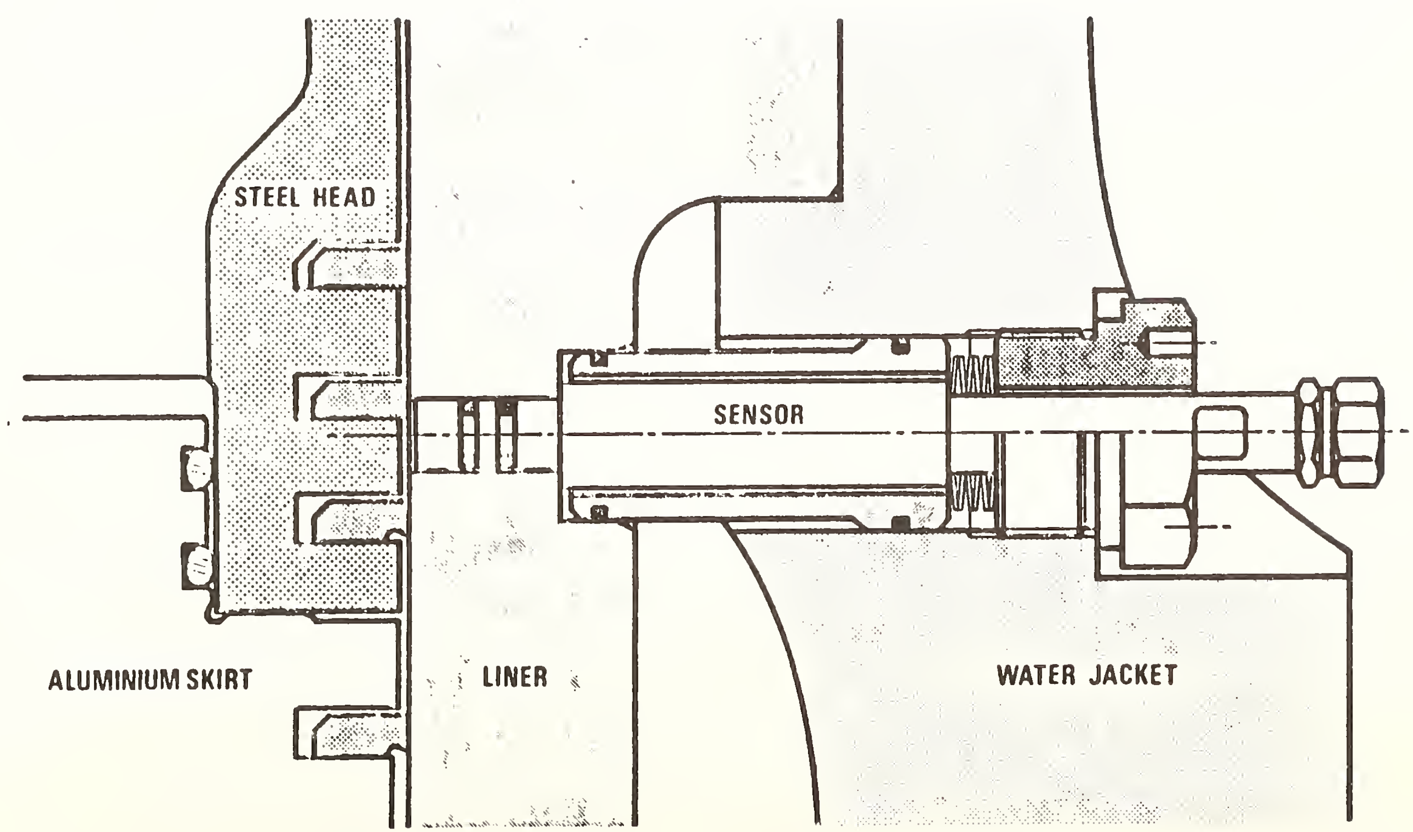
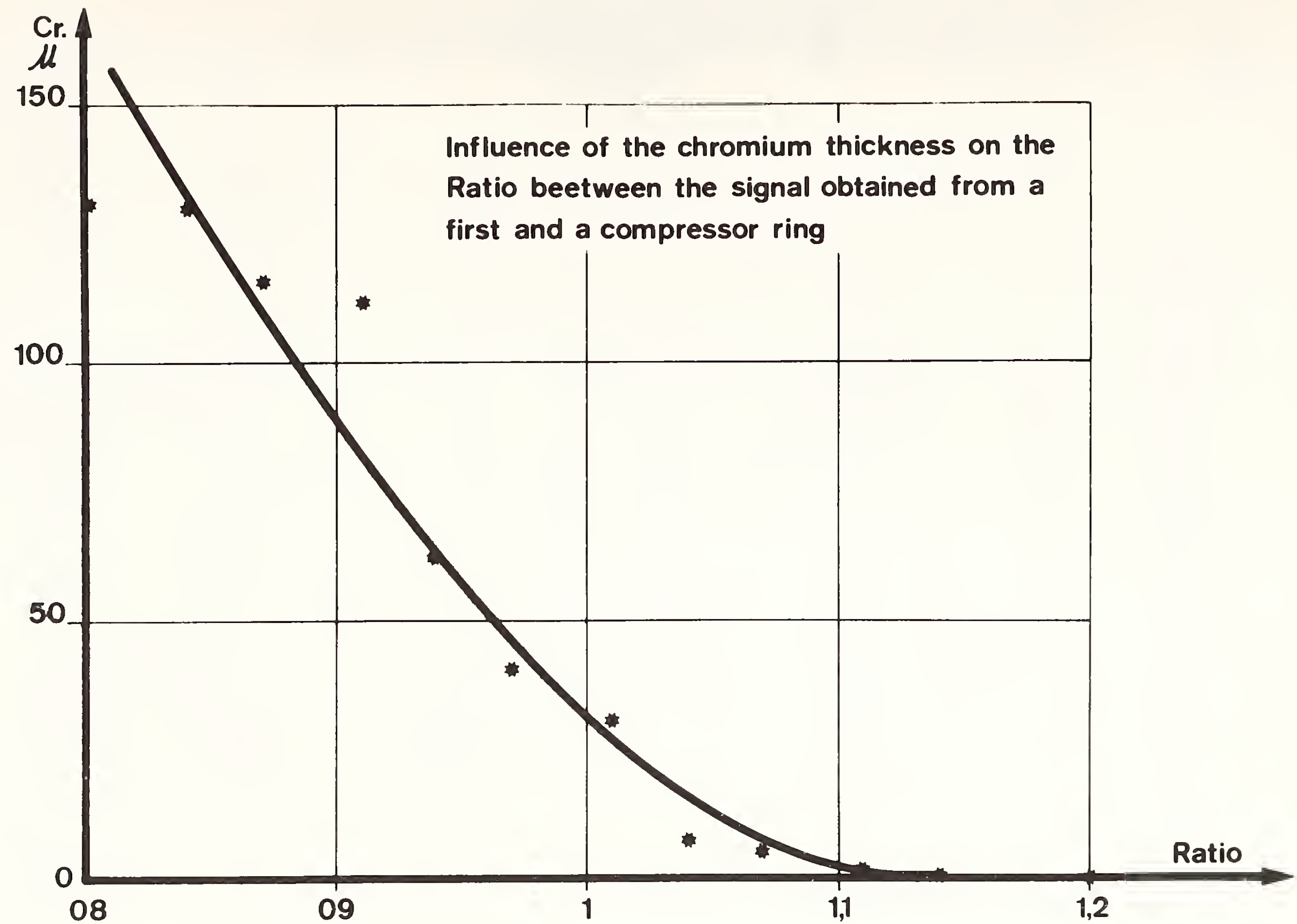



Figure 4.

Figure 5. Copper coated compression ring

Figure 6. Chromium coated fire ring (worn out)


Cr.
μ
150
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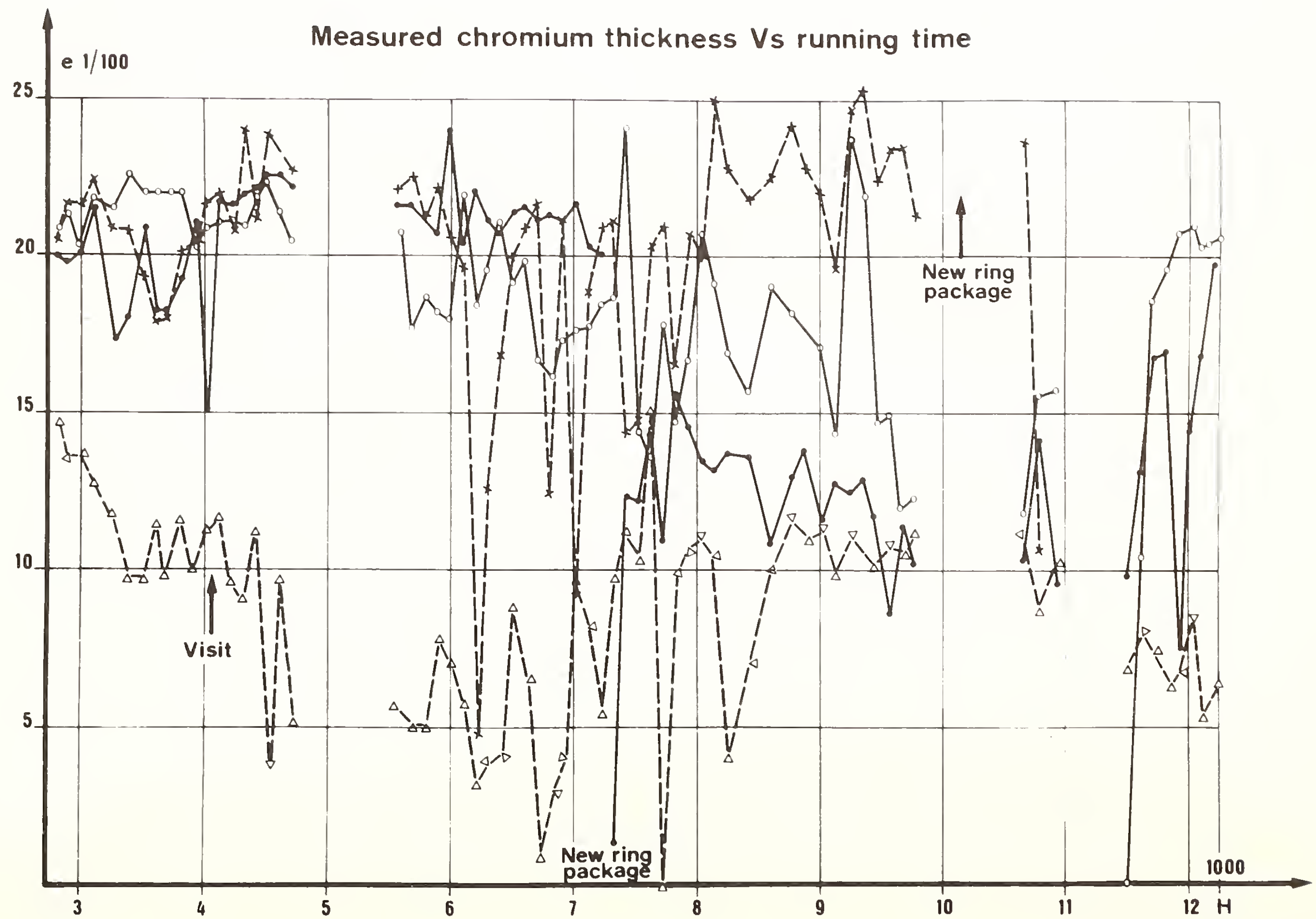
1
1,1
1,2
Ratio
Influence of the chromium thickness on the Ratio beetween the signal obtained from a first and a compressor ring

Figure 7.


Measured chromium thickness Vs running time
e 1/100
25
20
15
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5
3
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11
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1000
H
Visit
New ring package
New ring package

Figure 8.

## Average measured chromium thickness

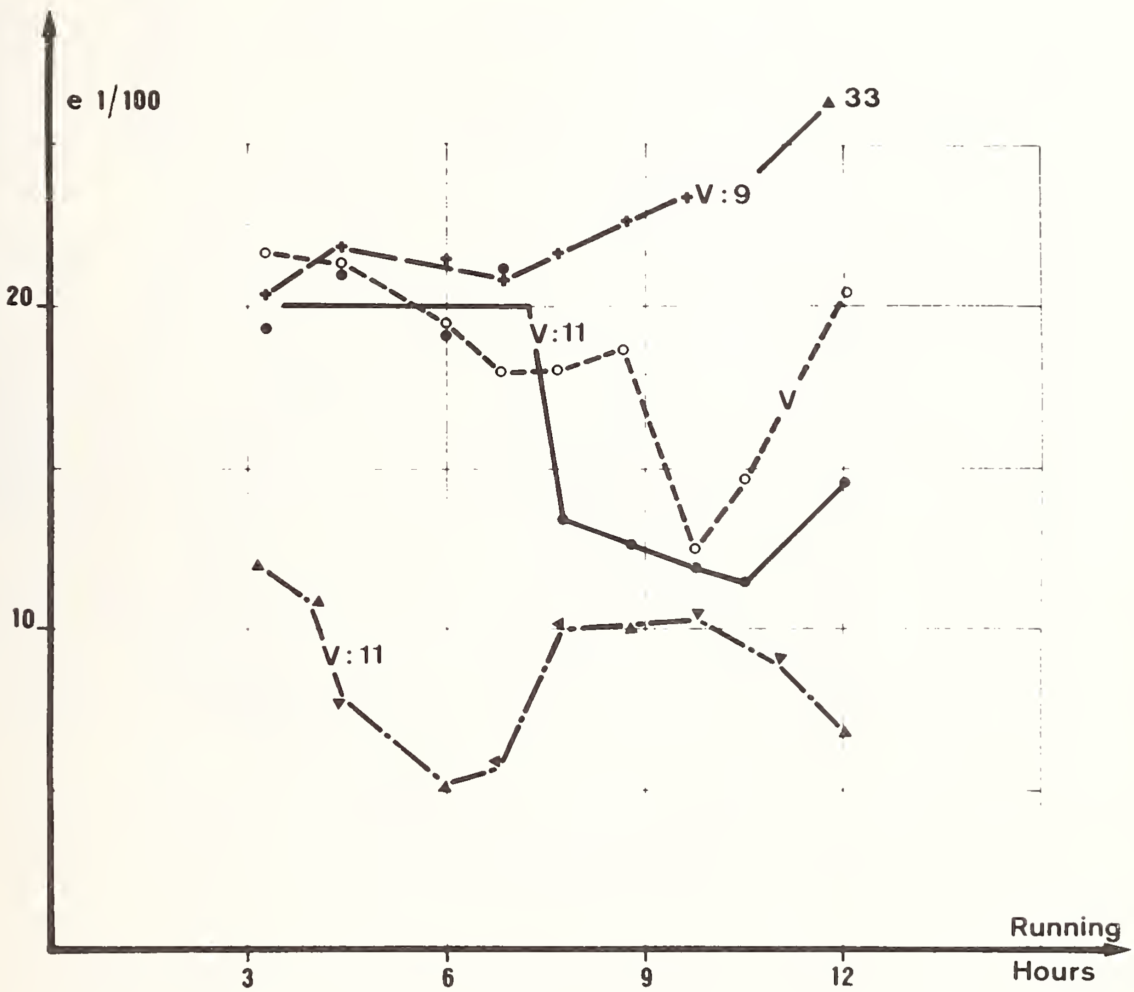


Figure 9.

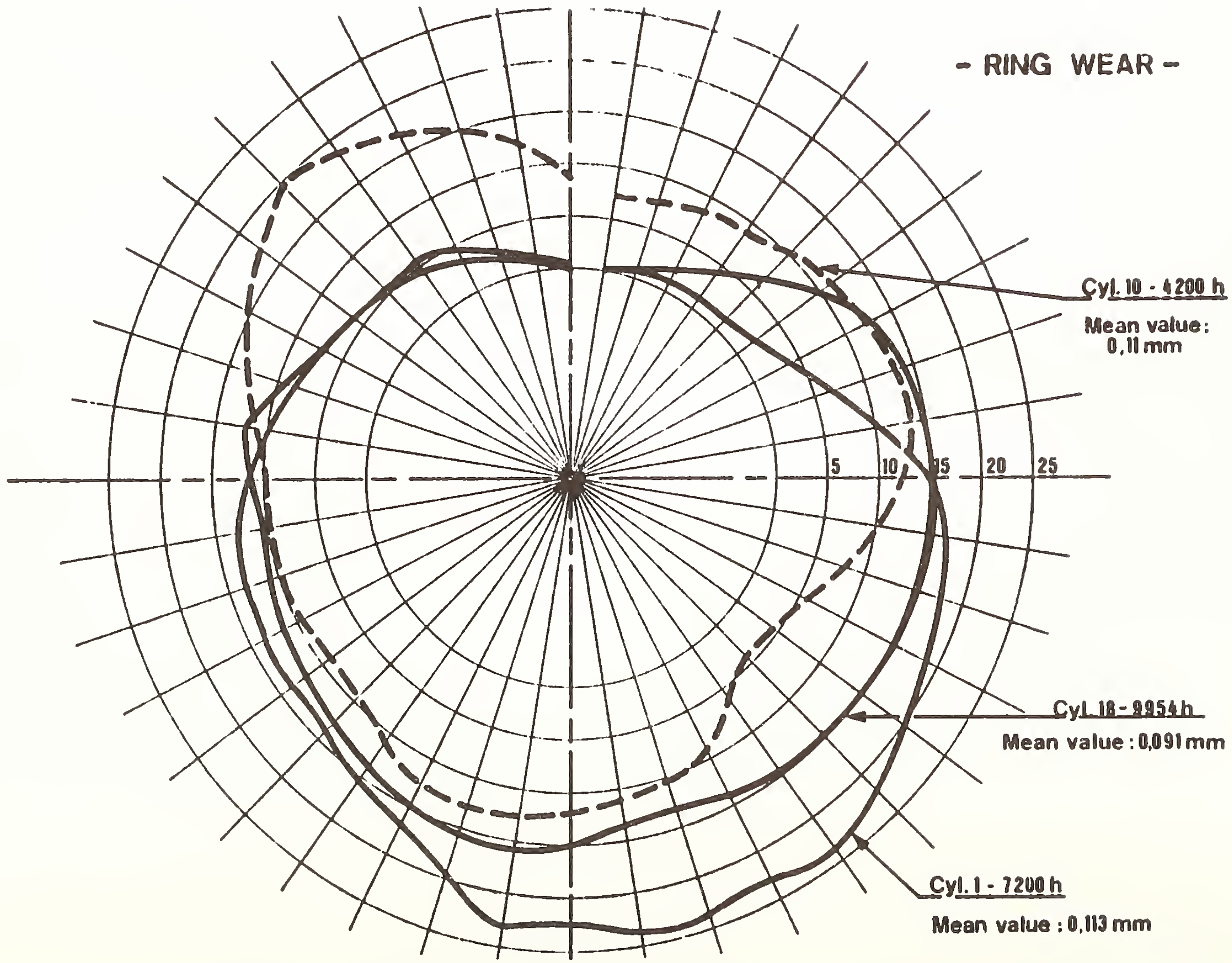
- RING WEAR -
Cyl. 10 - 4200 h
Mean value: 0,11 mm
Cyl. 18 - 9954 h
Mean value : 0,091 mm
Cyl. 1 - 7200 h
Mean value : 0,113 mm
5
10
15
20
25

Figure 10.

# AN ENERGY CONSERVATION CONCEPT FOR OPERATING MACHINERY

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Abstract: Existing planned maintenance systems for complex machinery, such as those used aboard ships and aircrafts, in power plants, and processing plants, is viewed as time consuming and sometimes leads to unnecessary maintenance actions being performed. Moreover, such maintenance practices have become expensive and are not effective to keep complex machinery functioning with maximum efficiency.

In this paper, a conceptual approach to improving planned machinery matenance as well as optimizing energy expenditures is described. This concept, making wide use of color displays and interactive techniques, represents a new practical solution to the implementation of an as-needed machinery maintenance, as opposed to the regularly scheduled open and fix procedure mostly performed. The implementation of this concept requires an on-line machinery monitoring system to detect equipment degradations. All performance monitoring and energy management tasks will be coordinated from a central location called the Engineering Maintenance Center (EMC).

This concept of machinery performance monitoring is discussed, highlighting such parameters as the functions to be performed by the EMC and the role of the maintenance personnel. Scenarios are also presented to illustrate the EMC functions such as energy management, machinery performance optimization, and detection, diagnosis and isolation of faults.

The immediate benefits to be obtained from this application of a reliability centered maintenance are improved energy utilizaton and machinery performance, and decreased maintenance costs.

Key words: Machinery monitoring; Reliability Centered Maintenance; Engineering Maintenance Center; Energy Conservation.

Introduction: As operating machinery maintenance becomes more diverse and complex, their operating, maintenance, and performance monitoring have become more difficult and costly. Development of modern maintenance monitoring techniques to provide machinery operating status, levels of performance, failure detection and prediction, and determination of maintenance requirements based on Reliability Centered Maintenance offers immediate benefits in improved energy utilization and machinery performance, and decreased maintenance costs.

The objective of this concept is to highlight the need for development of an automated system for monitoring operating machinery degradation/ failures with enough lead time to improve maintenance planning. With machinery operating at its optimum, energy is conserved and equipment availability is increased.

Approach: Existing planned machinery maintenance systems have been based on the myth that machinery operates as expected or with a proven life. Therefore, maintenance performed has sometimes been unnecessary. There is additionally a lack of adequate information (historical data) on operating machinery conditions with little feedback to determine the cause of machinery breakdown.

The need to improve upon present machinery maintenance methods has initiated this conceptual approach centered around Reliability Centered Maintenance (RCM). This concept represents a new solution to machinery maintenance that will be performed mostly on demand rather than regularly scheduled. Its implementation will also require an on-line monitor system to detect equipment degradation and direct maintenance personnel to appropriate equipment in a timely manner.

Reliability Centered Maintenance: A Reliability Centered Maintenance System is by definition a maintenance concept of making operating machinery more dependable. By monitoring critical machinery parameters, we can (1) conserve energy when running machinery in optimum conditions (2) better plan shutting down and rework of machinery under stress (3) improve safety conditions in machinery spaces and (4) the on-line machinery monitoring system can aid in training of personnel.

Present maintenance procedures are structured as shown in Figure 1; usually referring to preventative, corrective, and emergency maintenance. This historic method of accomplishing planned maintenance relies upon time based open-and-inspect procedures. This method does not allow for the variable life of several equipments of the same model. Also, it has been found that each open-and-inspect operation increases the chance of future failure of the equipment due to accidental improper reassembling practices.

The next section will describe the concept for an Engineering Maintenance Center (EMC) where the human operator assisted by an on-line monitoring system will document machinery operation in a suitable format to plan Reliability Centered Maintenance.

Engineering Maintenance Center Concept: The increase in size and complexity of operating machinery systems has necessitated the design of new on-line monitoring systems that can process large amounts of information. This new system had to be developed taking into consideration the experience and limitations of a human operator. A digital computer acts as the link between operator and machine. This link is implemented in software allowing for improved flexibility and upgrading

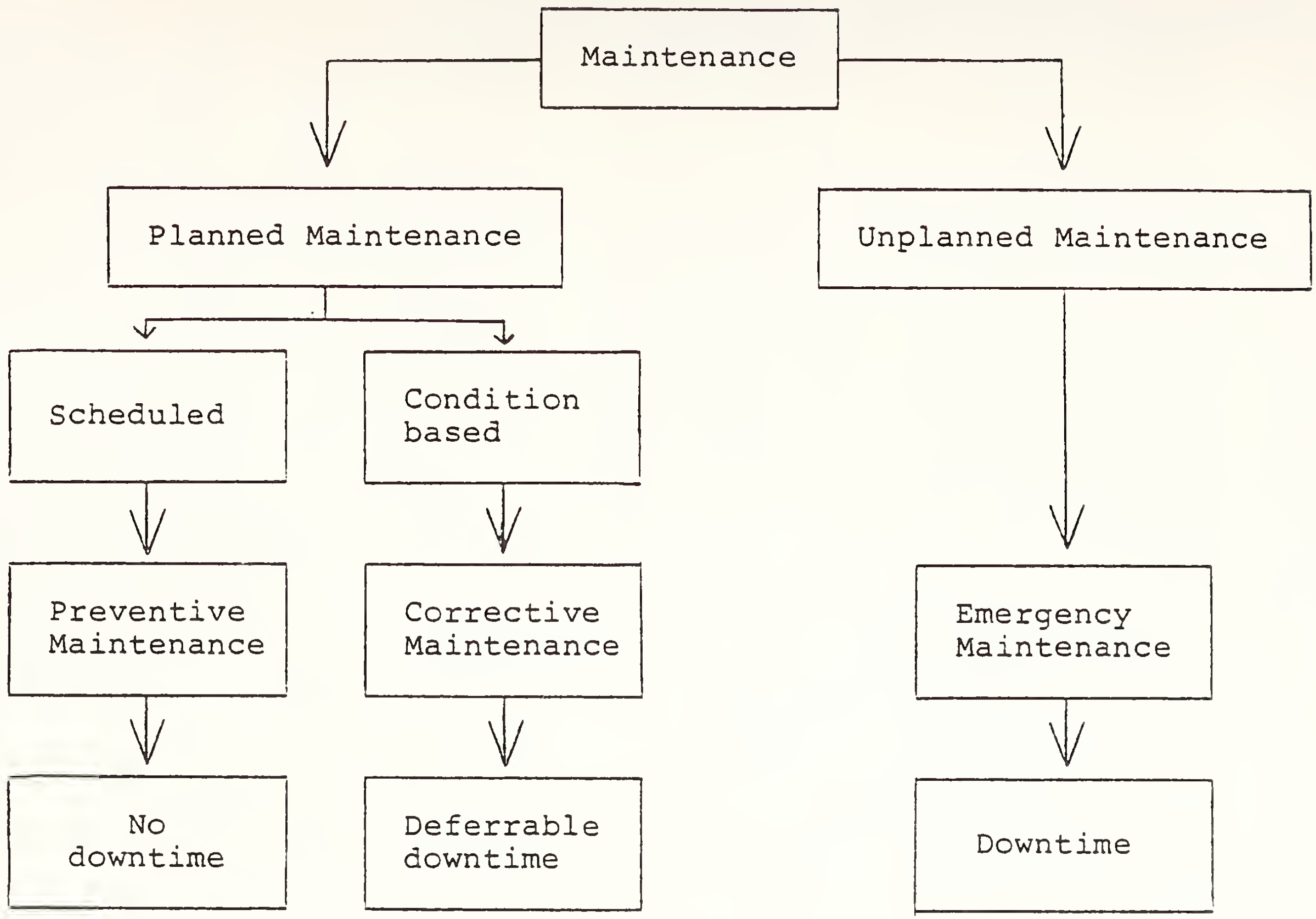



Figure 1 - Maintenance Procedure Classifications

capability. Also, the complexity of the machinery is handled by the computer software which converts the multitude of machinery sensor inputs into data directly interpretable by the monitoring system operator. The operator is only confronted with the data necessary for making a decision. The machinery operating data has to be presented to the operator in a format easily adapted to his level of expertise and understanding.

This operator oriented interface has been very successfully applied in industry, and represents an answer to the control of increasingly complex machinery. The same display principle has been adopted in the EMC. The maintainer is responsible for the running condition of the machinery. Such a duty requires the access to machinery parameters and the knowledge of possible failure causes, to forecast a correct diagnosis. These requirements are easily achieved with the help of a computer where large amounts of information can be processed. A step-by-step computer aided procedure will guide the maintainer through the complexity of the detection, diagnostic and prognostic phases.

The need of a monitoring system adapted to the operator might have seemed unrealistic until the occurence of the well publicized accident at the Three-Mile-Island (TMI) nuclear power plant. This accident illustrates the inherent problem with control rooms characterized by the display of all the monitored parameters. Investigations from the Nuclear Regulatory Commission (NRC) have indicated that this particular accident was preventable. It seems that it all started as a series of minor malfunctions which cascaded into a serious accident. It is believed that human error is one of the major factors that contributed to the event.

In the future, it is safe to say that the complexity of machinery and their controls will keep increasing, and more and more accidents are to be expected due to human errors. This new methodology of the EMC replaces analog monitoring display devices with alphanumeric/graphic devices such as color cathode ray tubes (CRT), and information displayed is automatically limited to the minimum set of data necessary to the operator for his present task. The EMC will be in charge of machinery surveillance including the planning and carrying out of maintenance.

The use of a computer in the new maintenance methodology will allow for efficient computer aided tasks such as training and inventory control. Typical machinery failure scenarios could be designed in the system to render the training more realistic and adapted to the level of the trainee.

The increasing complexity of operating machinery is being absorbed by a computer controlled interface whose function is to assist the maintainer in its detection, diagnostic and prognostic task. The prognostic capability of the EMC is a unique function not attainable by today's maintenance philosophy. It will cause a noticeable reduction in the number of emergencies/catastrophic failures by optimizing the performance of maintenance of machinery.

Computers and microprocessors constitute the needed technology to give maintainer accurate and timely responses, as well as the capability of processing large amounts of data. However, such an advantage would be cancelled if the interface between computer and maintainer is too complex. The level of sophistication of monitoring techniques has been rising at an accelerated pace while man has changed little. Therefore, a special emphasis should be given to the conception of displays that are adapted to the operator instead of having the operator adapt to fancy but ineffective displays.

Information on display techniques and major on-line monitoring systems is described in the following section. These applications were developed from extensive industrial research in man-machine interface, and reinforces the fact that the quality of the display determines the maintainer's ability to successfully perform his search and identifying tasks. Considerations for Alphanumeric/Graphic display devices for on-line machinery monitoring systems will be discussed also.

Man-Machine Interface: In order to successfully design display systems, an appreciation for the role of man in the system and his human information-processing capability is required. The human factor is a function of several variables: (1) complexity of information displayed, (2) operator tasks, (3) environmental factors, (4) operator characteristics, and (5) display characteristics.

The complexity of the information to be displayed and the operator's tasks are usually predetermined and therefore, can not be controlled when designing a display. Likewise, the environmental factors and operator characteristics are virtually uncontrollable. However, in order to generate an effective display, the designer must take into account all these variables when solving the final one display characteristic.

The following sections will be concerned with the display characteristics and their effects on human performance. These display characteristics include: (1) display techniques, (2) coding techniques, (3) color selection, (4) density of information, (5) symbol size and resolution, and (6) luminance.

Display Techniques: Display techniques deal with the manners in which information is organized within the overall display. All displays can be categorized into four major groups: alphanumeric, graphic, representational and combinational, i.e., a mixture of two or all of the others.

Alphanumeric displays are most useful while performing diagnostic functions. They use strictly textual (alphanumeric) and numeric coding.

Search times will be longer due to the amount of detail on each display, but this can be tolerated in light of the beneficial information received. However, there is no requirement for these displays to be arranged in a purely columnar form. Related parameters may be oriented on the screen in a logical manner to enhance search.

Graphic displays are usually used for prognostic functions. They include line or bar charts to indicate trending or historical information. They are realatively clean and very easily understood.

Representational displays are most useful while controlling or monitoring a system. Symbols denoting major system components indicate, in Figure 2, the overall functioning of a gas turbine power control system. Key parameters (e.g., temperature, pressure...) that affect overall system operations could also be displayed and used to evaluate performance.

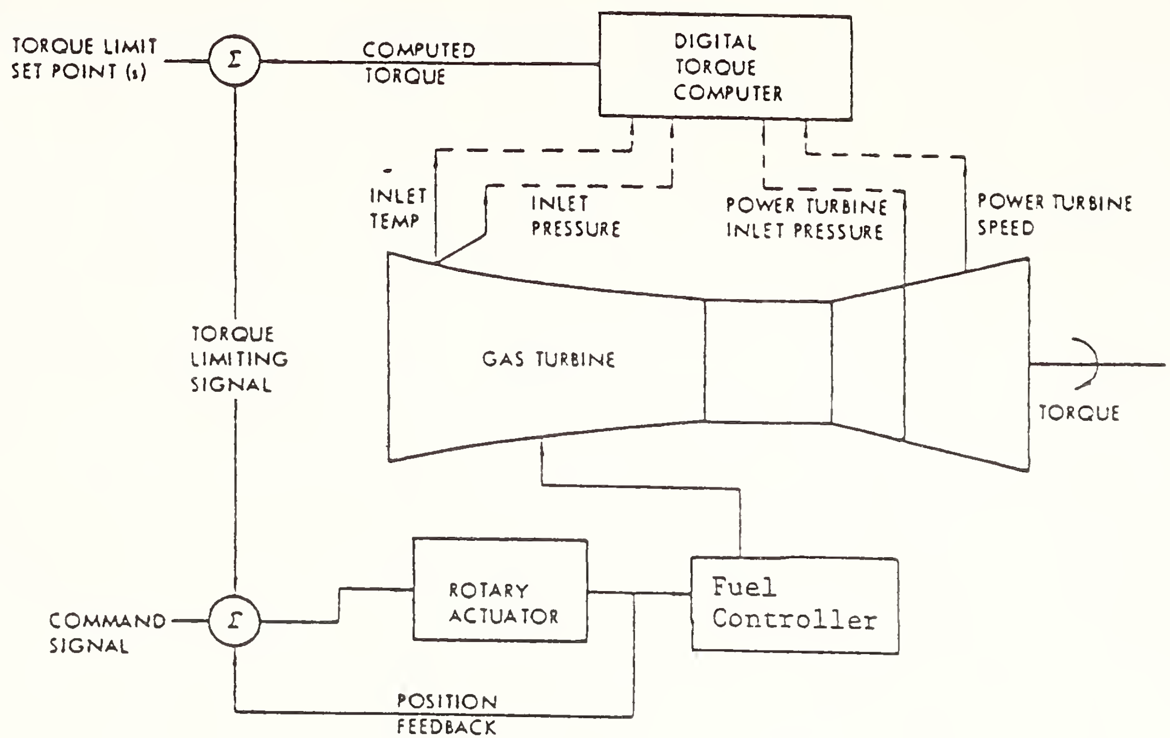
Coding Techniques: In choosing coding technique, a designer must decide how to best code information to perform a given task. To accomplish this, the designer must evaluate several factors, including: (1) space available for coding, (2) type of information to be coded, (3) coding function (i.e., alarm,identification), and (4) visibility for coding, i.e. resolution, contrast, luminance and illumination.

Coding techniques can be divided into five different methods: numeric, shape, color, blink and textual (alphanumeric). In general, color coding is very useful when searching or locating a class of object which is most important. Color coding is described in more detail in the next section. Textual, or alphanumeric, coding is most useful when identification is desired; symbols/shapes are useful coding devices when qualitative objectives are represented. Blinking is used almost exclusively as an alarm or warning signal.

As few coding levels as absolutely necessary should be used to specify levels within each code type; e.g., when using color coding it is recommended that no more than eight colors be used. Increasing the coding levels beyond these recommended values will have an adverse effect on the operator accuracy.

Color Selection: The most effective color display is one in which color is used sparingly, only when needed, and where it uniquely conveys information that other codes can not or do not provide. The criteria for selecting a specific set include: (1) maximun wave length separation, (2) high color contrast, (3) high visibility in specific application, (4) compatability of use with conventional meanings, (5) legibility and ease of reading, and (6) high saturation (i.e., the two colors labeled pink and red have approximately the same hue but the red would be highly saturated and the pink would be low in saturation or desaturated. Zero saturation colors are black, gray and white).

Figure 2:  Representational Display



Frame Density: Frame density involves determining how much data can be placed on the screen before the amount begins to adversely affect the operator's ability to perform his basic task. An analysis of existing CRT displays that were qualitatively judged "good" revealed a loading on the order of 15 percent. The remaining area constitutes "white space" that is essential for clarity in any display. Furthermore, the amount of variable data in these displays never exceeds 75 percent of the total active area. The product of these limits dictates that no more than 18.75 percent of the screen should contain information of continued interest to the operator.

Density within the display is also an important consideration. One would like to know the maximum number of characters, the appropriate symbol size, and proximity to other information areas.

Display Luminance: The appropriate overall or average luminance of any display depends upon the viewing conditions, which include: (1) background luminance, (2) ambient illuminance, (3) symbol size, and (4) display colors.

If the display is used in a dark room, and the observer is generally dark adapted, then the display luminance should be between 10 and 100 foot-Lamberts. In lighted areas, or under normal room illuminance, the display average brightness should be on the order of 100 foot-Lamberts. The advantage of a brighter display is the improvement in visual acuity of the eye. However, if the display is too "bright" for the given viewing conditions, then the display appears as a glare source and is quite annoying to the operator and generally fatiguing.

Existing On-Line Monitoring System Industry Experience, Honeywell/TDC 2000 (Ref: 8): The TDC-2000 Basic Operating Center usually provides: (1) Three color television-type monitors which, using a combination of analog and digital methods, provide a variety of displays, including overviews, group, loop, trending and alarm displays. All of these displays have been humanly engineered to best convey process control information to the operator, (2) a single keyboard with dedicated function pushbuttons for each CRT monitor. These keyboards allow any person, either operator or engineer, with a working knowledge of the process and no knowledge of programming to be quickly trained to operate the system, (3) the microprocessor and its associated circuitry, which permits communication along the Data Hiway with up to 63 devices located as far as a mile away, allow the customization of displays to a particular process plant and provide security of operations through self-diagnostic routines, (4) a cassette package which enables cassette loading and /or recording of data and displays. Once one Operator Station has been configured, the redundant Operator Stations can quickly be identically configured by loading the cassette. Information can be stored on cassette for needs. A plant's system can be preconfigured on cassette to speed startup operations, (5) three-pen recorders which allow hard copy recording of process variables located up to a mile away, and (6) printers which print on command alarm logs, process variable logs, historical and real-time trending and other display reproductions.

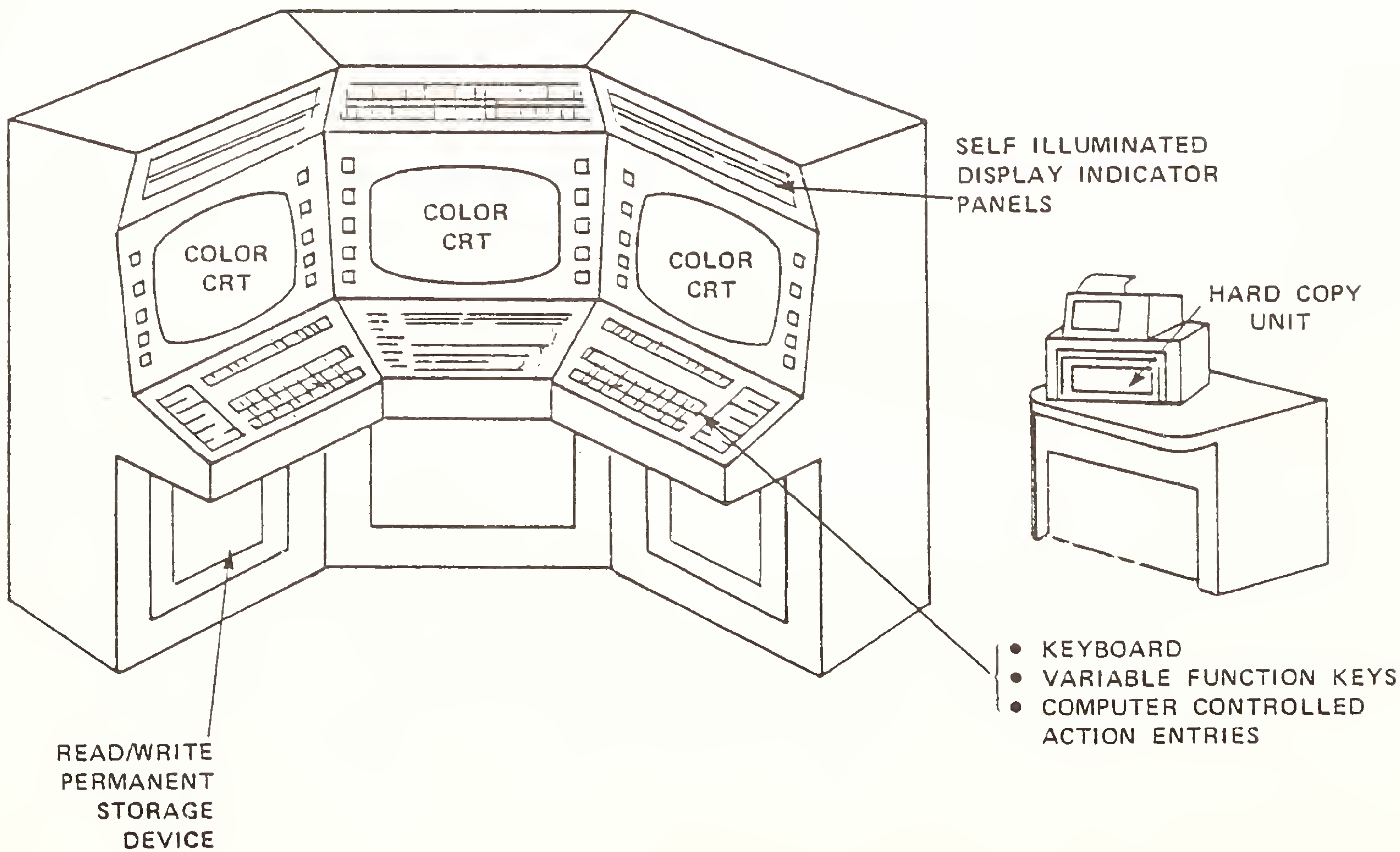
Taylor/MOD III (Ref: 4 ): The MOD III process control system provides flicker eliminating, high resolution P-39 phosphor, CRTs in combination with a separate, full-color, CRT. The television-type monitors provide a variety of displays including overview, status, loop, and trending displays. Alarms appear on all displays as flashing, reverse-video signals, highlighting the problem variable, and as flashing LEDs on the page selector/alarm panel.

The system also provides a cassette package for recording, a pushbutton keyboard, single loop microprocessor based controllers and printers.

In manufacturing and quality assurance, high reliability begins with component quality but does not stop there. MOD III specifications include; (1) system designed to operate in high humidity and over a range of 40°F to 120°F, (2) CMOS integrated circuits for maximun noise immunity, (3) ceramic ICs, impervious to moisture damage and stable over wide temperature range, (4) non-corroding, gold-plated contacts on PC boards and bus connectors, (5) PC boards coated to prevent corrosion, permit operation in high humidity, (6) long-life LEDs for system indicators, (7) ferroresonant power supply transformers for superior regulation, (8) semiautomatic backplane wiring with computer checkout to eliminate wiring errors, (9) 169-hour burn-in to eliminate weak logic components, and (10) 30-day burn-in of system and all spares.

Engineering Maintenance Center Functions: A description of anticipated EMC functions are below. It is emphasized that all of these functions will be software implemented. This software implementation of functions will be mandatory to increase the versatility of the system by allowing these functions to be added to, modified or deleted without requiring a system reconfiguration. The EMC functions are: (1) energy conservation - (a) machinery performance optimization, (b) resource management, (c) reliability centered maintenance, (d) inventory control. (2) ancillary functions - (a) equipment status, (b) data base management, (c) computer aided training, (d) system configuration, and (e) failure mode analysis.

Engineering Maintenance Center Description: The EMC operation will be centered around interactive techniques using terminals for input-output. A printer or hard copy unit will be required for recording permanent records of tasks such as alarm messages, machinery historical data, and documenting maintainer's activity. CRT alphanumeric/graphic type displays will be used to evaluate machinery performance and to efficiently perform the EMC functions. Figure 3 gives an example of an EMC arrangement.




EMC Arrangement

The CRT displays selected for the EMC console will provide graphics for diagrams and color for catching attention to status and other conditions. Color speeds identification, improves visualization and reduces response time. The use of color enables the display of more information simultaneously which results in the surveillance of a greater number of critical parameters.

A typical example of a on-line machinery monitoring display is shown in Figure 4.

Conclusions: Existing planned maintenance systems for operating machinery is viewed as time consuming, and sometimes leads to unnecessary functions being performed. With operating machinery systems becoming more complex and with a limited maintenance budget available, a catastrophic failure of machinery could have a staggering effect. A solution of these problems could be the implementation of an on-line monitoring system using modern color display techniques that could logically plan maintenance tasks in a reliability centered form. The results of implementing such a maintenance system will be twofold: (1) energy efficient machinery systems and (2) safer working environments for the maintainers.

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Figure 4 - MACHINERY CAPABILITY DISPLAY SAMPLE

| Systems | Available Maximum Capacity | Operative Load (%) | Efficiency (%) |
|---|---|---|---|
| Electric Plant | | | |
| o GTG #1 | 2,000kw | 50 | 82 |
| o GTG #2 | 2,000kw | 50 | 76 |
| o GTG #3 | off line | - | - |
| Air Conditioning | | | |
| o AC #1 | off line | - | - |
| o AC #2 | 600 tons/mn | 90 | 74 |
| o AC #3 | off line | - | - |

# ON-BOARD INSTRUMENTATION TECHNIQUES FOR THE DEVELOPMENT OF A GAS TURBINE ENGINE FOR TRUCK APPLICATIONS

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Abstract

The development of a gas turbine engine for use in highway and off-road trucks requires the implementation of considerable instrumentation techniques. The multitude of operational conditions that can be encountered in the operation of a vehicle on the roads are very difficult to accurately simulate in a test cell. A test vehicle was outfitted with a large, on-line digital data system with multi-readout capability, as well as a multitude of script chart and analog instrumentation.

COMMENT

Mr. Olivier arranged to have the test vehicle driven from Phoenix to Santa Monica for the MFPG Symposium and provided a demonstration of the instrumentation. Editor

# MARINE BALL/ROLLER BEARING MONITORING BY SHOCK PULSE MONITORING


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Abstract: The American Steamship Company (ASC) (a subsidiary of GATX Corporation) operates a fleet of modern self-unloading bulk carriers on the Great Lakes. A group of these ships with similar drive trains have experienced bearing failures in a 1740 Kw generator. GARD, INC., was charged with the task of evaluating available bearing monitoring techniques and equipment for their ability to detect and predict bearing damage prior to destructive failure.

SKF Technology Services of SKF Industries, Inc. was called in as a consultant in the area of roller and ball bearing damage detection. They have developed a production instrument (Automatic Shock Pulse Analyzer, or ASPA) based on an advanced application of this technique. During the summer and fall of 1979 a number of critical bearings were monitored on three sister ships. Readings were correlated as to spectral energy content and prototype ASPA readings. In this set of evaluation tests, a bearing on one ship was classified as damaged. During maintenance and repair operations that winter, the suspect bearing was removed and found to have serious outer race damage.

During 1980, a production ASPA has been installed in one of three sister ships (THE MV H LEE WHITE) and a log of five crucial bearings is being kept.

When thoroughly proved out, further installations will be evaluated for similar machinery in the fleet.



Key words: Automatic alarm of bearing malfunction; Marine Machinery Bearings; Microprocessor based shock pulse analyzer; Multiplexing bearing monitor channels; Prediction of bearing failure; Shock pulse bearing monitoring.

The American Steamship Company (a subsidiary of GATX) operates a fleet of 20 self-unloading dry bulk cargo vessels on the Great Lakes. These ships range in length from 580 to 1,000 feet and, in cargo capacity, from 10,800 to 66,000 tons. In three of these ships, propulsion consists of two 3600 HP 20 cylinder diesel engines driving a single propeller shaft and screw through clutches and a reduction gear set. On these three ships, as seen in Figure 1, the layout of the power train includes a large generator rigidly connected to the drive train of the port engine. This 480 volt 3 phase 1740 kw generator provides electrical power to the self-unloading conveyor system and the bow and stern thrusters used in docking and restricted space maneuvering. The after end of the generator shaft is connected to the port pinion gear of the reduction set through a clutch. The generator rotor turns whenever the port engine runs but is not used to generate electrical power unless it is declutched from the reduction gears and operating at synchronous shaft speed of 720 rpm. Thus, the generator rotor acts only as a flywheel during most hours of engine running when traveling between ports. Its use to generate electricity amounts to a very short period of maneuvering time when entering or leaving dock facilities and when unloading its cargo. Normally, unloading occupies a time span of 8 to 12 hours.

The forward end of the generator shaft is rigidly supported by the end of the diesel engine drive shaft. The after end of the generator rotor is supported by a ball bearing with the generator shaft coupled to the input flange of the clutch. On two of the three ships having essentially the same propulsion layout, seen in Figure 1, there have been disastrous failures of the generator bearing. These failures have proved very costly in terms of loss of vessel use and extensive repairs to the generator.

GARD, INC., a subsidiary of GATX was requested to investigate the feasibility of bearing failure prediction to mitigate this costly failure mode. GARD surveyed the field of bearing monitoring consultants and instrument suppliers for the latest and most promising instrument approaches compatible with marine installations. The instrument had to be reliable and simple enough to use by the operating engineers employed on this class of vessel. The selected instrument had the capability of monitoring multiple bearing sites through multiplexing provisions. It was decided to monitor five locations on a trial basis to evaluate both the technique and the instrument. These five locations are shown in Figure 1. They consist of the generator bearing (the prime monitored site) and the four pinion bearings of the reduction gear set. These components are large double row spherical roller bearings operating at engine shaft speeds ranging from approximately 900 rpm down to 400 rpm in speed range. The pinion bearings are force-lubricated by the reduction gear oiler system, while the generator bearing is lubricated by means of a low volume sump at the bottom of the bearing. The sump oil level is frequently monitored and replenished. Maintenance procedures for these vessels include frequent and periodic submission of oil samples to several analysis facilities for reports on contamination and evidence of metal debris. It has been recognized that this is not practical as a quick response feedback on bearing condition since failure can occur well within analysis response time.

The Automatic Shock Pulse Analyzer: The technology and instrument chosen by GARD is the Automatic Shock Pulse Analyzer (ASPA) designed and supplied by the SKF Technology Services Division of SKF Industries Inc.. The Mark IV analyzer shown in Figure 2 is based on the recognized phenomena of bearing defect shock pulse generation but utilizes this information in a more sophisticated statistical approach than several other proponents of this technique. The original use of this concept depended on monitoring only the peak amplitude of these pulses and counting the number that exceeded a selected threshold value over a measured sample duration at a known shaft speed. The SKF approach computes energy content for each shock pulse. All of these individual pulse energy quantities are summed over a fixed sampling period. This technique recognizes that some pulses have different durations than other pulses of a similar peak amplitude and also takes into account the many times that pulses of different peak amplitudes are partially superimposed on each other. The method is microprocessor based and in a very simple form is described in Figure 3.

The shock pulse approach depends on the generation of energy impulses by the impact of balls or rollers on defects on the inner or outer races, or defects in the ball or roller itself, or debris that may be trapped within the bearing assembly. These shock pulses propagate outward in a spherical geometry modified by reflections within the complex mechanical structure itself and can be detected almost anywhere within reasonable range of the origin. A very favorable characteristic of these pulses is that they are in the ultrasonic region of 20 kHz to 500 kHz and are thus well separated from the audible frequency range in which much of the stray machinery noise can be found. The machinery noise energy often masks the audible signals generated by the defective bearing. Thus, shock pulse generation results in a signal that can reliably be discerned as bearing phenomena and permits the investigator to discriminate against ambient machinery noise.

The transducers most commonly used are piezoelectric accelerometer type sensors, appropriately matched to the mechanical impedance of the propagating medium and the electrical characteristics of the shock pulse analyzer. There are several practical techniques for satisfactorily mounting transducers. A threaded stud is used in our application. The shock pulses detected by the transducers excite a damped oscillatory train at frequencies close to the natural resonance of the transducer. The transducers employed resonate at approximately 37 Khz. Figure 3a shows the rolling elements initiating spherical shock waves which excite the sensor to generate damped trains superimposed on the lower frequencies characteristic of machinery vibration mechanisms. A band-pass filter removes the unwanted frequencies leaving only the ultrasonic trains of shock excited transducer ringing. These are then envelope detected to produce a train of the shock pulse envelopes. The shock pulse train is then digitized and the microprocessor in the analyzer computes the total energy content of all pulses which occur during a controlled sampling period.

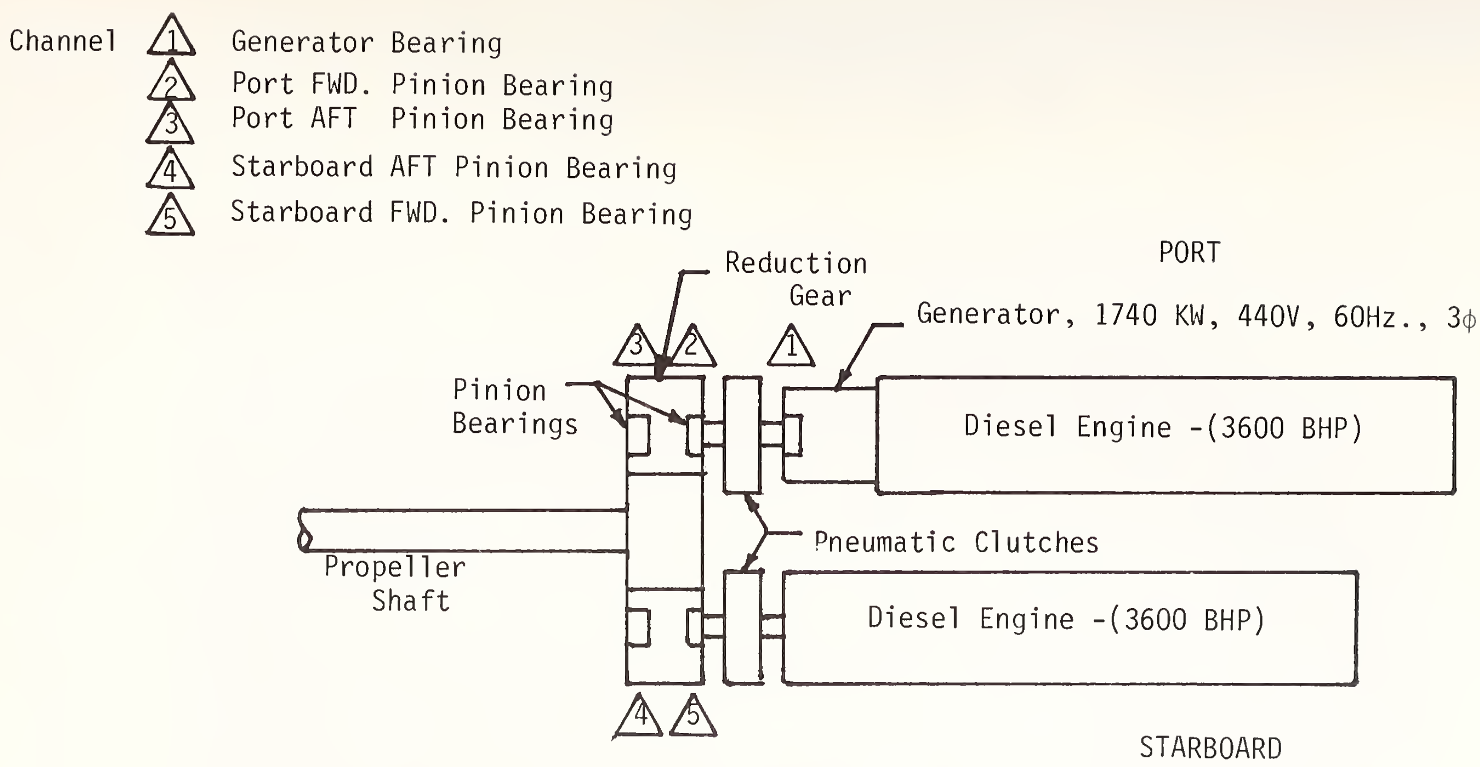
The system, as configured in the Mark IV, also controls the multiplexer switching from one channel to the next and displays the results of each data frame as compared to preset limits for "Caution" or "Replace" warnings. The instrument has logic which calls for operator attention when limits are exceeded. Both visual and audible alarms suitable to the noise and activity environment of the typical ship's engine control room warn the control room attendants of abnormal conditions. Although the Mark IV is capable of monitoring up to 8 groups of 31 channels each, the trial installation by American Steamship Company on the Motor Vessel H. LEE WHITE monitors only the 5 previously mentioned channels. In order to set the threshold levels contained in the instrument's Read Only Memory (ROM), it is necessary to measure typical acceptable bearing performance at specified speeds for a sufficient period of time to accumulate the statistics from which the "Caution" and "Replace" thresholds are computed. These quantities, once stored in the ROM, are the reference to which the warning decisions are made.

Evaluation Program: During the summer and fall of 1979, three ships of the American Steamship Company fleet, having similar propulsion and generator installations were visited. On these ships, transducers were mounted on the generator bearing housing shown in Figure 4 and on the four reduction gear pinion locations as shown in Figure 5. A portable battery powered prototype (the Mark III) was used to record shock pulse energy levels to compare bearings from ship to ship. In addition, a spectrum analyzer accompanied these experiments to define the frequency content of the shock pulse train. This spectral data was then used to fault isolate the specific mechanical source of abnormal shock pulse energy levels.

In the course of this program, the port forward pinion bearing on the MV H. LEE WHITE was found to have significantly higher shock pulse energy numbers than its counterpart on the starboard side or the rear pinion bearings. The bearing was subsequently disassembled and examined. Significant outer race damage was found to be present as shown in Figure 5. This bearing was replaced during the normal winter maintenance season.

In 1980, the Mark IV was installed on the H. LEE WHITE and the crew was trained in making entries to a log. Behavior has been monitored over a period of 12 weeks to date. Normal cruising speed for this vessel is roughly 15 miles per hour (statute miles are used on the Great Lakes) at a propeller shaft speed of 120 rpm (corresponding to an engine crank shaft speed of approximately 900 rpm). Early in the sailing season of 1980, the ship's operation was altered to a cruising propellor shaft speed of approximately 103 rpm to conserve fuel. At the start of the season it was found that the forward starboard pinion bearing was exhibiting a shock pulse energy number 10 to 15 times higher than the portside pinion bearings. Figure 6 shows the results of 7 weeks of data logging of the 5 bearings at speeds of both 120 rpm and 103 rpm. Additionally, the generator bearing is monitored at its synchronous speed of 720 rpm during power generation. Several interesting features can be noted on examining the results. Channels 1, 2, 3, and 4, show very consistent levels during the entire observation period for each propeller speed recorded. This demonstrates the consistency of the instrument and its ability to operate in the temperature, vibration and moisture environment of Great Lakes commercial vessels. Probably the most important aspect of information from this type of monitoring is seen in the slopes of these curves showing the trends of increased wear during the period of observation. This is also holding true for channel 5 at its higher level than the other bearings (at least for the period of observation). At one point, at 103 shaft rpm, its level increased by 35% for a period of 6 hours. The reading then dropped back to its previous level. This shows the sensitivity of the technique to what probably was particulate contamination that entered the bearing and was subsequently flushed out during its operation. This bearing is being closely monitored during the sailing season. Although the high level does not, in this case, indicate a damaged bearing it is probably warning that life for this bearing will be shorter than its neighbors and that its energy trend should be closely monitored. If there is a significant increase in energy trend over a relatively short period of time, this will indicate that this bearing should be replaced at the earliest opportunity to avoid destructive failure.

Conclusion: The shock pulse energy monitoring technique as embodied in the Mark IV Automatic Shock Pulse Analyzer has provided meaningful information on bearing condition. The bearing monitoring installation is still in a period of operational evaluation. Pending the final results of this trial period, the value of additional installations to appropriate fleet machinery will be determined.




PROPULSION TRAIN

Figure 1

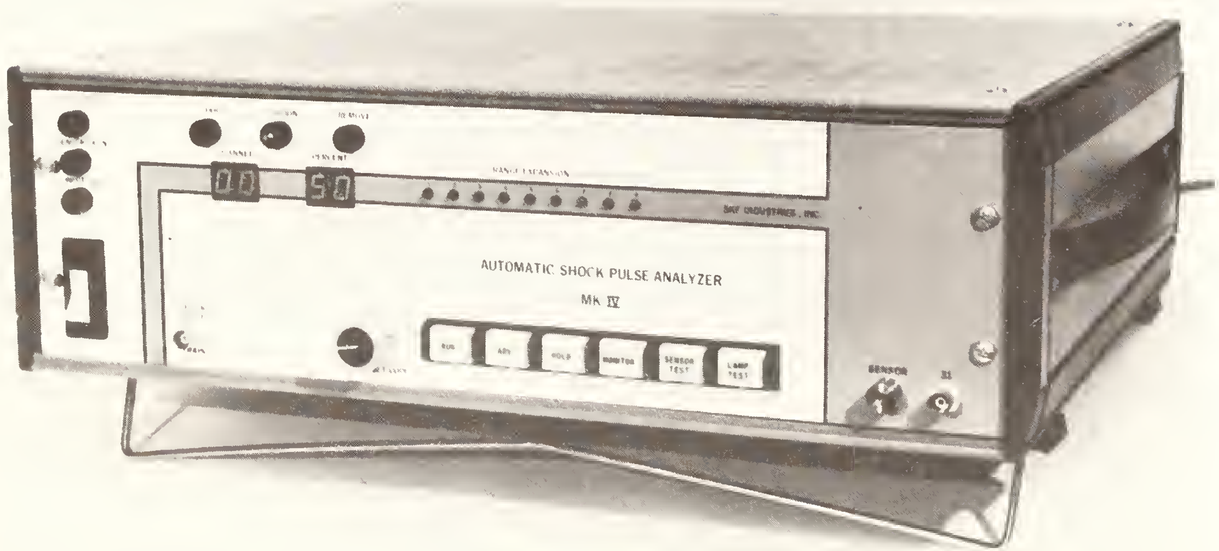



Figure 2 The MARK IV Automatic Shock Pulse Analyzer

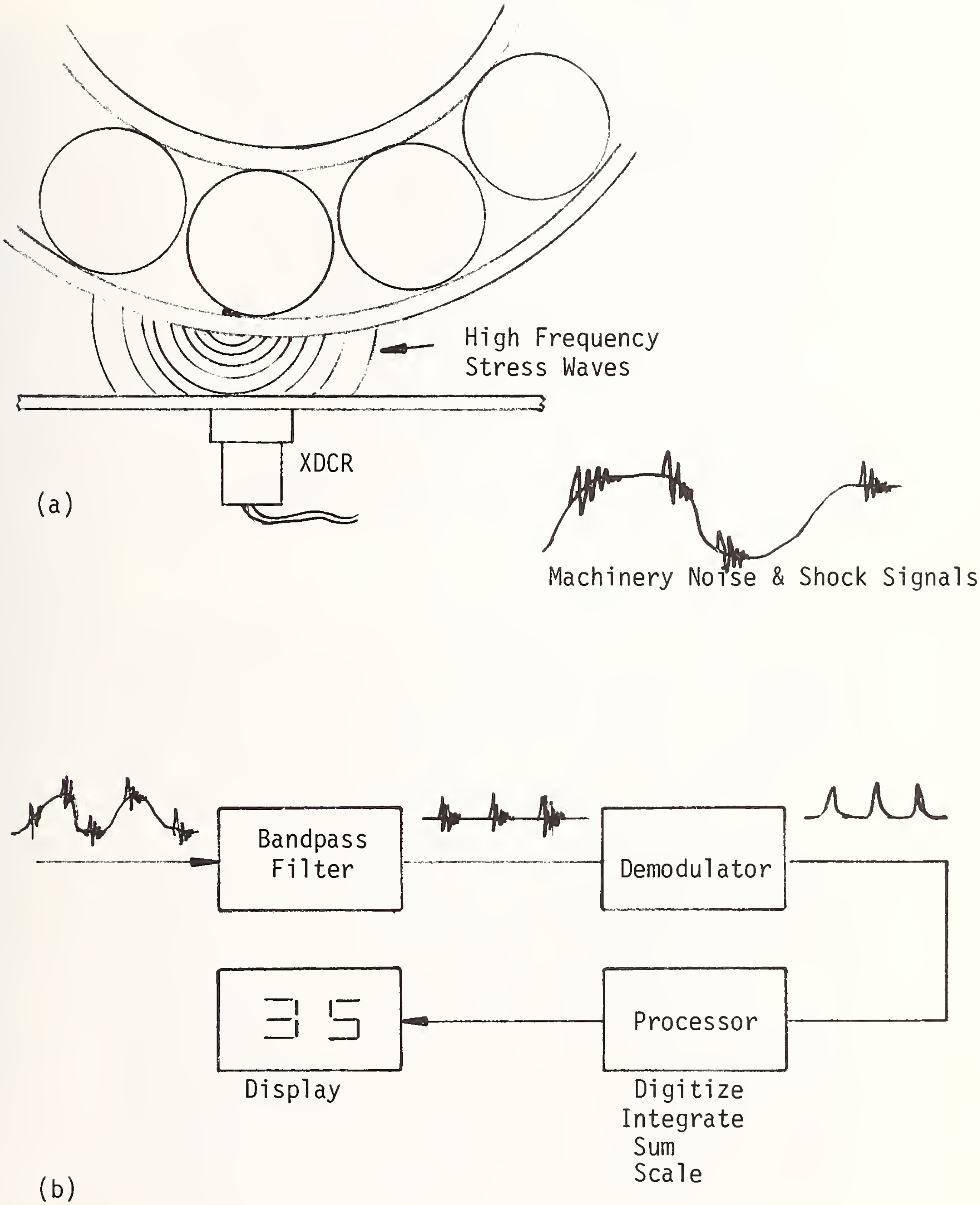



Figure 3 Shock Pulse Analyzer Process

(a) Generator Transducer Installation

(b) Pinion Bearing Transducer Installation

Figure 4

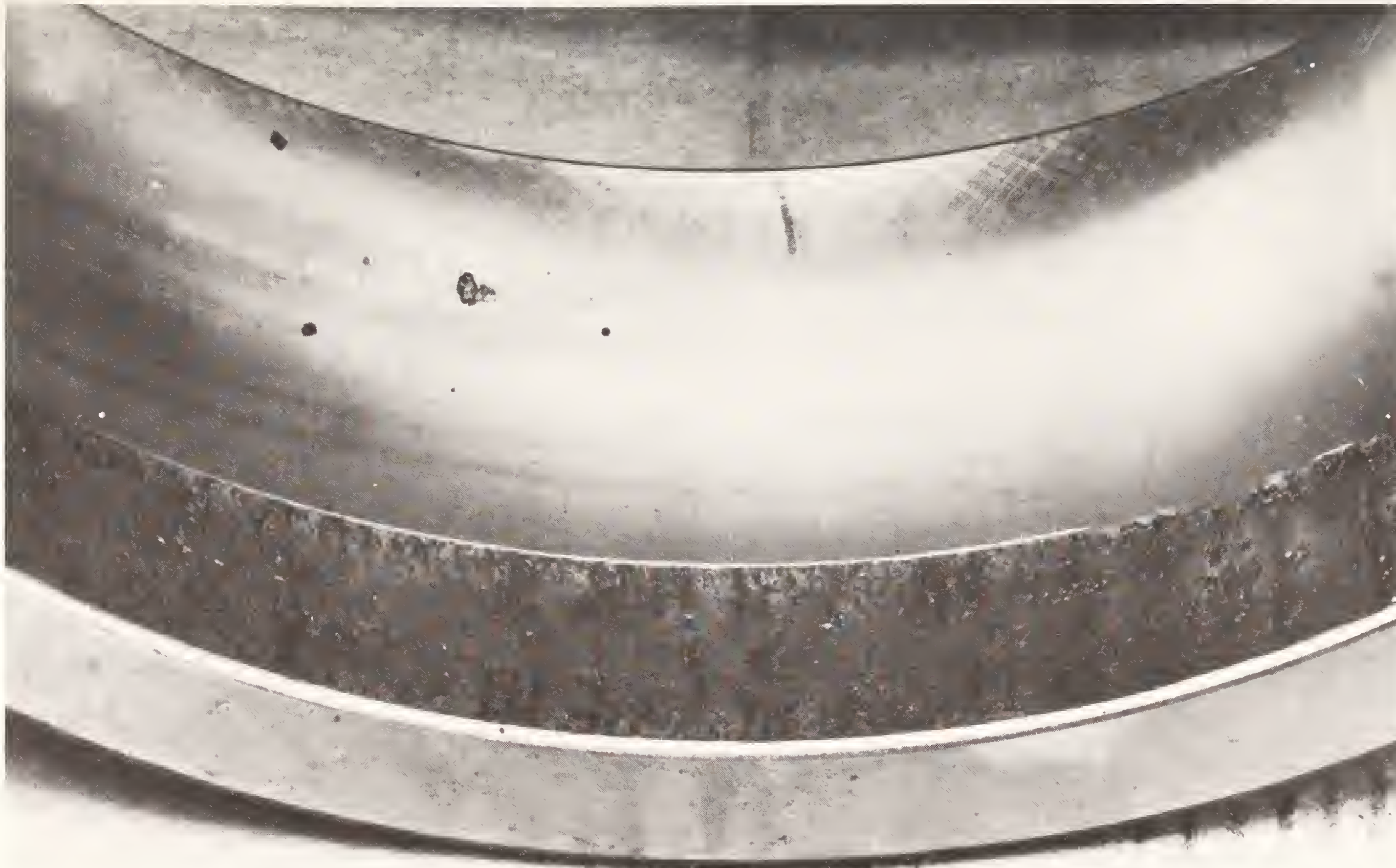


Figure 5 Damaged Port Forward Pinion Bearing

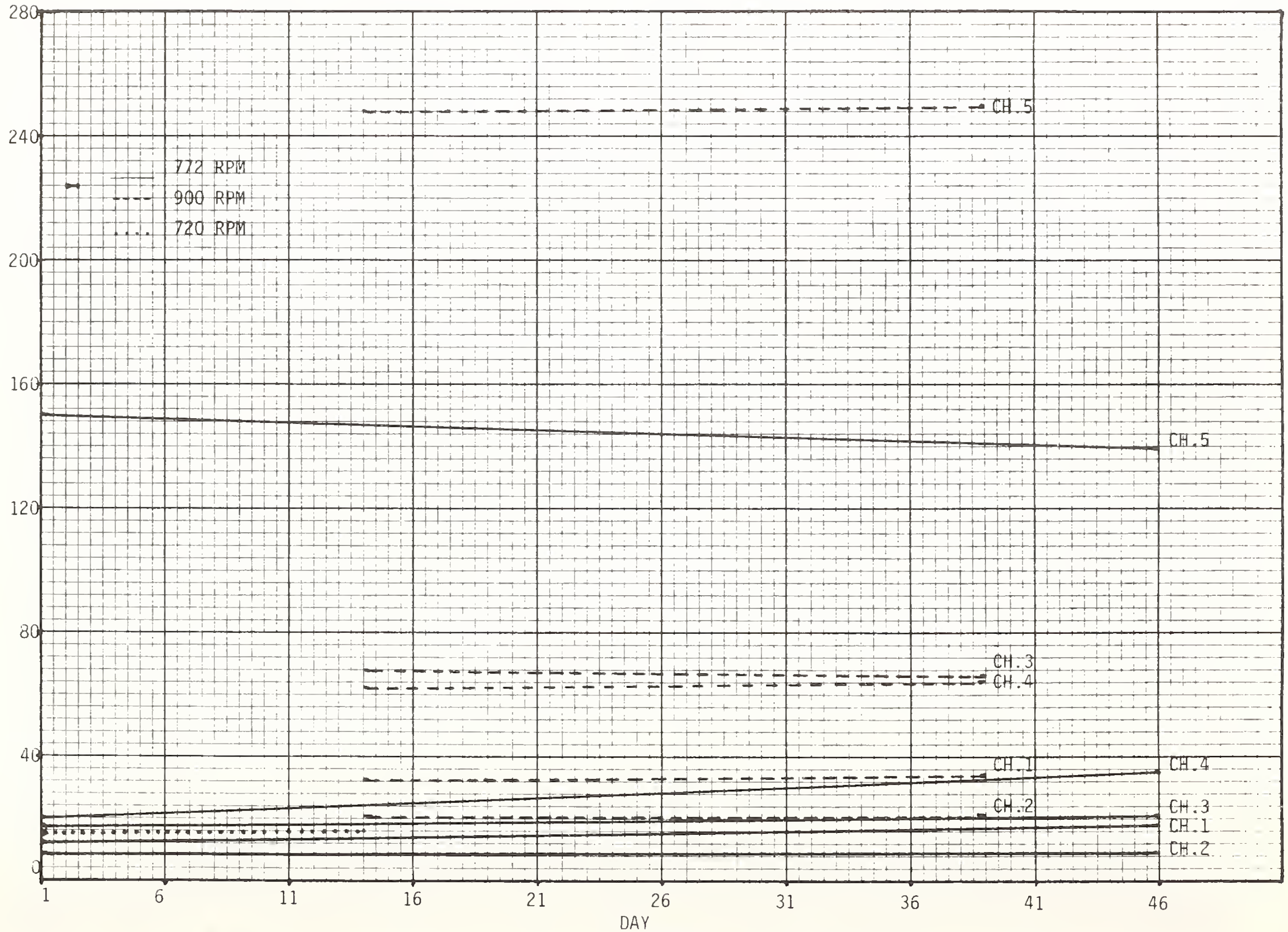


Figure 6 Plot of MARK IV Readings

# SESSION IV

## DIAGNOSTIC AND PROGNOSTIC TECHNIQUES

Chairmen: J. Anderson, Spectral Dynamics

P. Howard, Energy and Minerals Research Company

# CONSERVATION: LOSS PREVENTION AND RISK MANAGEMENT

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Abstract: Many factors are involved in Risk Management and Property Conservation. Loss prevention, predictive-preventive maintenance, cost of money and insurance are just a few of the many. The subject of cost can be further subdivided into real and subjective costs; the real cost of unexpected losses and the subjective (though some feel real) cost of uncertainty. Today, the intelligent management of all factors is of prime importance. With the high costs of downtime and replacement parts, one can no longer rely on previous methods.

In the 1970's some insurance companies which deal in industrial property coverages, i.e., turbine insurance, pressure vessel insurance, compressor insurance, etc., have taken an active interest in the early detection of incipient faults. This interest has led some of these insurance companies to purchase sophisticated electronic devices (Acoustic Emission Analyzers, FFT Spectrum Analyzers, Shock Pulse Analyzers, etc.) and to use these instruments to help the Insureds conserve their assets. Property conservation, loss prevention, and risk management have now combined their collective capabilities to ensure the early detection of faults and maximum life of machines for the most economical cost of these procedures and insurance coverages.

Key words: Acoustic Emission Analysis; Insurance; Loss Prevention; Modal Analysis; Property Conservation; Risk Management; Shock Pulse Analysis; Vibration Analysis.

In today's modern manufacturing and public utility facilities, risk management has taken on a new meaning and importance. With the economic situation being what it is, one must be aware of every possible avenue to save time, money, and property. The plant manager and other management officers must now know the meaning and importance of Risk Management and Loss Prevention.

To begin, we must define "risk management" and its associated elements. In a simple phrase, risk management is the treatment of loss exposures; loss exposures, further, have three basic elements. They are:

1. The item must be subject to a decline or disappearance in value, i.e., subject to loss.
2. The item must be subject to causal forces which may cause a loss.
3. Extensive negative impact from a loss must be a possibility.

## Item Subject to Loss

The term "loss" implies the decline in value or disappearance of an object or property. Be this the case, it is quite important in risk management that a thorough identification and evaluation be made of exactly what is subject to a loss, and to what type of loss it is subject.

Items subject to loss can be simply categorized into (a) assets (things of value owned) and (b) income (provided by these assets).

## Causal Forces

Once we have identified objects subject to loss, we must now deal with what exactly can bring about the adverse effect. We in the risk management and insurance business call such an element a "peril." A peril is a potential cause of a loss. Thus, an adverse event (a peril) may happen and create its subsequent economic loss.

It is quite important to be aware and identify the perils associated with each piece of property in one's care, custody, and control. We must identify what equipment is subject to loss and find out the associated perils. A program of peril identification can do this for us. As stated by Williams, Head, and Glendenning, knowing what exactly is subject to loss and what types of losses it is subject to must clearly be resolved before any method of risk management can be intelligently considered.

Perils may be grouped according to their origin as either natural, human, and/or economic. Natural perils are familiar, such as fire, windstorm, hail, flood, etc. Human perils may be considered as theft, riot, vandalism, negligence, and the failure to satisfy an expressed obligation. Economic perils can be inflation, recession, technological advances, etc. There are, obviously, many overlaps.

Keep in mind that perils are only a portion of loss exposures and that some perils are not insurable. What is of great importance and significance, however, is the problem of peril identification and analysis.

## Potential Economic Impact

This final element of loss exposure is both the quantitative and logical sequel to the other two. One must realize, however, that potential financial impact is a relative rather than an absolute term. What would constitute a severe financial impact to one party would not be as severe to another. Also, the potential impact varies over a period of time for the same party.

## LOSS EXPOSURES AND RISK

In risk management and insurance, the term risk is meant in practice to mean any one of the following:

1. The possibility or state of being exposed to loss.
2. The probability or chance of loss.
3. A peril.
4. A hazard.
5. The property or person exposed to damage or loss.
6. The potential losses that may be sustained.
7. Variation of these potential losses.
8. Uncertainty concerning loss.

Risk as the Possibility of Loss:

According to this usage, one faces the risk or possibility of a certain type of loss. Those who use this definition claim that risk (possibility of loss) cannot be measured. It either exists or it does not exist.

Risk as Chance of Loss:

In this usage, it is the probability or chance of loss. The chance of loss is the relative likelihood that a loss will occur within some stipulated time period. Thus, risk can be measured and chance of loss can be stated to be between 0 and 1, zero maintaining that loss will not occur and 1 that loss is certain.

As an example, if risk is thus defined as the chance of loss, and the risk of a turbine explosion during the next year is estimated to be 1/6, thus this would mean that on the average, one would expect this type of loss to occur once in every six years.

The chance of loss is an important element in pricing insurance, underwriting insurance, and deciding whether to recommend insurance in a particular situation.

Risk as a Peril:

A less common but still frequent practice is to define risk as a cause of loss (a peril). Examples of perils are fires, explosions, etc.

Risk as a Hazard:

A hazard is a condition that creates or increases the chance of loss arising from a particular peril. Hazards, therefore, are the real causes of loss in that they cause the perils that cause the losses. Examples of hazards are carelessness with property that may increase the chance of a certain loss, i.e., poor housekeeping, inadequate ventilation, improper storage of hazardous materials, etc.

Risk as the Uncertainty Concerning Loss:

This may be the most important definition of risk. When the potential of loss varies, or is believed to vary, a person is uncertain or doubtful about his ability to predict the outcome of events. This definition then equates risk with a person's conscious awareness of a variation in what might happen. Because two persons may have different information concerning the potential losses and may interpret that information differently, their uncertainty can vary greatly even though they are exposed to the same potential losses. An example is the person who believes that only a few outcomes (differing very little from each other) can occur. Another person facing the same facts and situations may conclude that widely varying outcomes can occur. One may be right or both may be wrong. The first person believes he or she can predict the future fairly well; the risk is small. The second has little faith in his or her forecasting ability and the risk is great. This doubt concerning their ability to predict the future is certainly an important concept because it influences many risk management and insurance decisions. Uncertainty is the key word in the economic impacts of risk.

## ECONOMIC COSTS OF RISK

A. H. Willet, in his discussion of the economics of insurance, refers to the cost of uncertainty arising out of:

1. The unexpected losses that do occur,
2. The uncertainty itself, even if there are no losses.

We all know the financial impacts of the cost of unexpected losses. When unexpected losses occur, production ceases and the business flow is interrupted. The economic consequences are both direct and indirect. The cost of repair or replacement of the damaged object can be considerable and the fact that the end product is no longer available for sale also causes a cashflow problem. In certain instances, valuable customers may be lost and never regained.

The cost of uncertainty itself, however, can be considerable. The first cost of uncertainty is the tendency to reduce the total satisfaction associated with a given economic status. This reduction may result from:

a) The diminishing marginal utility
b) Overestimating the chance of loss, and
c) Fear and worry

Another cause of uncertainty is its tendency to cause inefficiencies in the utilization of existing capital and to retard development of new capital. As a result, total production or potential of production is decreased. To explain, let's assume that there is a given fixed amount of capital. With other things being equal, the marginal productivity of this available capital in each industry tends to decrease as the amount of capital committed to that industry increases. Thus, this available capital is appropriated in an optimum way when the marginal productivity in each industry is the same. If this were not so, total production could be increased by transferring capital from an industry where the marginal productivity is least to one where it is greatest. Uncertainty disturbs this balance because relatively too many resources tend to flow into the safe industries and society may have to live without the products that could be developed from a risky industry. For example, could utilities have attained their present productive status if each individual utility had to bear the monetary losses due to accidents to their larger and larger turbo-generators? I think not. As stated by C. Arthur Williams, Jr., and Richard M. Heins, in their book Risk Management and Insurance, "Within an industry, the existence of uncertainty may affect the apportionment of capital among firms; within a firm, it may affect the selection of the methods of production and distribution."

To summarize, the cost of uncertainty, in addition to the unexpected actual losses, are generally

a) a reduction in the total satisfaction because of the existence of uncertainty and,
b) Less than optimum production, price levels, and price structures.

Five Steps in the Risk Management Process

1. Procedures and communications should be established throughout the organization to allow for complete inventory and discovery of the potential (pure) risks that may arise in the activities of their business firm.
2. After identification of risks, the next important step is the proper measurement of the losses associated with these risks.
3. Once the risk is identified and measured, the various alternative solutions of risk management should be considered and a decision made with respect to the best combination of solutions to be used in attacking the problem.
4. After deciding among the alternative methods of risk treatment, the risk administrator and appropriate management groups should establish means for effective implementation of the decisions made.
5. The results of the decisions made and implemented in the first four steps must be monitored to evaluate the wisdom of these decisions and to determine whether changing conditions should suggest different solutions.

Step No. 3 in the risk management process, meaning the identification and measurement of the various alternative solutions of a risk management problem, offers a number of various alternative solutions. Once the risk is identified and the amount of risk evaluated, we must decide how to live with this risk. We can decide to avoid the risk. If we do this, the amount of risk is reduced to zero. If one does not engage in an activity that has the associated risk, the risk does not exist. Thus, if a company decides not to manufacture an item, and does not purchase the equipment that is needed for the manufacture of that item, the risk is thus avoided.

Another way of attacking a problem is to transfer the risk to some other party, that is, to buy insurance. When insurance is purchased, part of the risk and financial burden is transferred to the insurer. Except for a certain deductible amount, the risk has been transferred. Still another way of dealing with this problem is to retain or bear the risk internally. In this case, if a loss occurs, the company will bear the financial loss itself. This is similar in consequences to ignoring the risk. The final alternative is to reduce the chance of loss or to reduce its magnitude if it does occur. This, of course, is the fundamental premise of loss prevention.

Loss Prevention and Incipient Fault Detection

After the risk has been identified and measured, and the appropriate steps to deal with it have been accomplished, one still has to deal with the fact that a machine will, in most probability, fail sometime in the future. If insurance has been purchased, or if the risk has been partially or wholly retained by the owner, it is quite important that some sort of loss prevention system be implemented.

Property Conservation, Loss Prevention, and Predictive Analysis

In times of soft economies, as well as in times of fierce competition, it is historical for corporate management to worry more about the marketplace and production than about loss prevention. Concern is frequently placed on research and development for new products and the possible penetration into untouched markets. Loss prevention (including predictive analysis) falls by the wayside, for after all, there is insurance coverage on the plant and machines. There may even be Business Interruption Insurance to cover lost profits for the downtime suffered. But there is no insurance that will cover the situation in which one of your salesman has to say, "we cannot deliver...our production ceased due to the destruction of our turbine (or whatever)."

Property conservation and loss prevention are closely tied to many factors, including your company's reputation or image in the business community, efficiency, employee morale, and most importantly, to its ability to produce and deliver its product day after day, consistently. A company's customers may sympathize with the problem; but they have an obligation in turn, the same as yours, to deliver their product.

As equipment and material costs continue to escalate and profit margins decrease, plant management efforts must now be directed toward methods that will ensure continued and uninterrupted production from their facilities. The same risk management criteria and procedures which were used by a corporate risk manager must also be utilized by the plant engineer. For a "quiet night's sleep" and peace of mind, he must weigh the various risks involved and the methods to deal with these risks. Predictive analysis is one way to deal with these risks.

Because of today's economic conditions and the high cost of everything involved in production and repair, it is virtually impossible to charge enough insurance premium to cover all losses. In many cases, small items costing \$10,000 five years ago costs over \$35,000 today. One way of dealing with this situation is for an insurance company to increase the deductible amount applied to the policy. Thus, the insurance carrier has transferred back to the

Policyholder part of the "risk." In the past, a manufacturing company may have been able to handle a number of $10,000 deductible nuisance losses, but it is not financially prepared today to handle the same number of $25,000 or even $30,000 losses. The deductibles today are, in many cases, in the six figures, which indicates the importance of having to evaluate the risks involved and develop methods with which to deal with them effectively.

It is both unproductive and wasteful to schedule a machine for internal inspection and dismantle for no reason other than "it's time to tear it apart." Likewise, it is inefficient to "run it until it breaks." Both are present philosophies of some large industrial corporations. What is needed is a blend of scientific procedures and criteria which would allow a method of determining machinery health in order to best utilize available resources to maximum benefit. It is a waste of precious oil to change it every 4,000 miles, on the other hand, it is also a waste of a good machine to let it run without an oil change.

Modern electronics and creative engineers have given us some new tools with which we can work. Among the fairly new analysis tools are "real-time spectrum analyzers," "acoustic emission analyzers," "high-frequency shock pulse analyzers," and "modal analysis systems."

Vibration analysis has been around for some years, basically since the days of the hand-held vibration meters with tunable filters. They were good at detecting unbalance and misalignment. Since then, however, the state of the art has escalated almost exponentially with the advent of the Fast Fourier (FFT) Real-Time Analyzer (RTA). The FFT RTA now allows us to peek at the more esoteric frequencies and nuances of machine vibration in order to figure out "what's going on." Figure 1 shows the setup using such a device to analyze the vibrations emitting from a high-horsepower, multi-stage centrifugal compressor used as the main incoming air supply at an air reduction plant. The loss of this compressor would not only be a large loss for the insurance carrier, it would also be a burden for the company itself; it must supply another plant which is dependent upon the various gases produced via pipeline.

In many cases, a replacement bullgear for such a compressor has over a three-month lead time. Many dollars and much "image" would go down the drain if such a loss were suffered.

Figure 2 is the spectrum of a steam turbine's HP-IP bearing in which was discovered a very large 600 Hz spike. All other similar machines had no such indication, let alone one of such high magnitude. This was brought to the attention of the plant personnel, who simply shrugged it off as being odd, but nothing more. Thirty-two days later the turbine tore apart, with a subsequent and very expensive loss. It was a classic case of not wanting or caring enough to find out what the indication meant. It was later determined by metallurgical analysis that there was some cracking; the spike may have been an early indication of such impending failure. In any case, the other machines had no such indication.

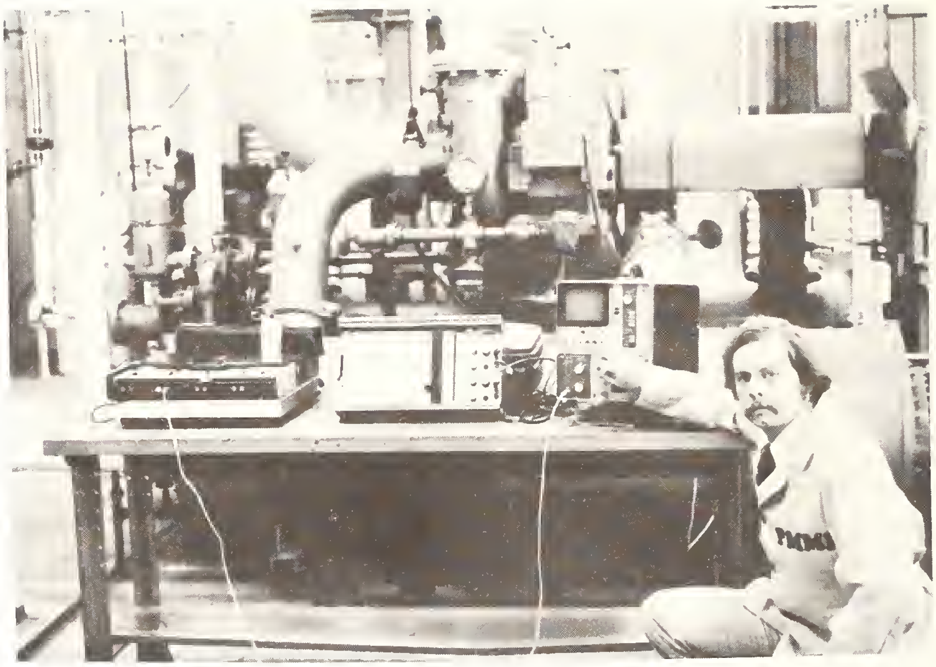

Figure 1. Author taking data for vibration analysis of a high speed centrifugal compressor vibration.

Shock Pulse Analysis is a fine method of determining very high frequency components of mechanical vibrations. It operates on the principle that certain discrepant parts within a machine (such as pitted, spalled, or brinelled bearing and gear elements) release abnormal amounts of frictional and/or kinetic energy. In the frequency domain, frictional energy generates "white noise." The kinetic event, for the most part, is lost in the noise and machine vibration. It so happens, however, that above a certain frequency the amplitude of kinetic energy release will be greater than that of the machine vibration. It is difficult, if not impossible to detect characteristic kinetic events using conventional vibration analysis techniques, but with the use of envelope detection, the natural resonance of certain types of accelerometers and subsequent FFT spectrum analysis, these important characteristic kinetic events can be "retrieved."

Figure 3 details the procedure for analysis of these kinetic events. It is interesting to note, that as with standard low-frequency vibration, the data can be used for either incipient fault detection and analysis or for machine monitoring. The shock pulse train can be sent to a spectrum analyzer or to a shock pulse monitor. In any case, a new vista of machinery health monitoring has been allowed by the use of imaginative engineering and state-of-the-art electronics.

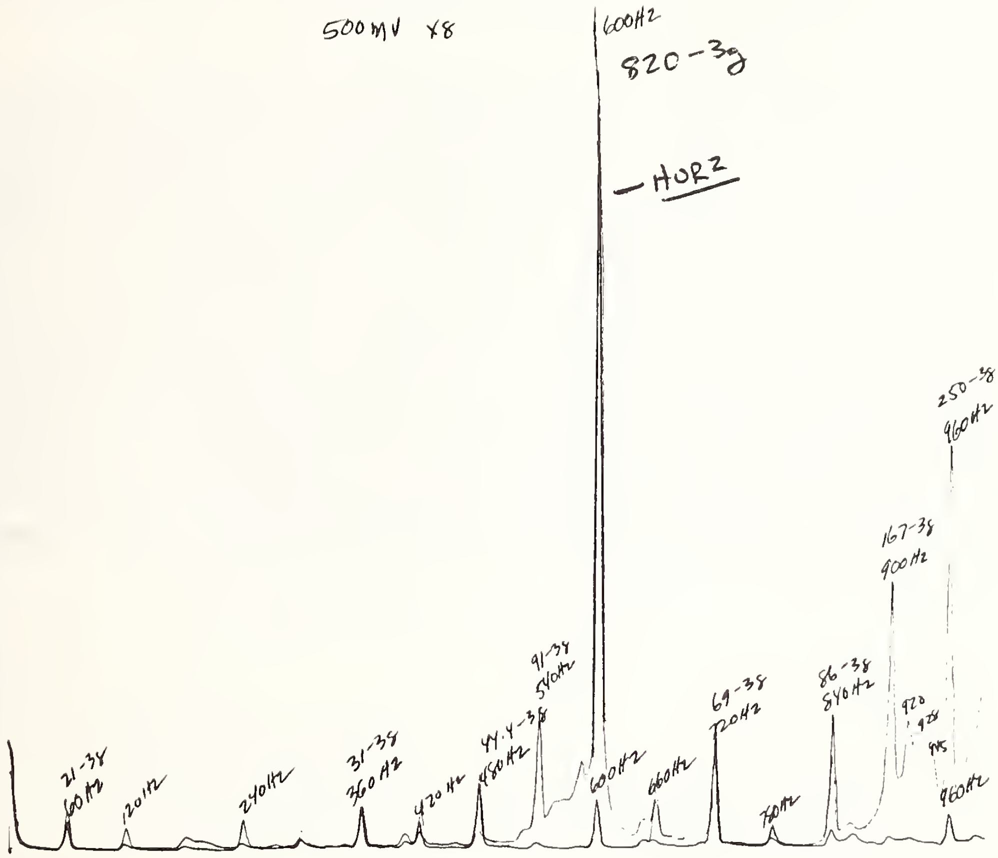


Figure 2. Signature from a bearing of a steam turbine; unit later failed.

Acoustic Emission Analysis is yet another advanced method for incipient fault detection. Similar in some ways to both vibration and shock, it studies the acoustic kinetic events produced by a material under stress. Acoustic Emission (AE) measurements are specifically intended to reveal very small (thus early stage) incipient faults. Unlike vibration, these AE measurements relate directly to sound energy emitted in a material and not to the machine component behavior. Even though most machines are quite noisy, this operating noise tends to be in the same part of the frequency spectrum as machine component vibrations. AE analysis studies that portion of the spectrum in which defect-originated energy extends itself; even higher than that of shock pulse frequencies. By definition, acoustic emission is a transient elastic wave that has been generated within a material by the very rapid release of energy. AE is quite sensitive and can detect quick displacements of as little as $10^{-12}$ inch. Many AE sensors have a natural resonance in the 300 kHz to 500 kHz range, and sensor response and types can be selected to fit the particular problem at hand.

AE techniques have been found to be very useful for incipient fault detection in areas where other techniques are weak in comparison. Typical applications are the location of underground leaks in gas transmission lines, fault detection and of monitoring of pressure vessels, and the testing of composite material integrity. Many industries are finding a useful tool in AE analysis including utilities, chemical plants, aerospace, railways, mining and metal fabrication.

The basis for this technique is that materials undergoing deformation, for one reason or another, normally emit acoustic energy. This energy is usually in the form of short bursts or trains of fast impulses in the ultrasonic frequency range. AE analysis can study these internal releases of energy by the use of transducers to detect these transient elastic waves. These signals can then be related to the physical integrity of the material or structure in which they were generated. Monitoring of these events permits the detection and location of flaws, as well as the prediction of impending failure. See Figure 4.

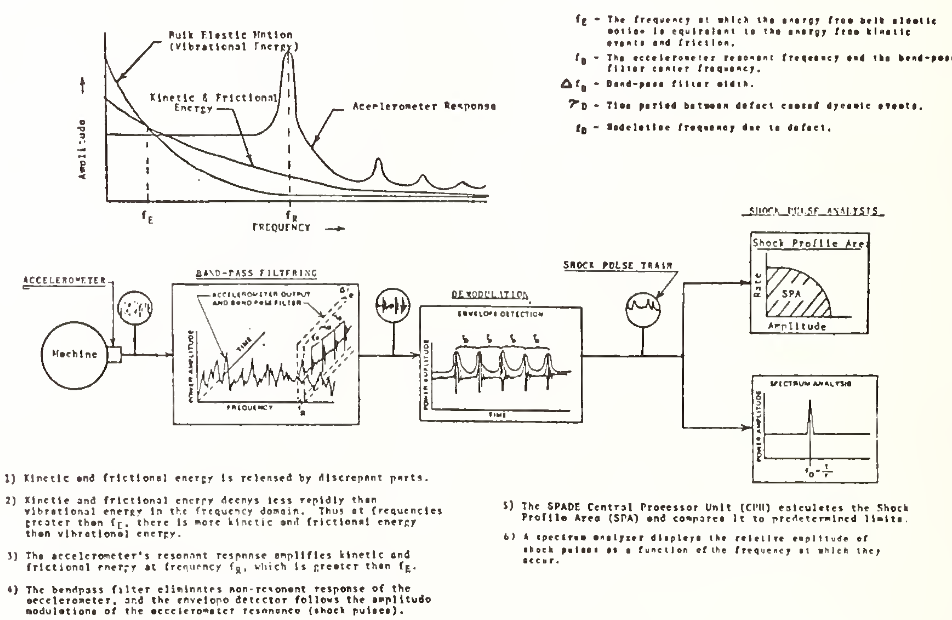


Figure 3. Shock pulse flow chart.

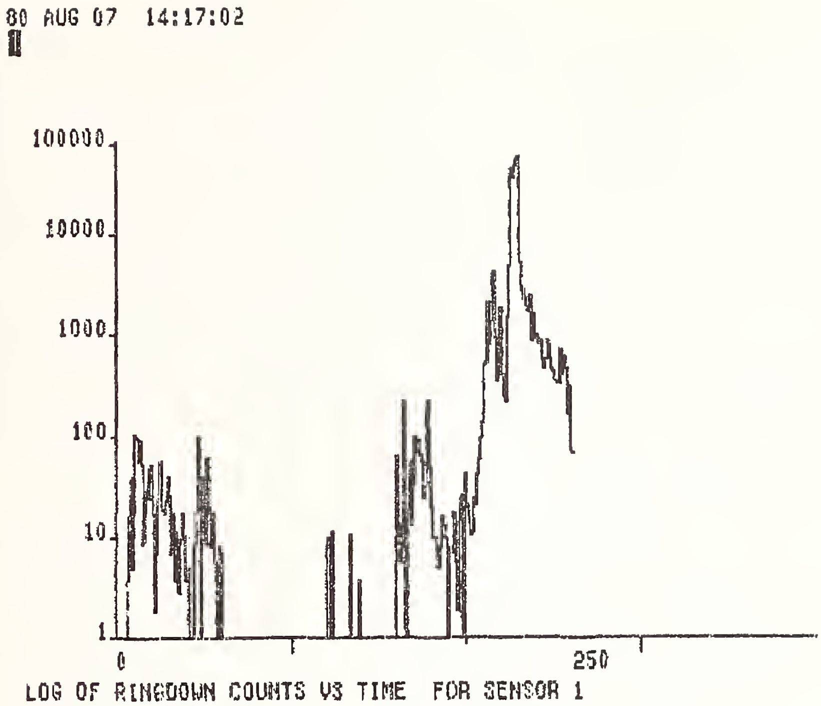


Figure 4.

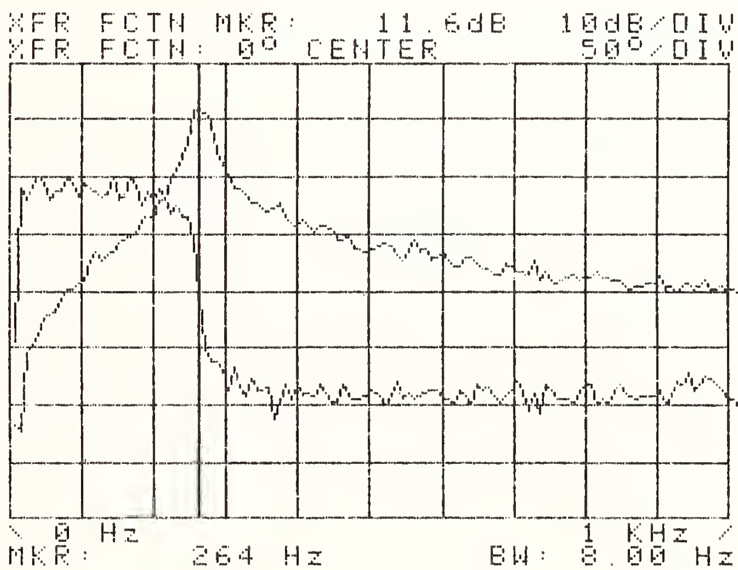


Figure 5. Typical display on a RTA of a transfer function. Resonance at 264 Hz.

An advance two-channel FFT spectrum analyzer allows the investigation of some more esoteric vibration-related information. One such parameter is called "the transfer function." For any machine, the harmonic part of the spectrum due to running speed is usually known in its frequency content. The non-harmonic part, that is frequencies unrelated to running speed, are unknown (both in frequency and amplitude content). This non-harmonic data has a wealth of information which is unrelated to running speed, though it may be excited by it. In order to retrieve this data, Transfer Function Analysis is employed. Transfer Function Analysis is done by exciting an object by external means, usually a shaker or calibrated force hammer. The recorded response is a measure of the structural characteristics of the machine. The response is actually the complex ratio of the cross spectrum to the input power spectrum, which defines the gain and phase lag introduced by a transmission system excited by the known force and responding with a measured result.

Figure 5 is a typical recorded transfer function. One can see that the system resonance is at 264 Hz by the peak at that frequency as well as the associated phase shift. Many times a machine has caused major problems because of unknown component resonance. This type of analysis allows the detection of such problems. Taking a number of transfer functions allows us the ability to do a Modal Analysis using an advanced two-channel spectrum analyzer with modal analysis capability. We can express the vibration of any structure as a sum of its vibration modes and thus represent any vibration as a sum of much simpler vibration modes. Modal Analysis can determine the shape and magnitude of the structural deformation in each vibration mode; and once these are known, it usually becomes apparent how to change the overall vibration.

Conclusion:

Predictive analysis is as important a part to a company's welfare as predictive medicine is to one's health. Just as periodic checks of one's heart, lungs, and other vital organs of one's body are made to prevent breakdowns and ensure better health and longer life...the protection of the vital organs of a business, the operating equipment and plant facilities, is also very essential.

To ensure against broken production schedules, maintenance stock shortages, unnecessary deterioration of equipment, and to prolong the life of operating machinery, regular inspection and service of equipment are major considerations of any business. Top performance of equipment for a healthy business is a direct result of creative and predictive analysis.

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# RECENT EXPERIENCES WITH STEAM TURBINE DISC CRACKING

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Abstract: Steam turbine disc cracking in large power generation turbines was essentially undetected until recently when utilities and turbine manufacturers started performing extensive nondestructive examination (NDE) programs. These were initiated in part due to the severe disc cracking problems experienced by British utilities during the 1970's. NDE being performed in the United States revealed significant disc bore and disc keyway cracking and erosion cutting previously undetected. Disc rim cracking in the blade attachment area also was experienced in some cases. The paper describes some typical areas where disc cracking has occurred and the NDE methods usually used to inspect for these defects. Metallurgical analysis work performed on several of the disc cracks shows intergranular stress corrosion cracking as the mode of failure. Repair methods presently employed to return the turbines to service and research programs under way to study the problem are also discussed.

Key words: Low alloy steels; steam turbine component failure; stress corrosion cracking; United States experiences; ultrasonic inspection.

## 1.0 Introduction

The U.S. electric utilities have been experiencing a significant increase in the number of cracking incidents in forged, low alloy steel discs used on low pressure steam turbine rotors. Since mid 1979, over 25 cases of disc cracking have been reported. In most of these cases, multiple disc cracks were found, indicating that the extent of the problem is very severe.

The increase in the discovery of disc cracking is due partially to the undertaking of extensive nondestructive examination (NDE) programs by the utilities with the assistance of the turbine manufacturers. The NDE programs were developed to inspect for disc keyway and disc bore cracks on U.S. turbines after the British electric utilities found extensive cracking in 124 low pressure turbine discs during the early 1970s[1]. In a few U.S. cases, NDE inspection found cracks that were close to the calculated critical crack size, and thus catastrophic disc bursts were prevented. However, in one unit which was uninspectable due to disc geometry access limitations, a disc burst did occur that resulted in extensive damage to the turbine.

In all cases where a failure analysis was performed, the cause of disc cracking was stress corrosion cracking (SCC). However, not all disc cracks have been submitted for metallurgical analysis since some ultrasonic (UT) indications were judged to be small enough to allow the turbine to be returned to service. In these cases, a reinspection is planned for the near future.

Because of the serious nature of the stress-corrosion caused disc problems, the electric utilities have continued their UT inspection programs and have initiated a massive research effort for the development of preventive measures through their research organization, the Electric Power Research Institute (EPRI). The utilities' goals are to prevent serious damage to their plants and harm to their personnel, while at the same time, to achieve the maximum productivity from equipment. For example, a unit outage day for inspection can cost up to $500,000, while a unit outage for repair from a disc failure can result in equipment replacement costs from two to six million dollars, in addition to removing the plant from service for up to one year. Techniques for disc cracking detection, diagnosis, and prognosis (DD&P) have been introduced and applied to prevent mechanical failures, but improvements in all three areas are desired to optimize inspection costs and to validate run/retire decisions.

In addition to the utilities' activities, the turbine manufacturers, the equipment insurance companies, and the Nuclear Regulatory Commission (NRC) have initiated their own fact gathering and data analysis programs. One result of this intense effort to determine the scope of the disc cracking problem is that a considerable amount of information has been generated from the crack analyses and research programs conducted so far. While all of this information is not presently available for public disclosure, the remainder of this paper will present information which is available and inform the reader from where future information will be coming.

## 2.0 Background

A steam turbine disc is a rotor component which connects the torque-producing blades to the torque-transmitting shaft. This disc is sometimes referred to as a wheel. A cross section of a low pressure steam turbine is shown in Figure 1. The typical locations where disc cracking has occurred for shrunk on designs, as shown in the figure, are disc rim/blade attachments, the disc external web and hub surfaces, and the disc internal bore and keyway surfaces. In integral rotor designs, where bore and keyways are not present, cracking has occurred on the disc rim, web, and hub surfaces. There are from ten to sixteen discs per rotor and one to three rotors per unit. A large 800 MW unit might have up to 48 individual discs. While cracking has been found on all disc surfaces, keyway and bore cracks have predominated and are more serious, since they are harder to detect before reaching critical size.

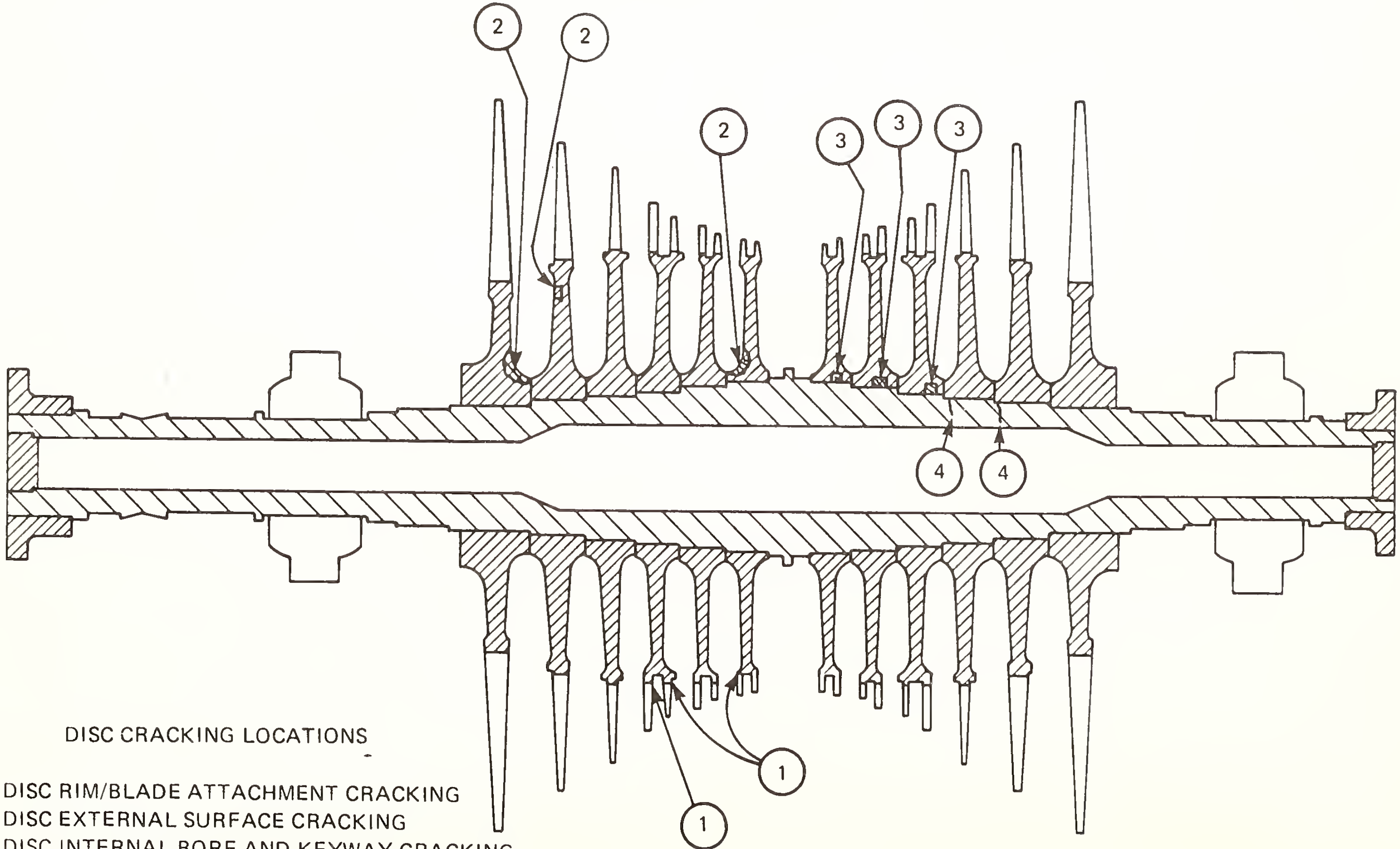


FIGURE 1. LOW PRESSURE TURBINE CONFIGURATION AND DISC CRACKING LOCATIONS

For built-up rotor designs with shrunk-on discs, the materials presently used for discs are 3.5 percent NiCrMoV alloys produced in accordance with a manufacturer's proprietary specifications which are similar to ASTM A471. The materials presently used for integral rotor designs are also 3.5 percent NiCrMoV alloys similar to ASTM A470. Figures 2 and 3 show that the specified mechanical properties and the material chemical composition for a Class 6 A471 disc steel and a Class 5 A470 rotor steel. Chemical composition of either steel is essentially the same. The A471 steel is generally produced at a higher strength level. Other steels sometimes used previously for these components include 2.5 percent NiCrMoV, 3 percent CrMoV and 3 percent Ni, although the last steel has not been used in the U.S.

Figure 4 shows the areas of a disc where cracking has occurred. The keyway and bore cracks have been radial-axial in extent and orientation as shown in Figure 5. The disc hub and web cracks have been both radial-axial and radial-circumferential. For disc rim/blade attachment cracks, orientation depends very much on the blade root configuration. Figure 6 shows the typical cracks found in each of the root designs used by U.S. manufacturers. The axial entry fir tree-type root has experienced both axial-circumferential Type 1 cracks and axial-radial Type II cracks. The Type I cracks are

MECHANICAL PROPERTIES

| | ASTM A 470 CLASS 5 STEEL | ASTM A 471 CLASS 6 STEEL |
|---|---|---|
| PROPERTY | SPECIFIED | SPECIFIED |
| UTS (KSI) | 90 MIN. | 140 MIN. |
| YS (KSI) | 70 (0.02%) MIN.<br>75 (0.2%) MIN. | 125 TO 145 (0.02%)<br>130 TO ±%0 (0.2%) |
| % EL. (2 IN.) | 18 MIN. | 15 MIN. |
| % RA | 50 MIN. | 43 MIN. |

FIGURE 2. MECHANICAL PROPERTIES FOR TURBINE DISC AND ROTOR STEELS

## CHEMICAL COMPOSITION

| ELEMENT | ASTM A470 -- CLASS 5 VACUUM-TREATED CARBON AND ALLOY STEEL FORGINGS FOR TURBINE ROTORS AND SHAFTS SPECIFIED | ASTM A471 -- CLASS 6 VACUUM-TREATED CARBON AND ALLOY STEEL FORGINGS FOR TURBINE ROTOR DISCS AND WHEELS SPECIFIED |
|---|---|---|
| C | 0.28 MAX | 0.40 MAX |
| Mn | 0.20-0.60 | 0.70 MAX |
| P | 0.015 MAX | 0.015 MAX |
| S | 0.018 MAX | 0.015 MAX |
| Si | 0.15-0.30* | 0.15-0.35* |
| Ni | 3.25-4.00 | 2.00-4.00 |
| Cr | 1.25-2.00 | 0.75-2.00 |
| Mo | 0.25-0.60 | 0.20-0.70 |
| V | 0.05-0.15 | 0.05 MIN |

* 0.10 MAX IF VACUUM-DEOXIDIZED.

FIGURE 3. CHEMICAL COMPOSITION OF TURBINE DISC AND ROTOR STEELS

confined to the serrations or steeples and propagate in the tangential direction relative to the disc rim. Type II cracks propagate from the innermost serration in the radial direction.

For notch entry, dovetail-type and T-slot type blade attachment designs, the cracks propagate axially and circumferentially.

Figure 7 shows the type of magnetic particle indication produced on a fir tree-type disc rim. The axial-circumferential type cracks have resulted in blade lifting and blade loss from fracture around the disc rim. Radial-axial cracks have resulted in complete fracture and disc burst as shown in Figure 8.

Another disc degradation problem has been found on non-reheat, shrunk-on discs of one design. In these cases one or two grooves have been found in the disc hub and keyway in the axial-radial direction. The cause of the grooves has been attributed to water cutting. No metallurgical analyses have been performed since the affected discs are still in service, but the grooving mechanism is thought to be corrosion-erosion and not stress corrosion.

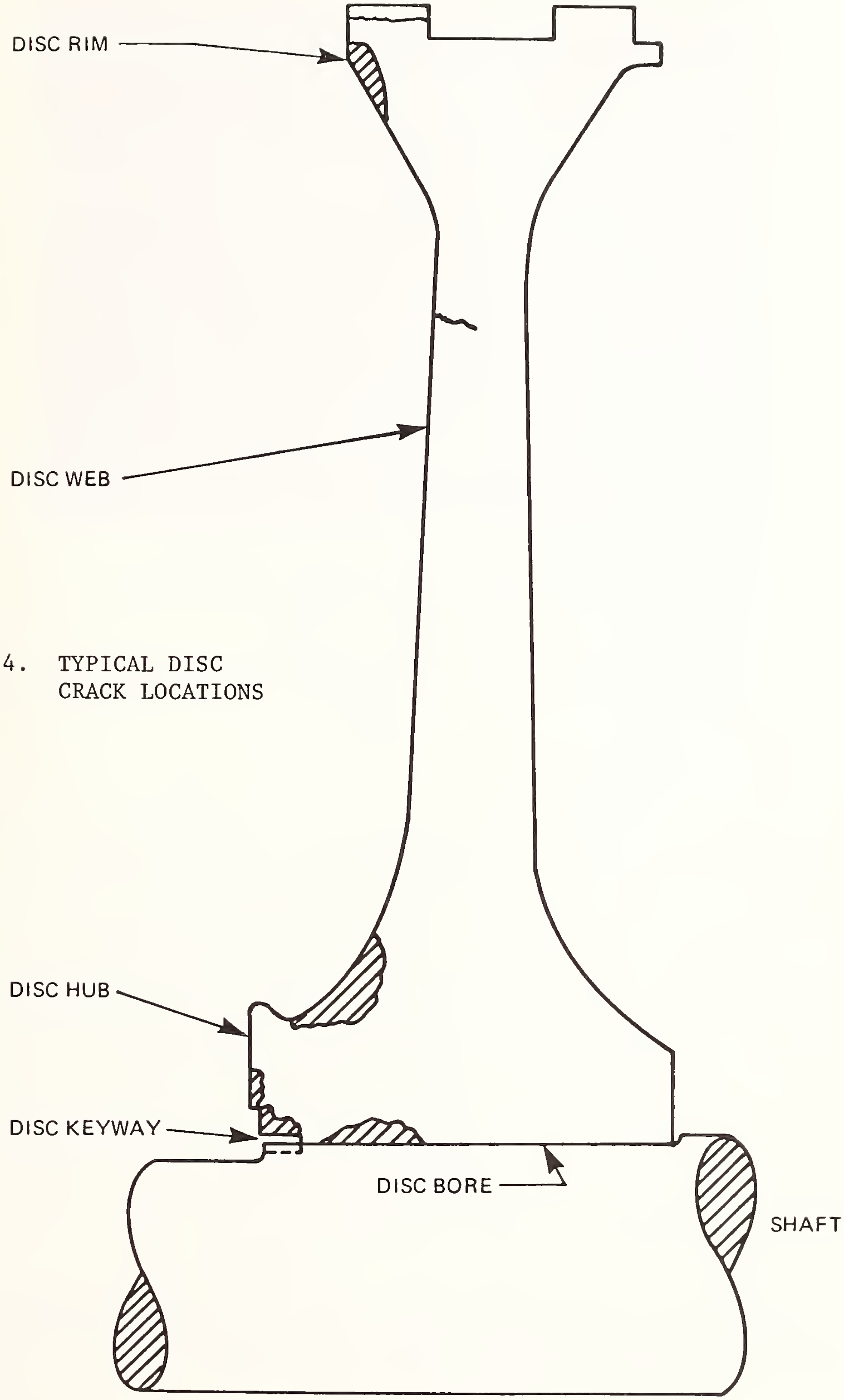


FIGURE 4. TYPICAL DISC
CRACK LOCATIONS

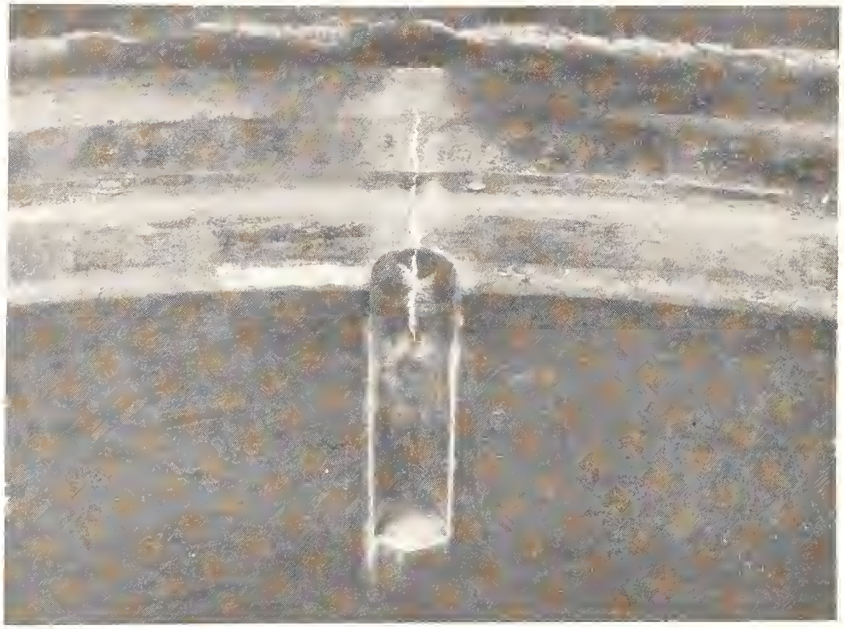

FIGURE 5. MAGNETIC PARTICLE INDICATIONS OF DISK KEYWAY CRACK

A few cases of shaft cracking have been experienced also in which corrosion fatigue cracks initiated on the shaft surface between the shrunk-on discs.

## 3.0 British Experience

The disc cracking problems in the U.S. were properly forecast by those in the industry who were following the British disc inspection and repair program in the early 1970s. After a catastrophic failure of a low pressure steam turbine disc at Hinkley Point A in 1969[2], the British initiated a program to prevent any further disc burst incidents[3]. The Hinkley Point A failure was determined to be caused by stress corrosion cracks in a keyway. The Central Electric Generating Board (CEGB) and the South of Scotland Electricity Board undertook the inspection of 810 discs from 102 turbine rotors to find cracks in these discs, and to determine the cause of the cracking.

A total of 124 discs on 50 rotors were found to have stress corrosion cracks in their keyways and/or bores. Table 1 presents a summary of the British experience[1]. The different type rotors represent a system where the British catalogued the rotors by blade design and keyway shape.

The British disc rehabilitation program was supported by a laboratory research program and both efforts were documented[4] and revealed significant findings. Several of these findings correlate well to the recent U.S. experience.

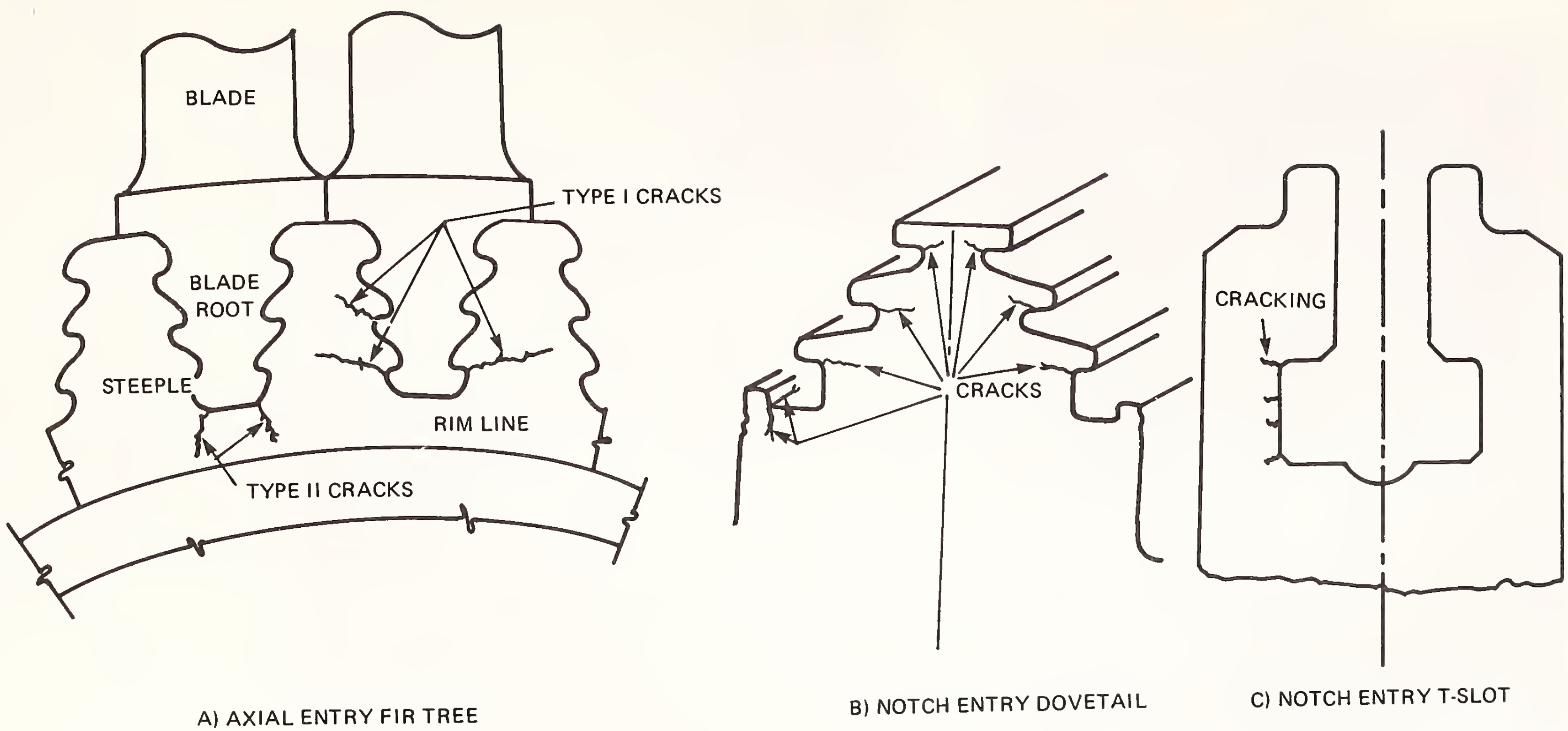


FIGURE 6. DISC RIM/BLADE ATTACHMENT CRACKS

FIGURE 7. DISC RIM MAGNETIC PARTICLE CRACK INDICATIONS FOR FIR TREE ROOT

TABLE 1. BRITISH SUMMARY OF DISC CRACKING EXPERIENCES (REF 1)

| (1) | (2) | (3) | (4) | (5) |
|---|---|---|---|---|
| Type of Rotor | No. of Rotors Examined | No. of Rotors Containing Cracked Discs (% of 2) | No. of Discs Examined | No. of Discs Cracked (% of 4) |
| Type 1 | 33 | 15 (45%) | 136 | 31 (23%) |
| Type 2 | 9 | 1 (11%) | 108 | 1 ( 1%) |
| Type 3 | 49 | 34 (69%) | 460 | 92 (20%) |
| Type 4 | 11 | 0 | 106 | 0 |
| Totals | 102 | 50 (49%) | 810 | 124 (15%) |

FIGURE 8. SURFACE OF DISC SEGMENT AFTER COMPLETE FRACTURE

The British found that the cracking was concentrated on those discs that carried turbine stages where the steam expansion crossed the condensation line. This would allow moist conditions to occur in the disc keyway and bore when the metal temperatures are below the saturation temperatures. Indeed, the disc-cracking experience statistics are more severe if only discs operating in this condition are considered.

The British found disc cracking in discs which had semicircular keyways (Rotor Types 1 and 3, Table 1) with high stresses and small clearance fits between the keys and keyways. Discs with rectangular keyways (Rotor Types 2 and 4) had higher stress concentration factors, but had either lower bore tangential stresses or larger key/keyway clearances. No keyway cracks were reported, and only one bore crack was found in these types of designs.

For both discs and laboratory specimens, the crack sites were associated with either nonmetallic inclusions, corrosion pits, machining defects, heavily worked surface layers, or blisters of oxide. The cracks were intergranular, heavily branched, and filled with dark oxide corrosion products. Some cracks began in a transgranular mode, but changed to an intergranular mode within the distance of one or two grains.

No correlation was found, nor evident, between the incidence of disc cracking and the presence of or lack of any aggressive chemical contaminant in the steam. Although some differences in cracking experiences exist between different plants operating similar units, operating practices could not be used to justify these differences.

Laboratory tests conducted with disc and rotor steels showed that stress-corrosion cracks could be produced in "pure" wet steam at high loads near or beyond yield stress, and with tight crevice conditions or stagnant conditions which promoted pitting.

For the most part the British discs were made from 3 percent CrMo steel, but some 3.5 percent NiCrMoV and 3.5 percent Ni low alloy steel were also used. The 3 percent CrMo and 3.5 percent NiCrMoV steels were found to be equally susceptible to SCC. No cracks were found on the 3.5 percent Ni steel, but this steel is not suitable for most disc applications.

The British did experience disc-rim cracking on one of their units, and they warn of the possibility of more extensive cracking in this area due to the high stress and crevice conditions present in this location. One British report[5] states that "the work suggests that highly stressed, tight crevices operating in wet, pure steam are required for pitting and eventual cracking of the typical turbine material used throughout the world."

## 4.0 United States Experiences

In the U.S., the incidences of disc cracking have been more extensive with cracks not only in the keyway and bore, but also on the disc hub, web and rim in both nuclear and fossil fuel power plants. The total number of disc- cracking cases will probably never be determined since many repairs are completed without reports being published. However, sufficient data is available from public reports to determine the extent and nature of the problem[6,7].

In addition to the 25 earlier mentioned disc-cracking cases for which detailed information is available, since 1969 approximately another 25 units have experienced disc cracks about which little information is publicly available. Table 2 presents a summary of the reported U.S. disc cracking experiences. At least three disc bursts have occurred and more have been prevented by correct operator action and utility precautions. One unit was shut down prior to a disc burst when the cracked disc caused high vibrations from mass unbalance and blade rubbing. Other units have been shut down specifically for disc inspections and found to have significant disc cracks.

Several correlations between the British and the U.S. experiences can be made. The cracking has occurred at the dry-to-wet steam transition zone, sometimes called the Wilson line. Only discs with semicircular keyways have experienced stress-corrosion cracks in the keyways and bores[1]. The cracks are predominately intergranular, heavily branched, and filled with dark oxide corrosion products[4]. A significant amount of an aggressive chemical corrodent is not always found. Although pitting has been evident and associated with some of the cracks, it has not been entirely correlated with all of the cracks as was the British experience.

## 5.0 Nondestructive Examination Methods For Crack Detection

Magnetic particle and dye penetrant NDE methods have been used to inspect for cracking of accessible disc surfaces; however, most of the cracking has occurred in crevice locations inaccessible to direct visual observation. The application of ultrasonic (NDE) methods has been necessary and successful to examine discs for keyway and bore cracks. UT systems used so far have been developed by the turbine manufacturers. Much of the detailed information about these systems is classified as proprietary by the vendors but enough information has been publicly reported to present the general approach used[7]. Figures 9a and 9b show the search unit arrangements used for scanning the disc hub and webs.

The transducers are fitted to plexiglass shoes which have been constructed to match the compound curvature of the disc surface while directing the sound beam at the correct orientation at the bore. A pulse-echo technique is used primarily, but for some disc shapes a pitch-catch method is necessary to inspect the region of the bore and

TABLE 2. SUMMARY OF STRESS CORROSION DISC CRACKING EXPERIENCES IN U.S.

| Location of Crack in Disc | Number of Reported Disc Cracks Experienced* | |
|---|---|---|
| | Nuclear | Fossil |
| Rim/Blade Attachment | 25 | 25 |
| Face, Hub and Web | 4 | 34 |
| Keyway | 32 | 2 |
| Bore | 4 | 0 |
| Complete Disc Burst | 3 | 2 |

*Some discs contained cracks in more than one location.

keyway not accessible when using a single, pulse-echo transducer. In this pitch-catch method a pair of shear wave transducers are placed on opposite sides of the disc web and are aimed tangent to the bore and keyway. Proper coordination of the inward and outward radial movement of the transducer is controlled by automated manipulator arms. Precise locations and angles have been calculated by a dedicated minicomputer which controls the manipulators. In one inspection system, two pulse-echo techniques are used: one for detecting crack indications while the other is used for verifying and measuring crack depth. In the initial technique, a 2.25 MHz transducer is mounted on a contoured plexiglass block so that the ultrasonic waves are directed tangentially toward the keyway. Any cracks above the keyway area will be perpendicular to the sound and will reflect it. Echoes are also received from the keyway and a time difference can be determined and analyzed. By circumferentially scanning in both clockwise and counterclockwise directions, enough sonic information is compiled to compare with a mathematical model of the disc. An analysis of the time differences between signals is made to determine if they are caused by true flaws or spurious and false indications. Time measurement in μs/division on the oscilloscope is multiplied by 0.113 in/μs for shear wave travel to convert to inches.

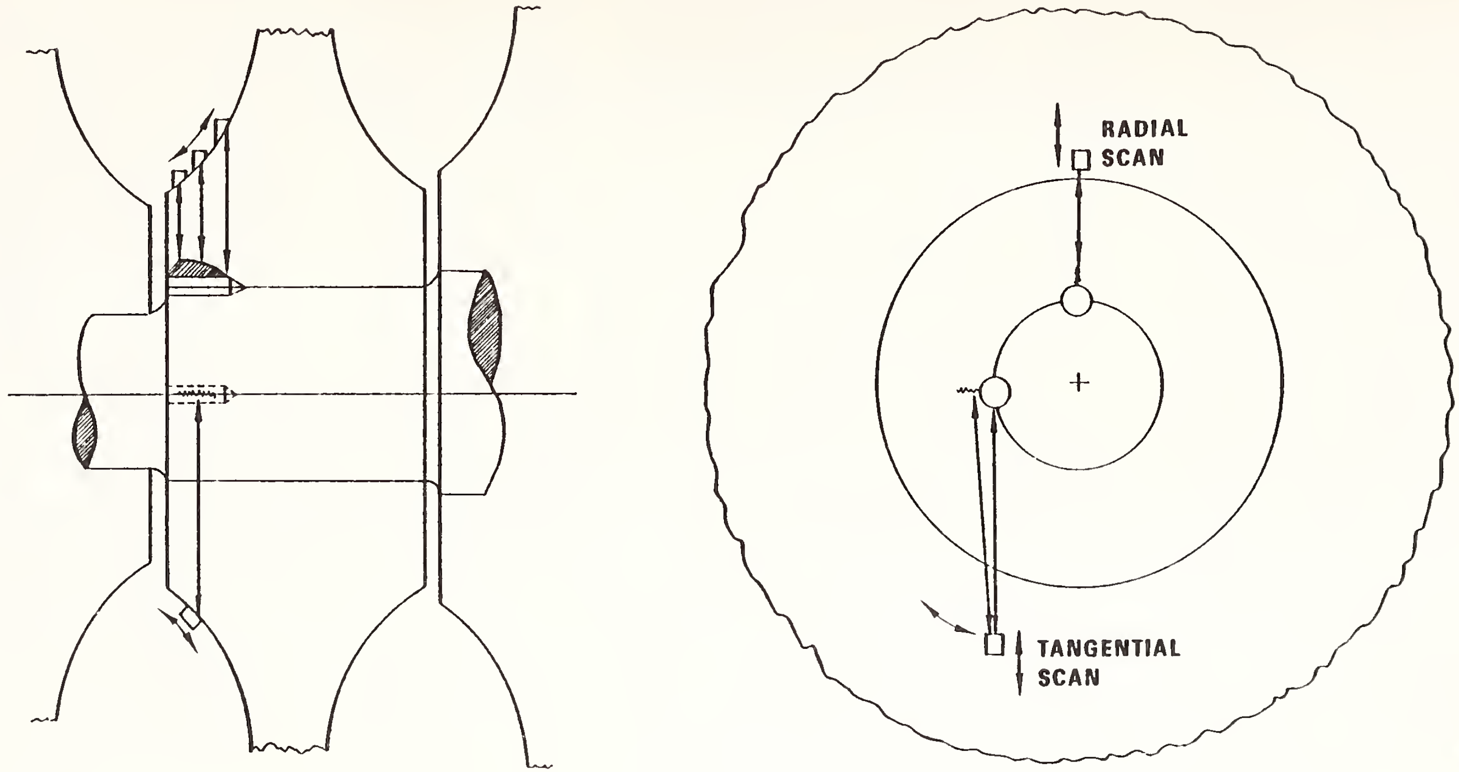


FIGURE 9a. SEARCH UNIT ARRANGEMENT FOR SHORT KEYWAYS

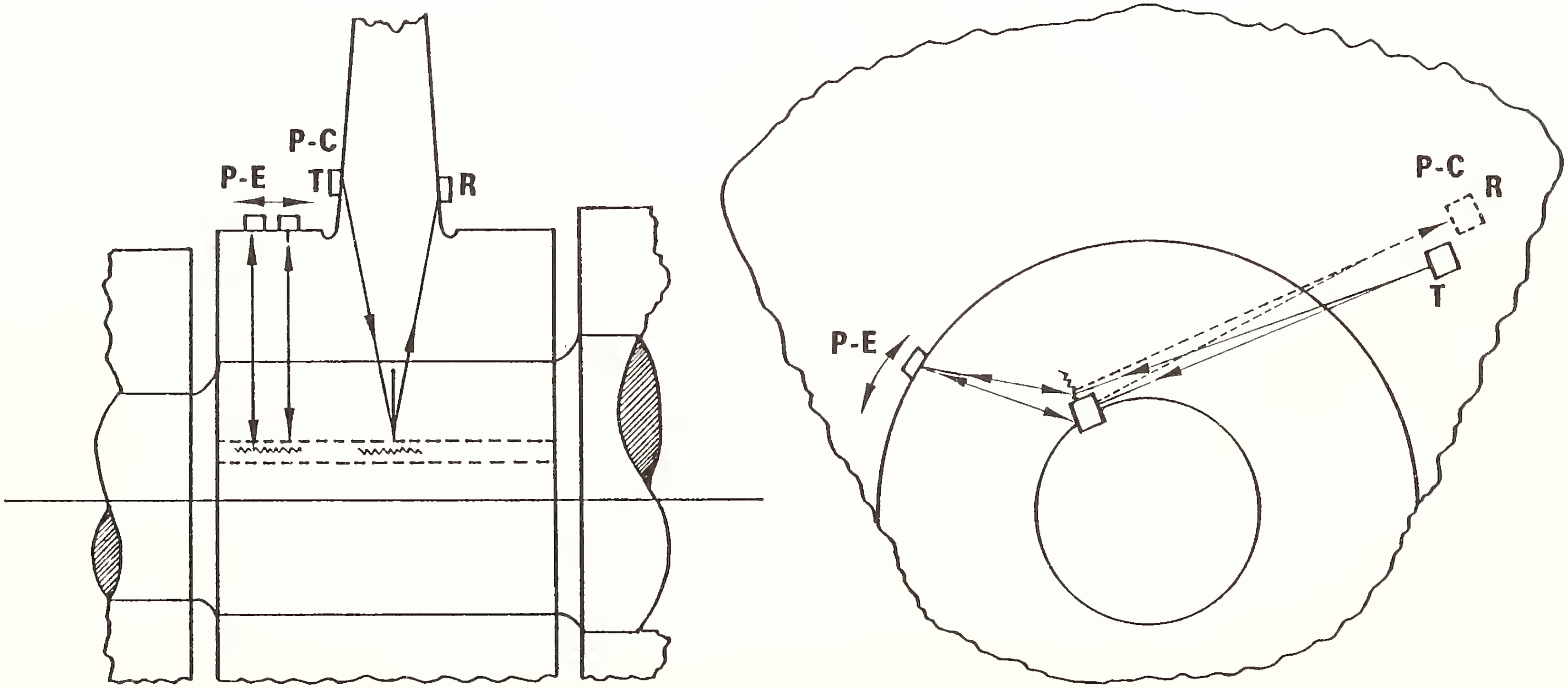


FIGURE 9b. SEARCH UNIT ARRANGEMENT FOR LONG KEYWAYS

Once a crack indication is located, a 5 MHz shear wave transducer is aimed perpendicular to the keyway and in line with the crack. An assumption is made that the crack is intergranular and is branching and spreading as it grows. Since this part of the crack is reflecting ultrasound back to the transducer, the time difference between the keyway echo and the flaw echo can be measured and converted to an estimated crack length. The assumption that the crack is intergranular and heavily branched has been proven to be correct by both U.S. and English experiences. However, an uncertainty factor of 0.060 inch is added to the UT-determined crack length to account for the crack tip beyond the branching extension and measurement errors.

Because of the complex disc geometry, additional signal sources often make interpretation very tedious and time consuming. Use of different transducer frequencies and computerized signal processing greatly increases the speed and accuracy of distinguishing between nonrelevant signals and true defects. It has been found that score marks produced on the bore during assembly operations generated large nonlinear indication amplitudes at a 2.25 MHz transducer frequency. A lower frequency transducer, which produces a more linear response to these small defects, is used to reexamine the disc for defect confirmation. Signals were also coming from mode conversions at keyways which could be mistaken for a defect. These signals have to be carefully examined to see if they are coming from a location which varies with the position of the transducer or if they are originating at a fixed position, as would be expected for a defect. A computerized signal processing correlation technique is used to enhance defect signals and suppress nonrelevant indications.

## 6.0 Metallurgical Analysis and Diagnosis

Metallographic and fractographic examinations of cracked-disc sections have been performed for both British and U.S. cases. Many U.S. metallurgical investigations are still in progress; therefore, not much failure analysis information has been published and publicly disclosed.

From the reports released the analyses have shown that the alloy microstructures have been normal and the chemical compositions have been within specifications. The cracking has been wholly intergranular in nature in 3.5 percent NiCr MoV steel, although transgranular cracking has been found in some discs made of other types of steel. Typical examples of metallographic sections and fractographs are shown in Figures 10 and 11, respectively. The cracks contain dark oxide corrosion products in varying amounts. Pitting damage has been found on the disc rims in most blade-attachment cracking incidents. Also, pitting corrosion has been seen in some of the disc keyway/bore cracking incidents, but insufficient information has been published to say that pitting has been present in all keyway/bore

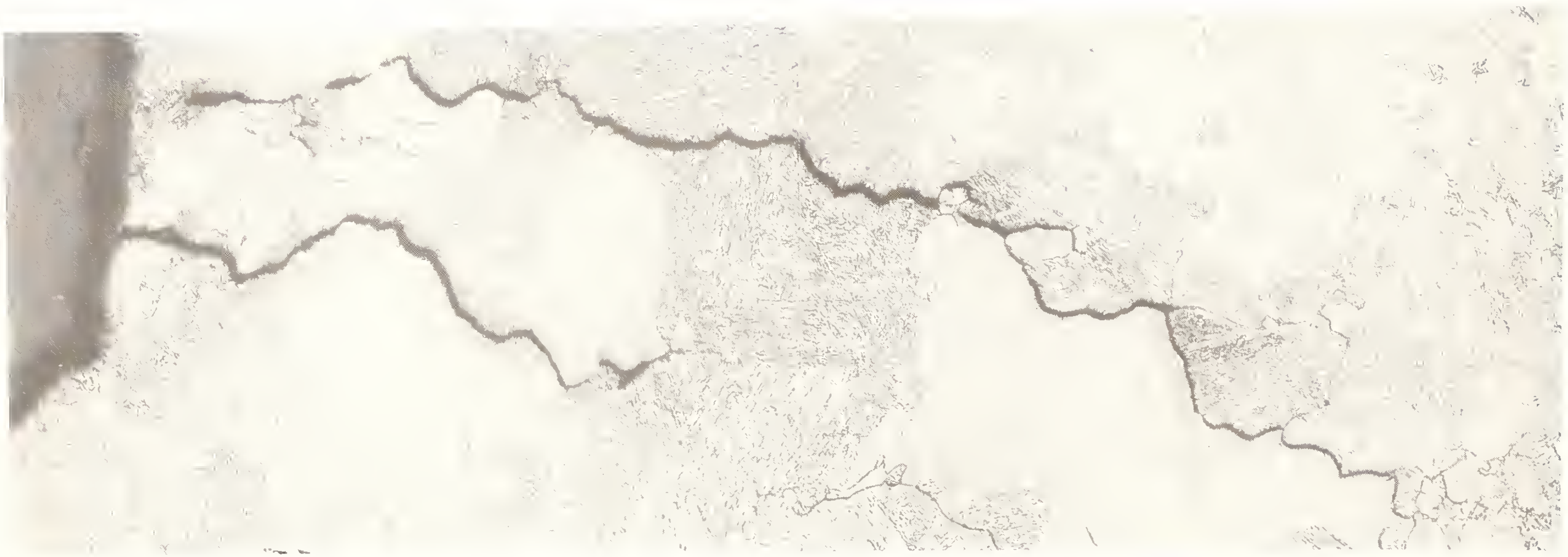

FIGURE 10. METALLOGRAPHIC SECTION OF DISC CRACK 150X

FIGURE 11. FRACTOGRAPH OF A DISC CRACK 300X

cracks. Unlike the British experience, a correlation between pitting and cracking cannot be inferred in the U.S., as yet. Cracking has been found in surfaces that were adjacent to pitted surfaces, but the cracks usually did not originate at the pits.

As stated earlier, the British disc-cracking failure analyses and laboratory testing programs revealed that the cracks were intergranular. In a few cases the initial growth was transgranular, but switched to intergranular at one to two grains deep. Crack initiation was associated with surface irregularities caused by either nonmetallic inclusions, corrosion pits, machining defects, or heavily cold worked layers. Laboratory-cracked specimens showed crack initiation was enhanced by stagnant conditions which promoted pitting. Tests in wet steam produced blisters of oxide on the specimen surfaces or in small, gapped crevices. Crevice tests were performed with pure steam passing through gaps between bent beams of disc steel and disc key materials. Of the four gap sizes employed [(1) 0.05 mm, (2) 0.05 mm, (3) 0.25 mm and (4) 1 mm], cracks were found only in the gaps of .05 mm or less. This correlated to the finding of "high spots" on disc bores that contained cracks and produced crevices between the bore and the shaft[9].

Manufacturing processes that changed the residual surface stresses have been shown to change the potential threat of SCC by affecting initiation. Residual compression stresses were shown to be beneficial in resisting crack initiation. Test specimens that were prepared by surface grinding and then thermally stress relieved had a reduced time-to-failure compared to specimens that were not stress relieved[4]. Other work performed by CEGB showed that residual stresses produced by machining could modify both the rate of corrosion pit initiation and the depth of pit required to initiate a stress corrosion crack[8].

Disc cracking is being experienced on discs that operate within the steam expansion region near the saturation line where condensate may be present on the disc surfaces. In the British turbine designs and the U.S. nuclear turbine designs, this region is toward the L.P. turbine inlet. In most fossil turbine designs, this region is toward the turbine exhaust due to higher superheated steam conditions at the turbine inlet. Changes in turbine loading by varying the steam pressure or temperature will shift the location of the moisture forming region.

In this region, trace impurities in the steam tend to precipitate out and be deposited on the metal surfaces. Concentrating mechanisms can then produce significant corrosive solutions on the disc surfaces. Some of these trace impurities have been identified as species which are known or suspected to cause SCC in turbine steels. Contaminants such as sodium, copper, hydroxides, chlorides, sulfates, sulfides, and organic acids can be introduced into the steam cycle by feedwater treatment system malfunction or from cycle in/leakage from cooling water systems. Oxygen levels may also be significant during operating periods when system leaks are occurring. Variations in the concentration of impurities within the LP turbine environment are difficult to monitor and control.

While it is recognized that the concentration of corrosive contaminants can promote SCC, further improvements in steam/water chemistry practices might prove to be impractical as a solution to the problem since research results indicate that solubilities of some contaminants in steam may be too low to detect with available analytical equipment and procedures.

Since the early 1960's, the yield strengths of the turbine steels have been increased to allow greater electrical ratings and power conversion while limiting equipment physical size. The yield strengths now range from 90 to 160 ksi. Extensive material testing programs are currently in progress to determine the effect of yield strength levels of steels used in the U.S. for susceptibility to SCC in turbine environments. Preliminary results indicate crack propagation rates increase with yield strength, but crack initiation is essentially independent of yield strength.

While high yield strength steels may be a factor, lowering the yield strength may not be possible on discs presently in service, since the discs' stress levels are determined by the physical space constraints within the overall turbine casing.

Other material properties and manufacturing procedures such as trace elements and their segregation at grain boundaries, embrittlement, and heat treatment methods, are being studied to determine their effects on the disc cracking potential.

As mentioned earlier, locations on the disc where tight crevices are formed seem more susceptible to cracking. Elimination or redesign of the crevices was performed by the British turbine manufacturers and operators and is now being considered in the United States.

## 7.0 Risk Assessment and Prognosis

As mentioned in the Introduction, the utilities want to prevent serious damage to their plants and harm to their employees while achieving high productivity from equipment. Each utility has to make a risk assessment as to its NDE inspection program and to its actions after an NDE indication has been found in a disc. An important factor in any prognosis is the crack growth behavior of a defect under actual operating conditions. Unfortunately, the SCC growth behavior of 3.5 percent NiCrMoV turbine steel has been difficult to assess due to some variations in disc steel material and mechanical properties and to large variations in disc environmental conditions, including temperature and chemical contamination. Available data on SCC growth rate, as determined on fracture mechanic-type specimens, is limited and variable and is summarized in Figure 12.

Noting the trend for increased crack growth rates with higher temperatures and higher yield strengths, empirical crack growth rate curves have been prepared from the British and U.S. experiences with certain assumptions about crack initiation and chemical contamination. An example of the type of curves used to predict crack sizes and calculate remaining life is shown in Figure 13. These curves take into account the two main parameters of yield strength and disc-operating temperature.

The assumptions made in determining the remaining life are:

(1) Crack growth occurs only in discs where moisture conditions could be present during operation. Discs that operate ahead of the moisture-forming zone are not considered to be susceptible to SCC. The moisture-forming zone can vary considerably, depending on steam superheat temperature at the turbine inlet.

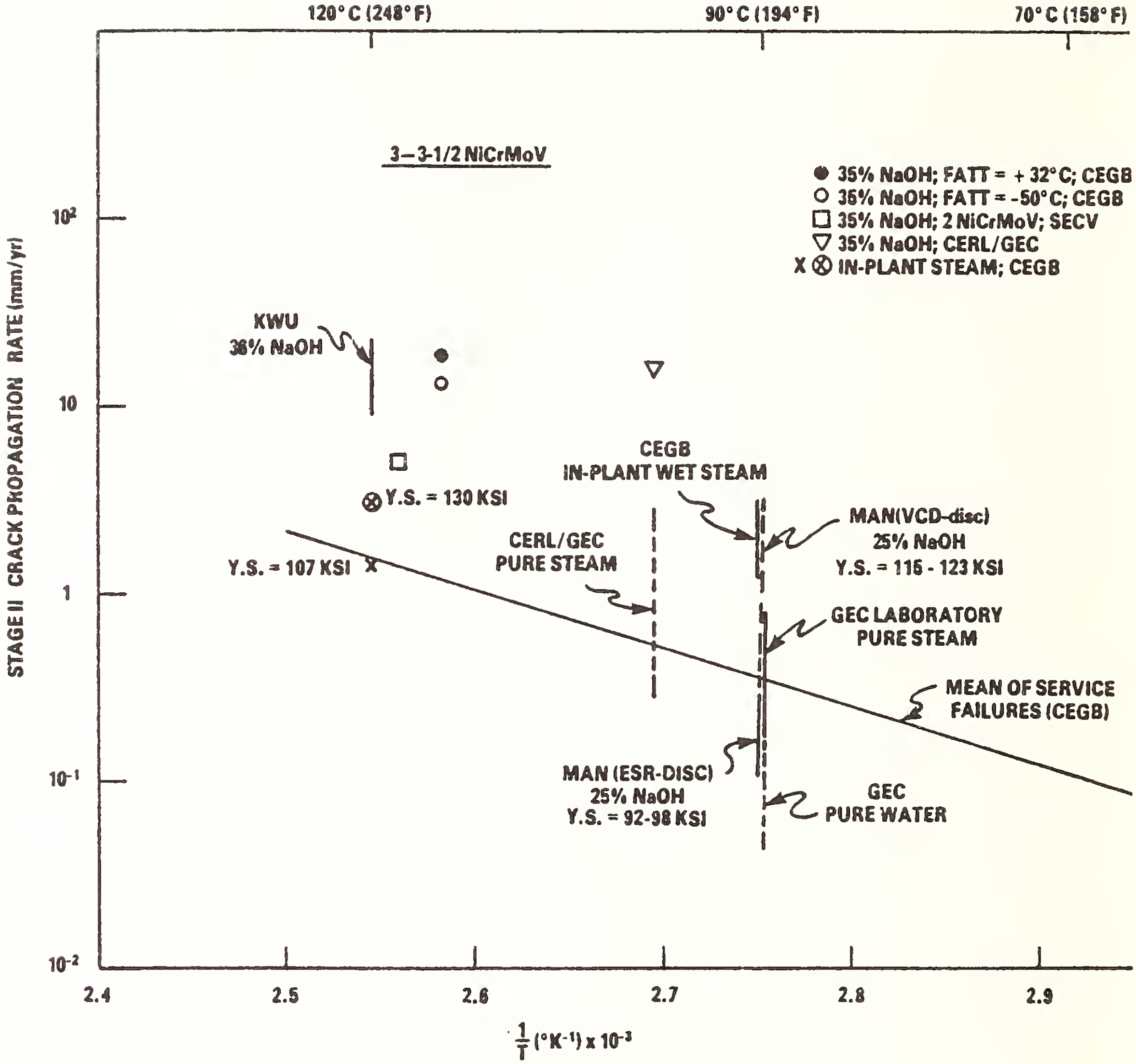


FIGURE 12. SUMMARY OF STAGE 11 CRACK GROWTH DATA FOR 3-1/2 NiCrMoV TURBINE STEELS

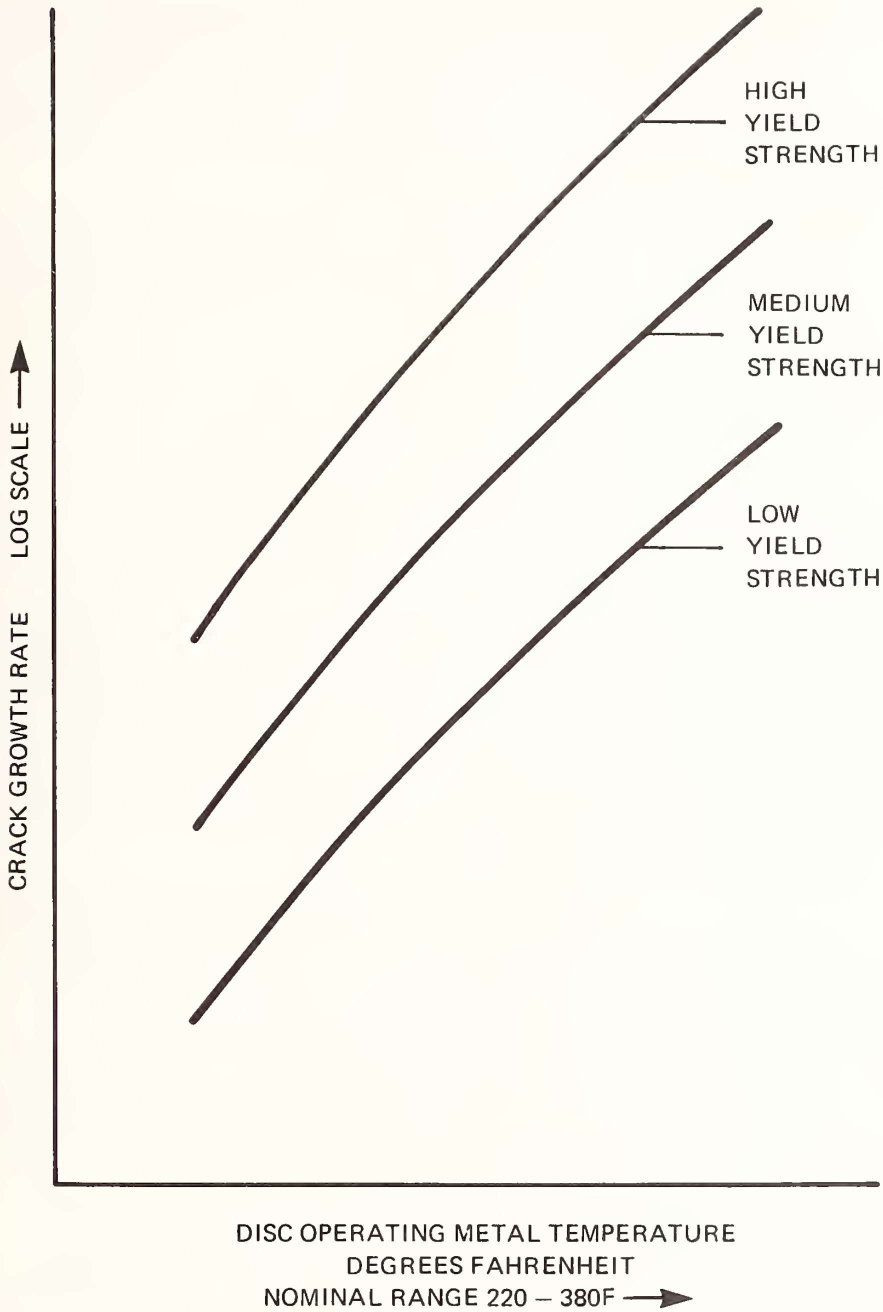


FIGURE 13. ESTIMATED CRACK GROWTH RATES IN 3.5% NiCrMoV DISC MATERIAL AT DIFFERENT STRENGTH LEVELS

(2) Crack growth begins at initial operation. Very little data is available concerning crack initiation time. The exact operating conditions for crack incubation are not adequately understood nor monitored so that any lifetime prediction analyses assuming anything other than zero initiation time are not possible.

(3) A factor for environmental conditions is considered for some plants where problems have existed in maintaining low steam purity limits, and corrosive contamination is known or suspected.

(4) Crack propagation is continuous and at a constant rate.

An example of the calculations used for prognosis is shown in Table 3. A critical crack size, $A_c$, is calculated based on a crack shape factor, fracture toughness, ($K_{IC}$), and operating stress. A postulated crack size is determined from the crack growth rate curves and the time period from initial operation to the planned inspection. In the table, two examples of medium strength discs are shown at different disc metal operating temperatures. From the crack growth curves, the rate for each example is determined and multiplied by the total operational period from start-up to time of inspection. The postulated crack size, a, is the product of the two. Critical crack sizes for the bore, $A_{CB}$, and the keyway, $A_{CK}$, are calculated with a fracture mechanics method. The postulated crack size is compared to the calculated critical crack size to determine a degree-of-criticality ratio, $a/A_c$.

The British found that they were able to predict crack size fairly accurately with fracture mechanics methods and were able to delay the replacement of some rotors, which were made of 3.5 percent NiCrMoV steel and had higher toughness than 3 percent CrMo steel discs. Their disc examination program revealed cracks in the keyways as big as the calculated critical size; however, these had not failed. They attribute this to conservative calculations in that an intergranular stress-corrosion crack branches significantly when it grows longer than about one millimeter[1]. It was also demonstrated from laboratory specimens with such branched cracks that their failure load was about 50 percent higher than identical specimens containing fatigue cracks of the same projected length. The English determined that the effective stress intensity for a branched stress-corrosion keyway crack was less than predicted by conventional analysis, i.e., the effective crack-tip stress intensity is lowered due to crack branching. To validate their prognostic technique the British recorded the maximum crack depths from 28 rotors following destructive examinations. The crack growth rate data for even the deepest cracks found were in close agreement with data obtained from laboratory tests showing that initiation times are short in comparison with crack growth periods.

The Midlands Region of CEGB used a "probability of survival" of a rotor risk assessment method in which they developed an empirical three stage model of disc failure[10]. Probabilities were ascribed to crack initiation, to crack growth in pure wet steam, and to brittle fracture when the crack grew to its critical depth. From these

TABLE 3. EXAMPLE OF PROGNOSTIC TECHNIQUE

(3a) Postulated Crack Size From Initial Operation

| DISC | DISC METAL TEMPERATURE °F | DISC YIELD STRENGTH | CRACK GROWTH RATE, IN./MO. | MONTHS TO INSPECTION | POSTULATED CRACK SIZE, A, IN. |
|---|---|---|---|---|---|
| A | 330 | MED | 0.030 | 30 | 0.90 |
| B | 210 | MED | 0.004 | 30 | 0.120 |

(3b) Critical Crack Size and Ratio

| DISC | $K_{ic}$ (KSI $\sqrt{\text{IN.}}$) | TANGENTIAL BORE STRESS, $\sigma_T$(KSI) | CRITICAL CRACK SIZE AT BORE, $A_{cb}$ (IN.) | CRITICAL CRACK SIZE AT KEYWAY, $A_{ck}$ (IN.) | CRACK RATIO AT INSPECTION $a/A_{ck}$ |
|---|---|---|---|---|---|
| A | 200 | 75 | 2.52 | 2.15 | 0.42 |
| B | 230 | 70 | 3.83 | 3.45 | 0.03 |

$A_{cb} = (Q/1.21\pi)(K_{ic}/\sigma_T)^2$ Where Q = SHAPE FACTOR FOR ASPECT RATIO AND STRESS TO YIELD STRENGTH RATIO, Q = 1.35 FOR EXAMPLE

$A_{ck} = A_{cb}$ - KEYWAY RADIUS

failure probabilities, a survival probability of each disc was calculated and the probability of survival of a rotor was determined as the product of its individual disc probabilities of survival.

Disc cracking statistics were obtained for each type of rotor, composition of disc, and the disc position on the rotor to generate probabilities for crack initiation. Growth rate probability was deduced from confidence limits superimposed on a plot of apparent velocity of the deepest cracks in keyways versus the operating temperature. Brittle-fracture probability was estimated from confidence limits superimposed on a plot of the fracture toughness versus disc operating temperature minus the 50 percent Charpy FATT.

Survival probabilities were calculated for the failed Hinkley Point A rotor and other rotors with known cracks. The Hinkley Point A rotor had a probability of survival of 0.38 while another type of rotor had a probability of survival of 0.85. One rotor of this type had not failed in service but was found to contain cracks as big as the calculated critical crack size. All other rotor types studied had a probability of survival of at least 0.9969.

Using probability of survival data, the Midlands Region was able to assign priorities in their rotor inspection and rehabilitation program which minimized the risk of rotor failure and had minimum effect on system operation.

In the U.S., a main concern is the possibility that a disc failure in a nuclear turbine would generate a missile that would penetrate the turbine casing and have enough energy remaining to strike and damage a plant component critical to the safety of the nuclear steam supply system. A missile generated by a fractured disc could weigh over 3100 pounds, have an exit velocity over 400 ft/sec, and a calculated injection energy over 10 million ft-lbs.

The NRC has an interest in this and the probability ($P_4$) of the occurrence of damage to safety-related components is calculated as the product of the probability ($P_1$) that a turbine failure will result in the ejection of a missile, the probability ($P_2$) that the missile will strike a barrier which houses a critical plant component, and the probability ($P_3$) that a missile will penetrate a barrier and thus damage a critical plant component. $P_4$ valves of $10^{-14}$ per year have been calculated for design overspeed accident conditions.

The probability ($P_1$) of a turbine failure resulting in the ejection of a missile has been reevaluated to consider the SCC problem experienced and the nondestructive testing program being performed. Even with the reassessment, the probability of turbine missile-caused damage to a safety-related component is extremely small but is dependent on the inspection interval.

## 8.0 Industry Response

As a result of the recent turbine disc-cracking experiences in the U.S., a multi-task response by electric utilities has been undertaken. Working with the turbine vendors, the detection programs have been accelerated on a priority basis for the more susceptible type rotors. Repairs and modifications to component configurations have been performed as expeditiously as possible.

Failure diagnoses have been performed and prognosis-based fracture mechanics evaluations have been completed which allowed some units to return to service with small NDE indications. In other cases, the discs have been mechanically removed and replaced by temporary pressure reducing plates to allow the unit to be returned to service at a reduced electrical generation output. Spare rotors have been used when available and additional spare equipment has been ordered by the utilities for future use.

The utilities, through their research organization, EPRI, have initiated a series of research programs to develop a thorough understanding of the causes of disc cracking, to identify techniques to eliminate or minimize disc cracking in the future, and to further develop the probabilistic methodology for risk assessment of turbine missile- induced damage to nuclear power plant structures and components.

A major program being conducted under EPRI sponsorship is Research Project (RP) 1398, "Stress Corrosion Cracking of Low-Pressure Turbine Discs." The objectives of RP 1398 are:

(1) to determine and rank the contributions of environmental, material and structural variables with regard to SCC crack initiation and growth in shrunk-on discs,

(2) to establish the costs versus detection probability of state-of-the-art NDE methods,

(3) to provide utilities with a basis for determining the most cost-effective maintenance plan for their turbines, and

(4) to evaluate methods for preventing or reducing the frequency of cracking incidents.

Other existing EPRI projects are concerned with turbine steam chemistry and environment, material properties effects, missile impact tests, and analysis techniques of missile risks. Table 4 is a listing of current EPRI projects in the turbine disc-cracking area.

TABLE 4. EXISTING EPRI PROJECTS RELATED TO TURBINE DISC CRACKING

| PROJECT | TITLE |
|---|---|
| RP399 | MISSILE IMPACT TESTING AND ANALYSIS |
| RP559 | ELIMINATION OF IMPURITY-INDUCED EMBRITTLEMENT IN STEELS |
| RP700-4 | TURBINE-RELATED FAILURE ANALYSIS |
| RP700-5 | STRUCTURAL AND MECHANICAL ANALYSIS METHODS |
| RP969 | TRANSPORT OF CORROSIVE SALTS FROM STEAM |
| RP1068-1 | DEPOSITION OF SALTS FROM STEAM |
| RP1124 | TURBINE CHEMICAL MONITORING |
| RP1166-1 | CORROSION AND CORROSION CRACKING OF MATERIALS FOR WATER COOLED REACTORS |
| RP1398-1 | A METALLURGICAL INVESTIGATION OF RIM CRACKING IN A LP STEAM TURBINE DISC |
| RP1398-2 | STRESS CORROSION CHARACTERISTICS OF TURBINE ROTOR MATERIALS |
| RP1398-4 | TURBINE ROTOR & DISC METALLURGICAL CHARACTERISTICS RELATED TO STRESS CORROSION CRACKING BEHAVIOR |
| RP1398-5 | CRACKING OF STEAM TURBINE DISCS |
| RP1549 | PROBABILISTIC ANALYSIS OF TURBINE MISSILE RISKS |

The major turbine vendors, the utility insurance companies, and the NRC have been involved and have responded with their investigations and analyses of the turbine disc-cracking problem.

Hopefully, the combined efforts of all interested and active organizations will result in the elimination of future disc-cracking problems by providing the necessary detection equipment, diagnostic evaluations, and prognostic techniques. Through these efforts the electric utilities will be able to continue to provide for our electrical energy needs.

## 9.0 Acknowledgement

This paper presents information that was obtained from several published sources and from individuals who have worked extensively in trying to resolve the disc cracking problem. I wish to thank the following for their contributions: Dr. G. R. Leverant, Dr. C. M. Teller, Mr. F. F. Lyle, Jr., and Mr. H. C. Burghard, Jr. I want to acknowledge the Electric Power Research Institute for encouraging the presentation of this information and for their support of the research programs for improved DD&P techniques. Special recognition is also given to the British utilities, turbine manufacturers, universities, and authors for their efforts in publishing their turbine disc cracking experiences so that all could benefit from their results.

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# A DOPPLER TECHNIQUE FOR DETECTING AND LOCATING EXCESSIVELY VIBRATING BLADES IN A RUNNING TURBINE

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## ABSTRACT

A method is presented for pinpointing those blades of a running turbine that might be vibrating excessively in resonance with either running speed, nozzle passing frequency, or one of their harmonics. Only one stationary sensor per stage is required, along with a once-per-rev phase reference signal. The method utilizes specific signal manipulations in both the time and frequency domains to uncover the telltale Doppler signature, which normally would be masked in the raw signal. It is shown in step-by-step fashion how each part of this manipulation process combines to achieve the desired goal. The technique appears to be well suited for eventual use in an on-line, microprocessor based, blade vibration monitoring system. However, since an experimental procedure to prove the technique has not yet been undertaken, it must still be considered as conceptual.

## INTRODUCTION

### The Need

Data collected by the Edison Electric Institute from 1967 to 1976 shows that turbine blade failures are a major cause of forced outages for U.S. utilities (1). The Electric Power Research Institute, EPRI, considers blade failures their highest priority problem item in the turbine-generator area (2).

The prime suspected cause of blade cracking is high cycle fatigue, or more likely, stress corrosion combined with fatigue. In either case, fatigue plays a major role. Therefore, some means for detecting excessively vibrating blades might be useful in helping to avoid catastrophic blade failures, and with prompt reaction, even the cracks themselves might be prevented.

To date, there are no good techniques for spotting either vibrating blades or cracked blades in service. The technique described in this paper, however, appears to hold some promise for finding excessively vibrating blades in a running turbine, and for detecting when a significant crack results from that vibration.

## Resonant Blade Vibrations

High amplitude, resonant, blade vibration results when one of the many forcing frequencies that act on the blade coincides with one of its many natural frequencies.

The forcing frequencies consist of all the multiples of nozzle passing frequency, and all the multiples of running speed. The former arise from the series of kicks each blade receives in passing the nozzle wakes, while the latter are due to flow non-uniformities arising from the horizontal split of the nozzle diaphragms and casing, from internal struts, inlet or exhaust openings, nozzle plates or the like. In each case, the multiples, or harmonics of the fundamental frequency are caused by the non-sinusoidal nature of the force waveform. Obviously, for a constant speed turbine, the values of all these forcing frequencies are precisely known for each stage.

This is not the case, however, for the natural frequencies. Here, uncertainties in the degree of fixity between fastener and wheel or tenon and shroud, and empirical estimations as to the effects of centrifugal loading, all lead to uncertainties in the predicted values of the natural frequencies. Also, normal manufacturing and assembly tolerances lead to variations from one blade to the next. Since the most predictable modes are the lower frequency modes, and since they are also the most likely to be subjected to high amplitude excitation, they are the ones that manufacturers concentrate on in attempting to avoid resonant matches.

Actual vibration amplitudes in a resonant match depend on the level of the input at the forcing frequency, on the mode shape and amplification factor of the affected mode, and on the degree of match between the natural frequency and the forcing frequency. Where the two frequencies differ, the vibration occurs at the forcing frequency, not the natural frequency. For some steam turbine blade modes, amplification factors as high as 400 are not uncommon. In addition to indicating the level of increase in amplitude for an exact resonant match, the amplification factor indicates how the vibration amplitude varies with the degree of match. For example, for just a 1% difference between the forcing frequency and the natural frequency, the vibration amplitude drops to 25% of its full resonant value for an amplification factor of 200, and to 12% of its full resonant value for an amplification factor of 400. Therefore, though the high amplification factor modes are potentially more damaging, they require a more perfect frequency match to excite them at their most damaging levels.

Coupling this with the previously mentioned variation in the natural frequencies blade to blade, one can see that even in a problem situation, only a few of the blades or blade groups in a given row might undergo excessive, high amplitude vibration.

The extreme sensitivity to the degree of match between the forcing and natural frequencies for high amplification factor modes, has still another potential benefit. If a resonantly vibrating blade develops a fatigue crack, it is likely that this will shift the affected natural frequency by 1% or more. The accompanying reduction in amplitude should be significant, and can be used as an indicator of the appearance of a crack. Thus, a monitoring system designed to measure resonant blade vibrations, might also be able to indicate crack inception and growth.

## The Basic Measurement Problem

The problem is how to find resonant blades before fatigue cracking actually occurs.

Stationary measurements. If one could measure all the natural frequencies of each blade or blade group of an exposed non-running rotor, then it seems that one ought to be able to spot those blades having an exact resonant match, and replace them even before the unit is ever run. Actually, measuring the natural frequencies of the non-running blades is easy. The problem is that these frequencies will be altered by centrifugal loading, and thus the natural frequencies in the running turbine will be different. For most modes, this difference is great enough, and uncertain enough, to make this procedure unworkable.

Rotating measurements. One obvious technique would be to install a strain gage or two on each and every rotating blade, and look for excessive outputs during operation. In addition to requiring telemetry, the sheer impracticality of this approach appears to be overwhelming.

Another possible approach centers around the fact that a resonantly vibrating rotor blade will result in some corresponding rotor vibration. This rotor vibration will occur at the blade vibration frequency when viewed from the rotor's own coordinate system. However, when sensed by a stationary sensor at one of the bearing caps, it will appear displaced from this frequency, both above and below, by the running speed of the rotor (3). Thus, one must look for these frequencies, not the actual vibration frequency. Still, one might have difficulty discerning the existence of any excessively vibrating rotor blades. The major problem here is that the signal will likely be very low level and may be small when compared to the corresponding harmonics of running speed which are also likely to be present. If this is true, no averaging, either in the time domain or in the frequency domain, can improve the situation. Nonetheless, assuming one were able to determine the existence of an excessively vibrating rotor blade with this technique, finding its location from this data would be virtually impossible.

## THE PROPOSED TECHNIQUE

## Moving Source Doppler

The vibrating turbine blade. Any rotating turbine blade that is vibrating excessively will radiate acoustic energy into the surrounding steam at the frequency of its vibration.

If this energy is to be detected by some fixed sensor, and if the radiating blade is moving such that it has a velocity component along a line joining the moving blade and the fixed sensor, then the frequency seen by the sensor will be altered.

This is known as moving source Doppler, and the new frequency is given by the following relationship (4),

$$f' = f \frac{(V + w)}{(V + w - v)} \qquad (1)$$

where f is the frequency of the source
f' is the frequency seen by the sensor
v is the instantaneous velocity of the source toward the fixed sensor along a line joining the source and the sensor
w is the average velocity of the steam toward the fixed sensor along a line joining the source and the sensor
V is the velocity of sound in the steam.

Figures 1 and 2 show a typical turbine stage, arbitrarily assumed to have 64 nozzles and 67 rotating blades. A single dynamic pressure transducer is mounted in the casing just downstream of the rotating blades. The turbine is assumed to rotate at 3,000 rpm. Thus, the rotating blades with an effective radius of 30 in. (.762 m), are moving with a circumferential velocity of 9,425 in/s (239.4 m/s). The radius at the sensor is 33 in. (.838 m). The velocity of sound in the steam is assumed to be 15,000 in/s (381 m/s), and since the absolute velocity of the steam leaving the blades is mostly axial, it serves to increase the effective sound velocity only when the source is near the sensor. For simplicity, this is neglected in the ensuing discussion.

Assume now that a single blade is vibrating excessively at nozzle passing frequency 3,200 Hz.

Figure 3 shows how this vibration would appear to a strain gage mounted directly on the vibrating blade. Note the 64 constant amplitude cycles in the single revolution record. The blade is presumed to be at position B of Figure 2 at the start of the record.

Figure 4 shows how a simultaneous record from the dynamic pressure sensor might look. This characteristic Doppler waveform was generated here in the following way.

First the waveform of Figure 3 was sampled at 512 equispaced points, each point obviously corresponding to both an instantaneous vibration value, and to a blade position in the rotational cycle. For each blade position, the distance from the blade to the sensor was calculated. Next, the propagation times were calculated by simply dividing these distances by the speed of sound. Then, assuming the amplitude at the sensor to be inversely proportional to the propagation distance, each instantaneous amplitude was first divided by this distance, then transformed to Figure 4, displaced to the right by its propagation time. Those points at the end of the record that fell outside the initial time span were folded around to fill in the missing points at the beginning of the record.

Though no use was made of Equation 1 in deriving the waveform in Figure 4, it can be helpful along with Figure 2 in obtaining a better physical understanding. With the vibrating blade in position A, the sensor sees the lowest amplitude, and the actual frequency. At B, with the blade closer to the sensor, and with a significant velocity component toward it, both the amplitude and frequency are substantially increased. At C, the blade is closer yet, and at the position of the maximum velocity component toward the sensor. Thus, the sensor sees still higher amplitude, and maximum frequency. At D, the blade is closest to the sensor, and so this is the position of maximum amplitude. However, the frequency returns to actual since there is no velocity component in the direction of the sensor. At E, the position of the maximum velocity component away from the sensor, the amplitude is the same as at C, but the frequency is at its minimum. At F, the amplitude is the same as at B, but the still negative velocity component keeps the frequency below actual.

Since the propagation times vary through the rotational cycle, the 512 points that make up Figure 4 are not equispaced, as they would be had an actual signal been sampled by an actual A to D converter. To put the signal in this proper form, Figure 4 is acted upon by an interpolation algorithm, and the result is displayed as Figure 5. Now the signal can be input to an analyzer (as if it had been obtained by the analyzer) and the corresponding frequency spectrum determined, shown here as Figure 6. As expected, the spectrum is broadbanded and extends essentially from 1965 Hz to 8,610 Hz, the lower and upper frequencies as predicted by Equation 1 for this case. Also, since it takes the same amount of time for the frequency to traverse 5,410 Hz on the upper end, as it does to traverse 1,235 Hz on the lower end, the higher sweep rate in frequency-per-unit-time on the upper end, comparatively attenuates this region of the spectrum.

Since the waveforms of Figure 4 and 5 start at a known rotor position, the time till the telltale frequency drop indicates the reverse rotational angular distance from the sensor to the radiating blade at this known rotor position. Thus, the characteristic Doppler waveform detects the vibrating blade, indicates its relative vibration level, and pinpoints its location.

It should be noted here that the known start time allows for the determination of the phase of the spectrum as well as the magnitude. This makes it possible to go back and forth between the frequency and time domains, a facility which will be put to good use in the section on signal manipulations.

Multiple blade vibrations. It is of course possible that more than one blade or blade group in the row will be vibrating in resonance. Because of the variations between blades, and the precise frequency match required for high amplitude, resonant vibration, a practical upper limit for the number of blades vibrating in resonance might be three.

In addition to the vibrating blade of the previous example, which was located 90 deg behind the sensor at the start of the time record, two additional vibrating blades are assumed in the present example. One is located 8 blades behind the original in the reverse rotational direction, and the other is located 9 blades behind that. Recall that there are 67 rotating blades in this example.

All three vibrating blades are assumed to vibrate with the same amplitude.

Figure 7 shows the resulting time waveform, and Figure 8 the spectrum. The spectrum, though somewhat different than before, still spans the same frequency range. The time waveform shows three distinct Doppler signatures located as they should be. In the section on signal manipulations, a method will be presented for greatly improving the pinpointing capability of the technique.

In any event, use of the basic Doppler waveform appears to show promise, even for multiple vibrating blades.

## Important Signal Manipulations

Until now, some knotty, real-world problems have been ignored, but they must be dealt with if this is to be a viable blade monitoring technique. Fortunately, there appear to be some straightforward signal manipulations capable of circumventing these problems. Both the problems, and their solutions are discussed in this section.

The first major problem is that of steam flow noise. The rush of steam around the sensor and the adjacent area is bound to result in a significant, broadband, random signal which could easily be an order of magnitude greater than the anticipated Doppler waveform.

The second major problem is that of other order related signals.

The most significant of these will probably be blade passing frequency, since the sensor will see a large impulse with the passing of each blade. If this series of impulses is not perfectly sinusoidal, harmonics of blade passing frequency will also be present. In addition, components at running speed and low orders of running speed are also likely to exist. The blade passing component could be more than an order of magnitude greater than the Doppler waveform, while the orders of running speed will likely be somewhat lower.

Time domain averaging. The steam flow noise can be substantially reduced through a procedure known as time domain averaging.

A once-per-rev phase reference signal is needed, and this can be obtained by using a fixed fiberoptic probe to sense the passing of a small reflective tape segment, located on the rotor shaft. A pulse is generated each time the tape segment passes beneath the probe, each pulse corresponding to that same, known, rotor position.

The pulse is used to start a sampling process of the dynamic pressure transducer signal. The sampling interval is adjusted so that when multiplied by the total number of samples in the window, 512 for instance, it equals the time for one shaft revolution. Also, the signal must be low pass filtered at a little less than half the sample rate, prior to being sampled to prevent aliasing. After the first sampling process is complete, the averager waits for the next pulse then starts sampling again. It keeps repeating this process, adding corresponding sampled points and dividing by the total number of averages. The result is that anything that is not repetitive in the rotational cycle will eventually average to zero. Obviously, the steam flow noise falls in this category and can be substantially reduced. The Doppler waveform and the other order related components will remain undiminished.

A feel for how fast the steam flow noise could be reduced by time averaging can be gained from Figures 9 through 11. In Figure 9 is shown an un-averaged broadband random signal. Figure 10 shows the same signal after 100 averages, Figure 11 after 10,000 averages. Figure 12 shows the corresponding spectra. It should be noted that for the case of a turbine running at 3,000 rpm, 10,000 averages could be completed in less than 7 minutes, even taking into account the nearly full revolution rest between each block of samples.

Frequency domain blanking. It will be seen that even though time domain averaging is very powerful in reducing the noise, it is still not enough. Figure 13 shows how a real, single revolution waveform might look after significant time averaging. Figure 14 shows the corresponding spectrum. Recall that even though only the magnitude is shown here, this is a spectrum with both magnitude and phase and therefore it is possible to inverse transform this spectrum back into time domain. This capability will be made use of shortly.

Though Figure 14 shows the characteristic Doppler spectrum, the waveform of Figure 13 gives no hint of how many blades are involved, or where they might be. In fact, it shows no evidence of Doppler at all. The problem, of course, relates to the presence of a few large discrete components, clearly visible in the spectrum. The largest component at 3,350 Hz is the blade passing component, and it clearly dominates the scene. In fact, anyone so inclined will be able to count the 67 peaks in the single revolution waveform. Also included are large components at running speed, and the 2nd and 3rd harmonics of running speed, 50 Hz, 100 Hz and 150 Hz. These are evidenced in the general up and down movement of the waveform.

All of these components are order related and therefore not removable through time averaging. But they can be removed, or blanked, directly from the spectrum. Then this modified spectrum can be inverse transformed back into the time domain, and the Doppler waveform should then be apparent. These components can be either completely blanked (replaced with zeroes), or the real and imaginary values of the affected spectral points can be replaced with interpolated values. It is this latter technique that was used here to modify the spectrum of Figure 14.

Figure 15 shows the modified spectrum, and Figure 16 the new time waveform obtained by inverse transforming the modified spectrum. The Doppler that was buried in the original waveform was that of the three blade case previously cited. Note the similarity between the waveform of Figure 16 and that of Figure 7. The three Doppler signatures are still readily apparent, and just as important, no shifting in the time domain occurs as a result of this procedure. Thus, the locating capability remains unimpaired.

The essence of the technique should now be clear. The Doppler mechanism transforms the single frequency of the vibrating blade into a broadband multi-frequency function which contains information as to its location. Like the original frequency, all the new frequencies are order related since the waveform retains its rotationally repetitive nature in the time domain. Thus, it is at once possible to use time domain averaging to remove the non-rotational masking effects, and to use discrete frequency blanking to remove the order related masking effects, the latter process hardly distorting the desired signal since the information in the Doppler is spread so broadly in the frequency domain.

Envelope detection. A key feature of this technique is its ability to locate the vibrating blades. This paragraph will describe a process for simplifying and improving this aspect of the Doppler technique.

The object is to know exactly when the vibrating blade is directly beneath the sensor. This does not coincide with the maximum frequency point or with the minimum frequency point, each of which is difficult to ascertain from the waveform anyway. Rather, it coincides with the maximum of the amplitude envelope of the waveform, corrected by the time it takes for sound to travel to the sensor from a blade directly beneath it. For the example cited in this paper, this time is 3 in. (.0762 m) divided by 15,000 in/s (381 m/s), or 0.20 ms.

The amplitude envelope can be developed in the following manner. First, using the Doppler waveform of Figure 7 as an example, all the negative going peaks are made positive as in Figure 17. This has the dual effect of shifting the frequency of the original Doppler band upward, and adding in a new low frequency band as shown in the corresponding spectrum, Figure 18. Next, all the components above 1965 Hz, the original Doppler minimum frequency, are blanked to zero, leaving just the low frequency portion of the spectrum. Reconstructing this low frequency portion then yields a new waveform, proportional to and synchronized with the envelope of the original Doppler waveform (see Figure 19).

The three time markers at 5.20 ms, 7.59 ms, and 10.27 ms (with 0.20 ms added for the travel time correction) correspond to the times that the three blades pass beneath the sensor. Recall that the resolution from one blade to the next is 0.30 ms in this example. Clearly, when comparing Fig. 19 with Fig. 7, a great improvement in the ease and accuracy of determining exactly which blades are vibrating has been made.

## SUMMARY

A technique has been presented for detecting and locating turbine blades vibrating excessively in resonance with either running speed, nozzle passing frequency, or one of their harmonics.

Only a single dynamic pressure transducer per stage is required, along with a once-per-rev phase reference signal. The pressure sensor sees the radiated energy from the vibrating blade as a rotationally repetitive Doppler waveform.

The nature of the Doppler signal allows it to be greatly enhanced through the use of specific signal manipulations such as time domain averaging, frequency domain blanking, and envelope detection. These enhancement procedures are essential to the technique since they help uncover the Doppler signal itself and the information it bears. Consequently, the technique has been dubbed Enhanced Doppler at FRC.

A mathematically developed Doppler signal has been utilized in this paper, and with it, the techniques described appear to work well in being able to detect and locate three vibrating blades in a hypothetical example.

The next step would be to conduct tests on a test stand, utilizing an actual turbine stage. With the rotor blades strain gaged and their outputs telemetered sequentially, the speed could be adjusted till one or more of the blades is vibrating in resonance. The corresponding pressure transducer signal could then be manipulated as described in this paper to see if the vibrating blades can be successfully detected. Means of dealing with problems such as multiple transmission paths, or echoes, may also have to be developed.

Assuming positive results here, the next step would be to develop a prototype monitoring system to be put in service on an actual turbine. As indicated in the paper, this system might also be able to detect the appearance of a crack in a blade that has been vibrating excessively.

## ACKNOWLEDGEMENTS

The author would like to acknowledge the very valuable assistance of his colleagues Dale Righter and Mark Altschuler of FRC, and the early encouragement of Dr. Simon Braun of Technion Israel Institute of Technology.

This paper was initially presented at the ASME 1980 International Conference on Reliability, Stress Analysis and Failure Prevention, August, 1980, and published in the conference proceedings.

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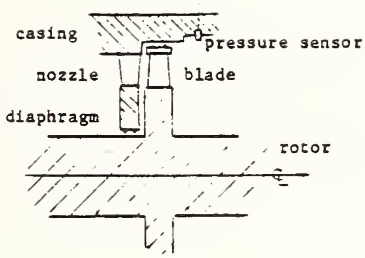


Fig. 1. Typical turbine stage, with pressure sensor location

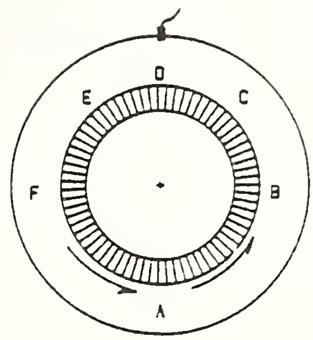


Fig. 2. Rotating blades with angular position references

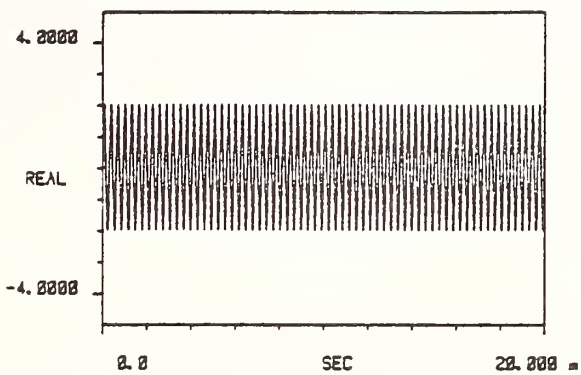


Fig. 3. Blade vibration as seen by a strain gage mounted on it

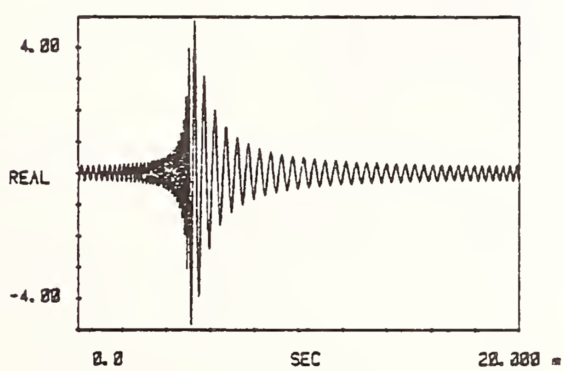


Fig. 4. The blade vibration as seen by the pressure sensor

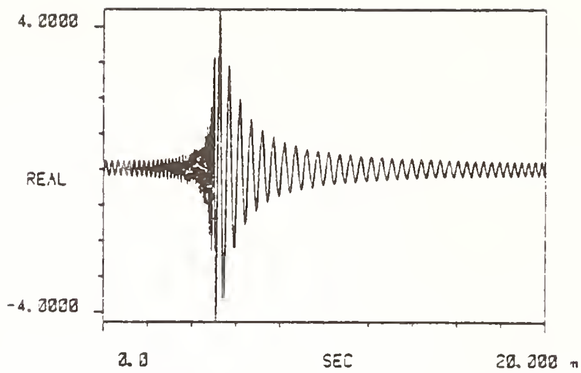


Fig. 5. Waveform of Fig. 4, but with equispaced sampling

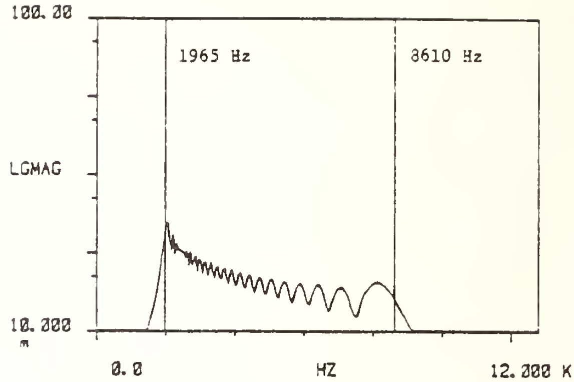


Fig. 6. The frequency spectrum of Fig. 5

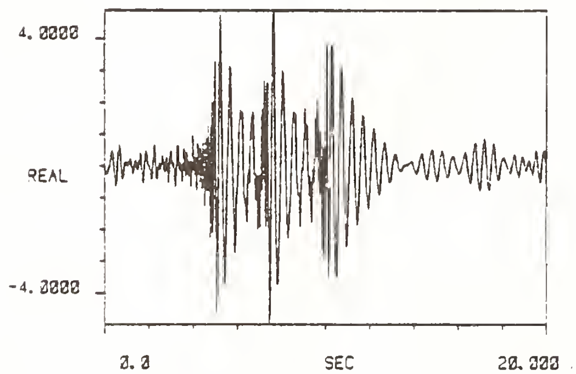


Fig. 7. The Doppler waveform with three vibrating blades

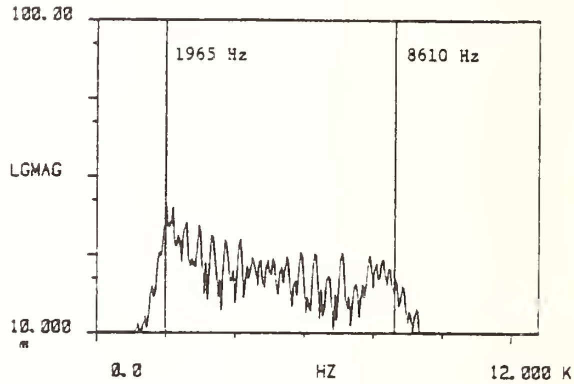


Fig. 8. The frequency spectrum of Fig. 7

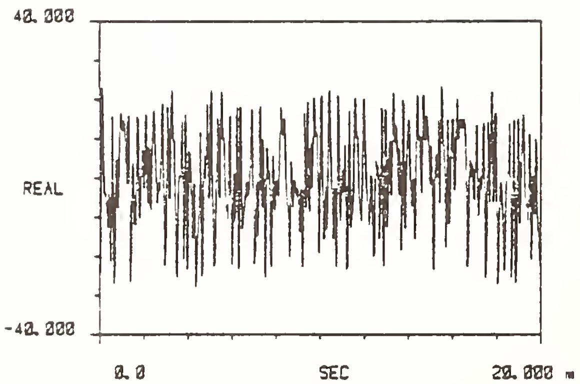


Fig. 9. Unaveraged random signal indicative of steam noise

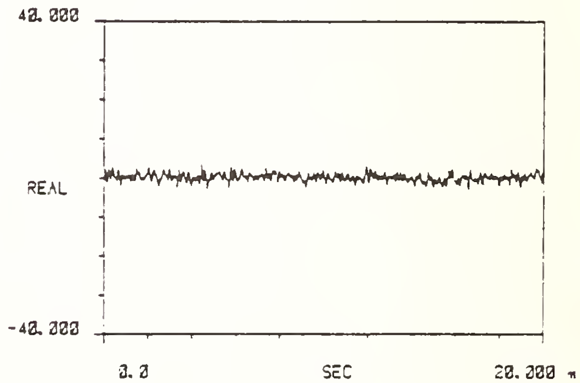


Fig. 10. Signal of Fig. 9 after 100 time averages

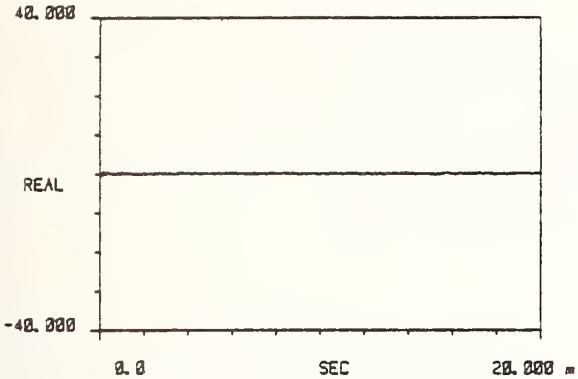


Fig. 11. Signal of Fig. 9 after 10,000 time averages

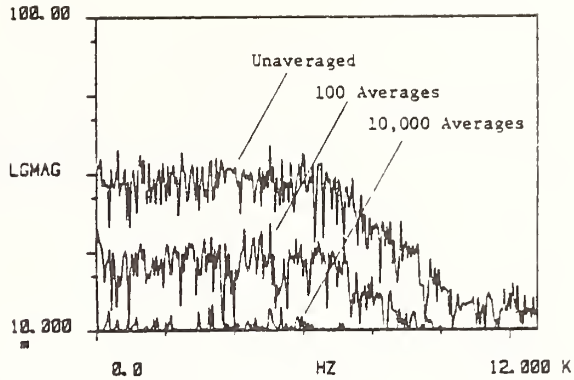


Fig. 12. The frequency spectra of Figs. 9, 10 and 11

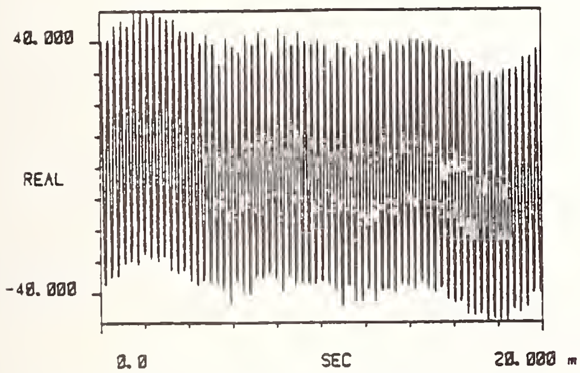


Fig. 13. Hypothetical waveform after significant time averaging

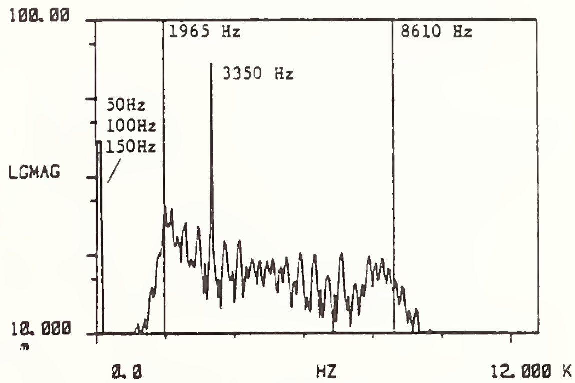


Fig. 14. The frequency spectrum of Fig. 13

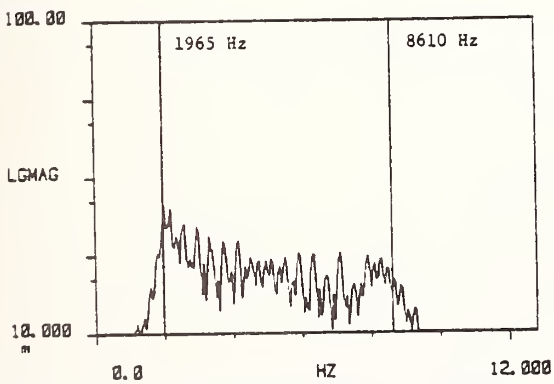


Fig. 15. The spectrum of Fig. 14 after blanking of discrete components

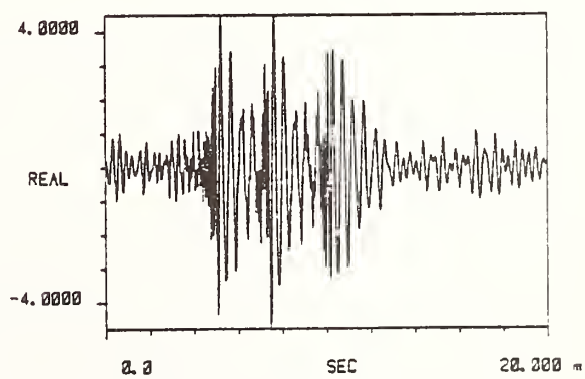


Fig. 16. The time waveform reconstructed from Fig. 15

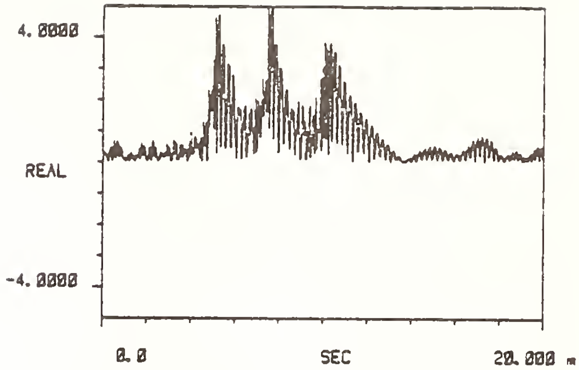


Fig. 17. The waveform of Fig. 7 with negative going peaks made positive

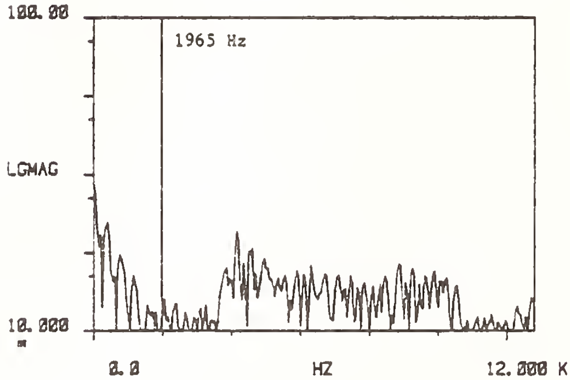


Fig. 18. The frequency spectrum of Fig. 17

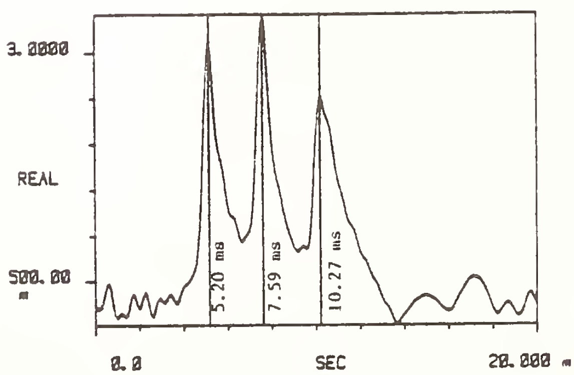


Fig. 19. Reconstructed low frequency portion of Fig. 18

# IMACS COST SAVINGS ESTIMATOR
# A TRADE-OFF ANALYSIS FOR A BUILT-IN MONITORING SYSTEM

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Abstract: Using aircraft as an example, the author presents the trade-off benefits achievable with a built-in, on-board monitoring system that instantaneously alerts users to malfunctions in their fluid system. The system, also applicable to naval vessels and military combat vehicles, is called IMACS (Instantaneous Maintenance and Analysis Check-out System).

Continuous in-system active sensors monitor fluid system flight-critical components for limit parameters such as temperature, leakage, flow, pressure, fluid level, presence of air, differential pressure, etc. The paper analyzes problems that can and do occur in aircraft hydraulic systems causing primary and secondary failures. An economic analysis is presented contrasting an aircraft with and without IMACS experiencing these problems. The effect of the problem is indicated and the necessary corrective action spelled out in terms of maintenance, man-hours, material and downtime.

Additional benefits derived from IMACS, such as reduced energy consumption, reduced logistic support, improved maintenance forecasting, reduced turnaround, improved scheduling and improved capability of personnel skill levels, are also discussed.

Key words: Aircraft hydraulic systems; cost reduction; energy savings; failure detection; failure prevention; flight safety; fluid system components; maintenance effectiveness; on-board monitoring system.

## INTRODUCTION

Early detection and identification of system component malfunction is the key to reducing ever rising maintenance costs and the cost of ownership of today's sophisticated vehicles. Fluid system components are subject to a number of adverse conditions during operation for a variety of reasons, some of which are controllable and some of which are not controllable. These conditions can result in malfunction or a primary failure of a single system component which, if not detected, can cause additional component failures and added cost of repair, or even affect safety.

An additional factor that often contributes to maintenance costs is the inability of the maintenance system to detect and identify which component in the system is in an early state of failure. Maximizing this factor can achieve major cost savings.

Another factor contributing to maintenance costs is the skill level of personnel and equipment available to perform maintenance on today's and tomorrow's ever increasingly more sophisticated vehicle systems. Some maintenance systems benefit from a stable, experienced, well equipped system, but many lack the necessary combination of ingredients to adequately, from a minimum operating cost point of view, deal with today's complex systems.

Therefore, monitoring of critical fluid system component operation has been recognized as an increasingly important contributor to cost control in the maintenance and operational areas.

One of the most neglected fluid systems to benefit from cost effective monitoring is the hydraulic system. Maintenance of this system has tried to cope with increasing system compexity with too few maintenance tools, relying mostly on experience of personnel. Real cost savings can be gained by a serious close look at what a reliable built-in monitoring system can provide and what it has to deal with.

## COST SIGNIFICANCE OF SYSTEM COMPONENT FAILURES

As previously discussed, early diagnosis of an imminent system component primary failure can be very cost effective, especially in preventing secondary system failures. Examples of common hydraulic system component primary failures are shown in figure A. Let us analyze some of these failures in an aircraft to really see what the consequences can be.

One common hydraulic system component failure is caused by internal wear in a pump, resulting in excessive case drain leakage. If this condition continues unnoticed, increased temperature will result, with the possibilities of a series of secondary failures such as fluid breakdown, filters becoming contaminated and overheated, loss of pump seal and loss of system.

Another more difficult problem to diagnose is free air in a nonvented system reservoir. If this problem goes undetected, it can cause pump cavitation or loss of pump prime and possible broken lines with the resultant loss of the system. Safety obviously can be affected by this problem.

Contaminated or plugged filters can cause a variety of system failures which depend on whether the filter assembly incorporates an internal relief valve. Filters are usually located throughout the system and often require panel removal for maintenance checking. This is time consuming and dependent on dedication of maintenance personnel whether <u>all</u> filters are serviced.

If a filter element becomes contaminated and is not changed and causes the filter bypass to open, the system fluid contamination level will increase. This can cause a number of possible secondary failures such as pump overheat due to internal wear, high fluid temperature, pump seal failure caused by wear or high temperature and loss of system fluid and system due to pump seal failure, and possible servovalve malfunction caused by contamination and wear.

On the other hand, if a system filter element becomes contaminated and the filter assembly has no bypass, the differential pressure across the filter element will continue to increase. This will cause the flow to become restricted, and the fluid temperature can rise which, in turn, can cause an external pump seal to fail and breakdown of the hydraulic fluid.

Excessive system relief valve leakage, if allowed to continue, will cause a fluid over-temperature condition. This condition can be extremely detrimental since it can affect the integrity and safety of critical system components such as the pump, filters and reservoir. If the system over-temperature condition is severe, replacement of the identified components may be required.

Loss of minimum accumulator precharge and/or improper position of the internal piston can result in inadequate discharge and system malfunctions. It is common practice to check accumulators for precharge only.

Moisture and loss of charge in pneumatic bottles can cause a failure to discharge or an insufficient discharge resulting in a system malfunction. Pneumatic bottles are commonly checked for charge only.

Still another common system component is excessive internal servoactuator leakage. This condition is usually caused by wear of the metering edge and spool and sleeve surfaces. Because of the relatively large number of servoactuators used in today's vehicle hydraulic systems, determining which servoactuator must be replaced is not easily or quickly accomplished. The diagnosis of excessive internal leakage ranges from involved experienced maintenance techniques to trial and error by replacement.

Excessive aircraft main gear internal leakage is another component problem requiring extensive maintenance action which is usually accomplished at a main overhaul facility with scheduling.

Moisture in reservoir pressurization lines puts unwanted water into the hydraulic system, which causes corrosion and erosion of system components and can further cause decreased lubricity, oxidation, and acid formation in the system fluid.

Low reservoir fluid level is a very common hydraulic system problem that can cause a variety of secondary failures with the possible loss of the system.

The examples discussed are the more common component failures experienced but are not all the problems that occur. However, they serve to illustrate that an undetected primary component failure can lead to far more extensive and serious secondary failures which always involve increased cost.

## OBJECTIVES OF MONITORING SYSTEM

It would seem reasonable, based on our review of today's maintenance capabilities and the significance of penalties to operation, that we can define the objectives of a built-in monitoring system that can reduce cost and demonstrate it. With this in mind, the following are listed:

- Provide early warning of system component malfunction and/or failure.
- Monitor critical and vital component parameters on a real-time opportunistic basis.
- Be able to detect and identify faulty components anywhere in the system.
- Display readout conveniently centrally located requiring no interpretation thus reducing maintenance skill required.
- Utilize available flight tested hardware which will not interfere with system operation, even if it fails to operate.
- Provide signal to ground maintenance and let the signal be the basis for maintenance action.
- Monitoring system must be self contained requiring no auxiliary equipment to operate.
- Operation of monitoring system must minimize energy consumption
- Provide a system which is retrofittable to today's operating systems and newly designed systems.
- Provide a system which is cost effective.

## IMACS IS AN ALTERNATE TO HIGH MAINTENANCE AND OPERATING COSTS

As shown in figure B, IMACS is a built-in system that provides Instantaneous Maintenance Analysis and Checkout of critical system components acceptable limit parameters. It provides small, lightweight sensors permanantly installed at each monitored component that signal by activating a red light on a centrally located panel that a particular component has reached the limit for the parameters being monitored. It detects and identifies the component and indicates which parameter limit has been reached. This simply indicates that a maintenance action is required. The system performs this important maintenence function on demand, reliably, and requiring no interpretation or auxiliary equipment. All that is needed is that the system being monitored is operating. Instant diagnosis is provided indicating a maintenance action to be performed on the identified component. A simultaneous local visual signal is also provided for some components. This signal must be cancelled at the component after the component has been replaced. This feature provides a quality assurance check that the proper maintenance has been performed. A vehicle fluid system utilizing IMACS can be operated on condition basis thus providing the opportunity for planned maintenance and additional cost savings.

## FAILURE ANALYSIS AND COST

Now let us analyze one of the most frequently occurring hydraulic system component failures to see what cost factors are involved.

The more obvious factors to be accounted for in our analysis will be hardware, labor, and loss due to unscheduled non-availability. An accounting of the effect on safety of the component failure also seems appropriate. For our analysis, we will use the schedule of estimated costs shown in figure C. These costs will have to be updated from time to time and altered to suit individual maintenance and supply systems.

Referring to figure D, we have chosen the problem of excessive pump case drain leakage in an aircraft hydraulic system to illustrate our analysis of the maintenance actions required.

The maintenance action steps involved are:

- Diagnosing the problem,
- Performing the actual corrective labor,
- Performing the system integrity check after corrective action has been taken,
- Accounting for materials used,
- Accounting for loss due to unscheduled non-availability.

The possible effect on flight safety is also listed in general, non-monetary terms.

Our analysis will be performed on the basis of a system monitored by IMACS and one without IMACS.

As shown in figure D, if excessive pump case drain leakage is permitted to continue, it results in increasing fluid temperature. Over-temperature can lead to fluid breakdown, filter contamination and overheating and eventually the loss of the pump seal and the entire system. This failure can occur in flight and affect safety.

The costs attributed to diagnosing the excessive pump case drain leakage, labor to replace the pump, perform the system integrity check, account for the pump replacement, account for the aircraft unscheduled downtime are shown in figure D using the cost values listed in figure C. Although no monitoring value is ascribed to flight safety, it is listed as an important factor in our analysis. The costs associated with the primary failure are listed in figure C for the aircraft not monitored by IMACS.

If the aircraft was monitored by the IMACS system, more certainly the excessive case drain pump leakage would be detected in the post-flight checkout by ground maintenance. Replacing the pump before further system damage occurs eliminates secondary maintenance actions and any effect on flight safety.

The costs are listed for one of the secondary failures requiring filter elements change due to high fluid temperature. The IMACS system avoids this secondary failure and the associated cost.

Listed in the same fashion is another possible secondary maintenance action -- that of decontamination of the system brought about by the fluid breakdown. Again, this secondary failure would have been avoided with the use of IMACS.

The summary of cost trade-offs clearly indicates the substantial savings that the IMACS system can accomplish with just one of the common failures associated with this system. This cost can quickly escalate with multiple failures and the size of the fleet. Break-even costs for IMACS applied to a particular system can easily be determined using this analysis.

Military applications concerned with operational readiness can analyze additional flying hours available using this analysis due to anticipated improvements in maintenance or convert this improvement to fewer aircraft required.

Individual analysis sheets are included in this paper for ten other common hydraulic system failures. No cost values have been listed and the reader is invited to apply the described analysis to failures of his concern using his own cost values.

There are additional cost factors that can be considered to extend the present analysis.

ADDITIONAL COST FACTORS

As shown in figure E, fuel savings may also be considered, taking into account the cost of fuel consumption attributed to internal hydraulic leakage. As internal surfaces wear in pumps and servoactuators, clearances increase and internal leakage results in wasted horsepower. A reported United Airlines study[1] estimates that a reduction of internal leakage of one GPM could reduce engine fuel consumption by 2000 gallons per year based on eight (8) hours a day utilization.

Additional fuel savings may be considered where main engines and/or auxiliary power such as APU and ground carts are used for hydraulic system troubleshooting.

Improvement of logistic support can also be considered as reduced cost of ownership.

A long term benefit of IMACS is reduced inventory management of hydraulic system component spare parts. Another factor is improved scheduling and utilization of people and auxiliary equipment. Removal of worn components at convenient, opportune times can also be considered. Also improved logistic energy conservation can be estimated.

Still other cost factors affecting depot and main base maintenance that can benefit from utilization of IMACS include improved maintenence forecasting and scheduling, reduced turn-around and improved capability of personnel skill levels with maintenence requirements.

Life cycle costs of vehicles with and without IMACS can be shown to contribute significantly to reducing cost of ownership by improving maintenance.

ADVANTAGES OF IMACS

There are many advantages of IMACS that can improve maintenance such as:

- Reduces preflight hydraulic system checkout time of flight critical components.
- Display panel identifies malfunctioning components throughout the vehicle.
- Reduces skill level required for routine maintenence.

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1. Reported in the minutes of the SAE A-6 Committee Meeting, October, 1978

- Allows for component removal on "as required" basis between major overhauls as scheduled, thus reducing logistic support and costs.

- Maintains in-flight check of all critical components.

- Identifies faulty components on post-flight check.

- Requires no ground support equipment for check out.

- Increases maintenance efficiency and operational utility.

- Utilizes proven and available hardware.

- Is simple, reliable, practical and essentially maintenance free.

- Can be tailored to incorporate additional design and maintenance requirements.

Shown in figure F is the installation of IMACS in a U. S. Navy A6E aircraft. The flight test is in progress and is proceeding with no reported problems. A full report will be issued at the conclusion of the program.

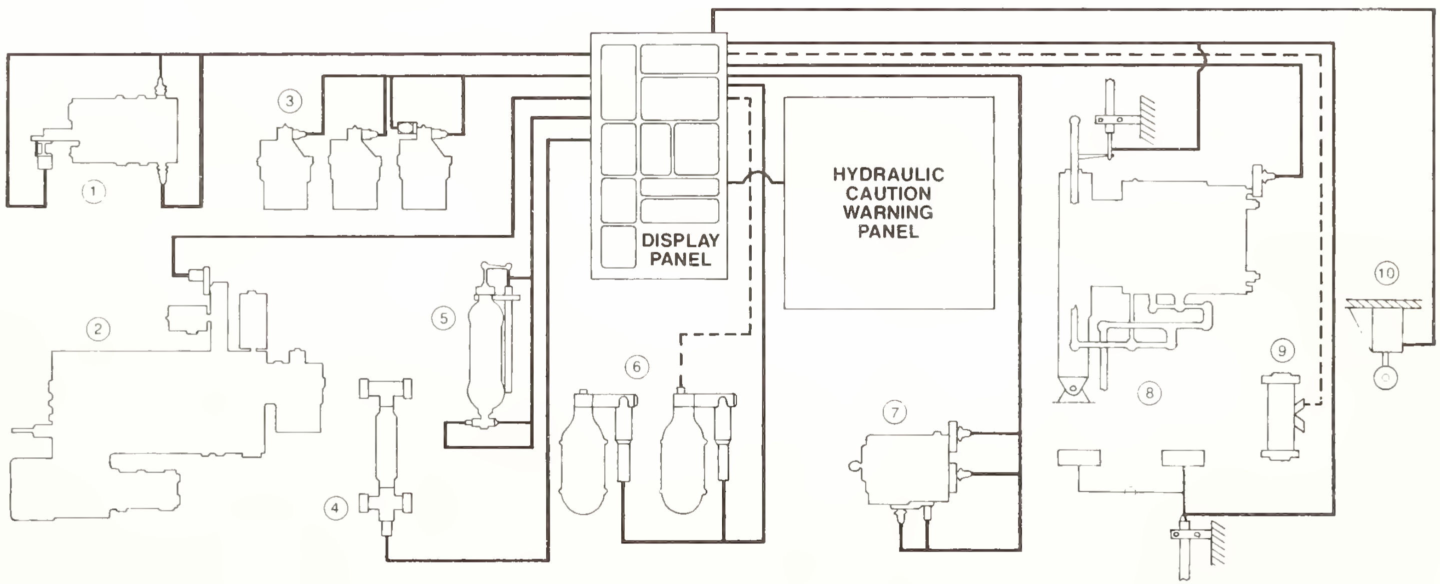
IMACS has broad applications in the field of diagnostics and monitoring and is worth serious investigation.

## COMMON HYDRAULIC SYSTEM COMPONENT PRIMARY FAILURES

- EXCESSIVE PUMP CASE DRAIN LEAKAGE
- AIR IN A NON-VENTED RESERVOIR
- FILTER ELEMENTS CONTAMINATED
- EXCESSIVE SYSTEM RELIEF VALVE LEAKAGE
- LOSS OF MINIMUM ACCUMULATOR PRECHARGE AND/OR IMPROPER PISTON POSITION
- LOSS OF CHARGE AND/OR PRESENCE OF MOISTURE IN PNEUMATIC BOTTLE
- EXCESSIVE LEAKAGE IN SERVOACTUATOR
- EXCESSIVE LEAKAGE IN MAIN GEAR ACTUATORS
- MOISTURE IN RESERVOIR PRESSURIZATION LINE
- LOW RESERVOIR FLUID LEVEL

FIGURE A

# TYPICAL IMACS MONITORING SYSTEM DIAGRAM




| ITEM | SYSTEM COMPONENT | MONITORED PARAMETER | IN SYSTEM SENSOR | SIGNAL TYPE |
|---|---|---|---|---|
| 1 | RESERVOIR | LEVEL<br>AIR<br>TEMP<br>PRESS | MECH ELECTRICAL<br>MECH ELECTRICAL<br>ANALOG SIGNAL GEN<br>MECH ELECTRICAL | POTENTIOMETER<br>POTENTIOMETER<br>I C<br>TRANSDUCER |
| 2 | BACKUP PACKAGE | TEMP | MECH | SWITCH |
| 3 | FILTERS | | | |
| | PRESS | $\Delta P$ | MECH | SWITCH |
| | RETURN | $\Delta P$ | MECH | SWITCH |
| | CASE | $\Delta P$ | MECH | SWITCH |
| 4 | RELIEF VALVE | TEMP | MECH | SWITCH |
| 5 | ACCUMULATOR | PRESS<br>TEMP<br>DISP | MECH ELEC<br>ANALOG SIGNAL GEN<br>MECH ELEC | POTENTIOMETER<br>I C<br>TRANSDUCER |
| 6 | PNEUMATIC BOTTLE | PRESS<br>LIQUID | MECH PRESS<br>OPTICAL PROBE | SWITCH<br>OPTICAL COUPLER |
| 7 | PUMP | | | |
| | OUTLET | PRESS | MECH | SWITCH |
| | CASE | LEAKAGE FLOW<br>TEMP | MECH<br>MECH | SWITCH<br>SWITCH |
| | SYSTEM | LEAKAGE FLOW | MECH | SWITCH |
| 8 | RUDDER ACTUATOR | DIFF DISPL<br>QUIESCENT FLOW | MECH ELEC<br>MECH | POTENTIOMETER<br>SWITCH |
| | PEDALS | DIFF DISPL | MECH ELEC | POTENTIOMETER |
| 9 | DESSICANT DRIER (RESERVOIR) | MOISTURE | OPTICAL PROBE | OPTICAL COUPLER |
| 10 | STRUTS | PRESS<br>LIOUID LEVEL | MECH ELEC<br>OPTICAL PROBE | TRANSDUCER<br>OPTICAL COUPLER |

Figure B

# Schedule of estimated costs

| | | |
|---|---|---|
| • MAINTENANCE MAN HOURS: | Military/ Commercial | $30/Hr |
| • UNSCHEDULED DOWNTIME: LOSS OF OPERATIONAL READINESS: | Commercial | $7500/Hr |
| • HYDRAULIC FLUID. | Phosphate Ester | $17/Gal |
| | MIL-H-5606 | $3/Gal. |
| | MIL-H-83282 | $10/Gal |
| • PUMP: | New | $3500 |
| | Overhaul | $2000 |
| • SERVOVALVE: | New | $800 |
| | Overhaul | $400 |
| • RELIEF VALVE (SYSTEM): | New | $200 |
| | Overhaul | $100 |
| • ACCUMULATOR: | New | $400 |
| | Overhaul | $100 |
| • ACTUATOR | New | $8-15,000 |
| | Overhaul | $4-7,000 |
| • FILTER ELEMENTS: PRES, RET, C.D. | | $100 Ea. |
| FILTER MANIFOLDS: | Overhaul | $1000 Ea |
| • DESICCANT REPLACEMENT: | | $70-150 |
| • JET FUEL / GAL: | | $1/Gal. |
| (RESERVOIR NEW) | | $4-8000 |
| (RESERVOIR SEALS REPLACE) | | $2-4000 |
| ACTUATOR: | New | $3-7000 |
| | Overhaul | $1-2000 |
| • NEW HYDRAULIC LINES | | $500 |

Figure C

## EXAMPLE (SUMMARY)

**COMPONENT:** Pump

**PROBLEM:** Excessive Case Drain Leakage — High Temperature

**EFFECT:** Filters Contaminated and Overheated, Pump Seal Lost, Pump Lost, and System Lost. Fuel Loss (1 GPM Internal Leakage = 2000 Gal. Fuel/Year)

| Maintenance Man Hour Requirements | CORRECTIVE ACTION | | | | | |
|---|---|---|---|---|---|---|
| | PRIMARY Replace Pump | | SECONDARY Change Filter Elements | | SECONDARY Decontaminate System | |
| | WITHOUT THE IMACS™ SYSTEM | WITH THE IMACS SYSTEM | WITHOUT THE IMACS SYSTEM | WITH THE IMACS SYSTEM | WITHOUT THE IMACS SYSTEM | WITH THE IMACS SYSTEM |
| Diagnose Problem | .05 HRS | .08 HRS | — | — | — | — |
| Maintenance Labor A/C | 1 HR | 1 HR | 1 HR | AVOIDED | 6 HRS | AVOIDED |
| System Integrity Check | 1 HR | 1 HR | — | AVOIDED | — | AVOIDED |
| Component Effected | NEW PUMP | O.H. PUMP | N/A | AVOIDED | N/A | AVOIDED |
| Total Man Hours | 2.5 HRS | 2.1 HRS | 1 HR | — | 6 HRS | — |
| Aircraft Downtime | 8 HRS | 2 HRS | — | AVOIDED | — | AVOIDED |
| Flight Safety | YES | NO | YES | NO | YES | NO |
| Costs: Man Hours | $75 | $63 | $30 | AVOIDED | $180 | AVOIDED |
| Material | $3500 | $2000 | $100 | AVOIDED | $340 | AVOIDED |
| Downtime | $60,000 | $15,000 | — | AVOIDED | — | AVOIDED |
| Total Costs | $63,575 | $17,063 | $130 | 0 | $520 | 0 |

TRADE OFF COSTS:

Cost without the IMACS System $64,225    Cost with the IMACS System $17,063    Net Savings $47,162

Figure D

## ADDITIONAL COST FACTORS

- FUEL COSTS DUE TO INTERNAL SYSTEM LEAKAGE
- FUEL COSTS ATTRIBUTED TO SYSTEM TROUBLE SHOOTING
- LOGISTIC MAINTENANCE SUPPORT
- INVENTORY MANAGEMENT OF SPARE PARTS
- SCHEDULING AND UTILIZATION OF PEOPLE AND AUXILIARY EQUIPMENT
- OPPORTUNISTIC AND CONVENIENCE RENEWAL OF WORN COMPONENTS
- LOGISTIC ENERGY CONSERVATION
- MAINTENANCE FORECASTING AND SCHEDULING
- REDUCED TURN-AROUND

FIGURE E

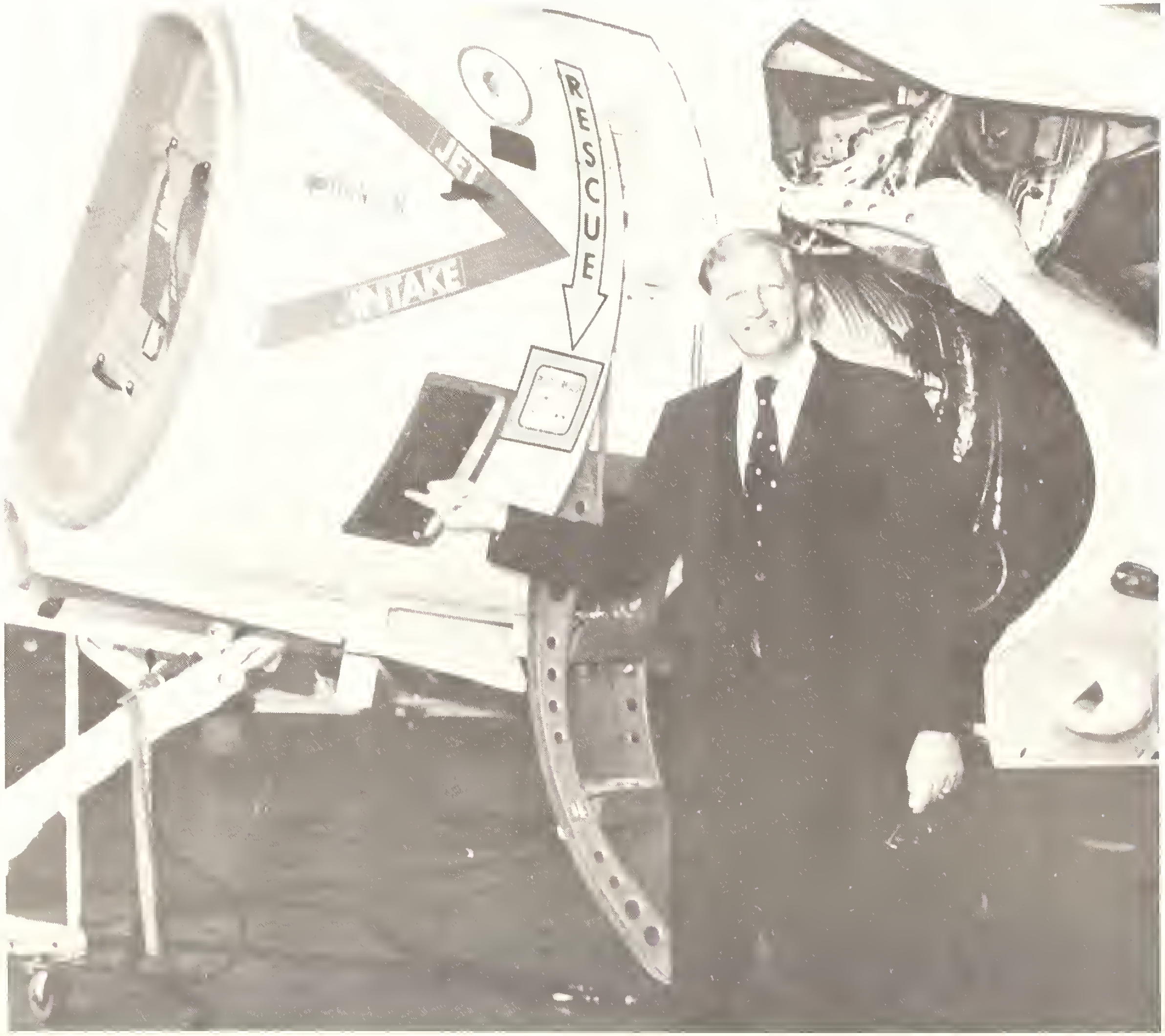



Figure F.

# ESTIMATE OF COST EFFECTIVENESS PER INDIVIDUAL SYSTEM

## CORRECTIVE ACTION

| Maintenance Man Hour Requirements | PRIMARY Cycle & Bleed System | | SECONDARY Replace Pump | | SECONDARY Replace Lines | | SECONDARY Decontaminate System | | SECONDARY Replace Fluid | |
|---|---|---|---|---|---|---|---|---|---|---|
| | WITHOUT THE IMACS™ SYSTEM | WITH THE IMACS SYSTEM | WITHOUT THE IMACS SYSTEM | WITH THE IMACS SYSTEM | WITHOUT THE IMACS SYSTEM | WITH THE IMACS SYSTEM | WITHOUT THE IMACS SYSTEM | WITH THE IMACS SYSTEM | WITHOUT THE IMACS SYSTEM | WITH THE IMACS SYSTEM |
| Diagnose Problem | NO PRO-VISION | .08 HRS | 0.5 HRS | — | 0.5 HRS | — | 1 HR | — | .08 HRS. | — |
| Maintenance Labor A/C | 1 HR | 1 HR | 1 HR | AVOIDED | 2 HRS | AVOIDED | 6 HRS | AVOIDED | 0.5 HRS | AVOIDED |
| System Integrity Check | — | 08 HRS | 1 HR | AVOIDED | 1 HR | AVOIDED | — | AVOIDED | — | AVOIDED |
| Component Effected | N/A | N/A | NEW PUMP | AVOIDED | INSTALL NEW LINES | AVOIDED | N/A | AVOIDED | ADD FLUID | AVOIDED |
| Total Man Hours | 1 HR | 1 HR | 2.5 HRS | AVOIDED | 3.5 HRS | AVOIDED | 7 HRS | AVOIDED | 0.5 HRS | AVOIDED |
| Aircraft Downtime | 5 HRS | 1 HR | — | AVOIDED | — | AVOIDED | — | AVOIDED | — | AVOIDED |
| Flight Safety | YES | NO | YES | NO | YES | NO | YES | NO | YES | NO |
| Costs: Man Hours | | | | AVOIDED | | AVOIDED | | AVOIDED | | AVOIDED |
| Material | | | | AVOIDED | | AVOIDED | | AVOIDED | | AVOIDED |
| Downtime | | | | AVOIDED | | AVOIDED | | AVOIDED | | AVOIDED |
| Total Costs | | | | 0 | | 0 | | 0 | | |

Cost without the IMACS System

Cost with the IMACS System

Net Savings

# ESTIMATE OF COST EFFECTIVENESS PER INDIVIDUAL SYSTEM

## CORRECTIVE ACTION

| Maintenance Man Hour Requirements | PRIMARY Replace Pump | | SECONDARY Change Filter Elements | | SECONDARY Decontaminate System | | SECONDARY Replace Fluid | |
|---|---|---|---|---|---|---|---|---|
| | WITHOUT THE IMACS™ SYSTEM | WITH THE IMACS SYSTEM | WITHOUT THE IMACS SYSTEM | WITH THE IMACS SYSTEM | WITHOUT THE IMACS SYSTEM | WITH THE IMACS SYSTEM | WITHOUT THE IMACS SYSTEM | WITH THE IMACS SYSTEM |
| Diagnose Problem | .05 HRS | .08 HRS | — | — | N/A | — | .08 HRS | — |
| Maintenance Labor A/C | 1 HR | 1 HR | 1 HR | AVOIDED | 6 HRS | AVOIDED | 0.5 HRS | AVOIDED |
| System Integrity Check | 1 HR | 1 HR | — | AVOIDED | — | AVOIDED | — | AVOIDED |
| Component Effected | NEW PUMP | O.H. PUMP | NEW ELEMENTS | AVOIDED | N/A | AVOIDED | ADD FLUID | AVOIDED |
| Total Man Hours | 2.5 HRS | 2.1 HRS | 1 HR | — | 6 HRS | — | 0.5 HRS | |
| Aircraft Downtime | 8 HRS | 2 HRS | — | AVOIDED | — | AVOIDED | — | AVOIDED |
| Flight Safety | YES | NO | YES | NO | YES | NO | YES | NO |
| Costs: Man Hours | | | | AVOIDED | | AVOIDED | | AVOIDED |
| Material | | | | AVOIDED | | AVOIDED | | AVOIDED |
| Downtime | | | | AVOIDED | | AVOIDED | | AVOIDED |
| Total Costs | | | | 0 | | 0 | | 0 |

Cost without the IMACS System　　Cost with the IMACS System　　Net Savings

# ESTIMATE OF COST EFFECTIVENESS PER INDIVIDUAL SYSTEM

## CORRECTIVE ACTION

| Maintenance Man Hour Requirements | PRIMARY Replace Filter Elements | | SECONDARY Replace Pumps | | SECONDARY Decontaminate System | | SECONDARY Replace Fluid | |
|---|---|---|---|---|---|---|---|---|
| | WITHOUT THE IMACS™ SYSTEM | WITH THE IMACS SYSTEM | WITHOUT THE IMACS SYSTEM | WITH THE IMACS SYSTEM | WITHOUT THE IMACS SYSTEM | WITH THE IMACS SYSTEM | WITHOUT THE IMACS SYSTEM | WITH THE IMACS SYSTEM |
| Diagnose Problem | .08 HRS | .08 HRS | 0.5 HRS | AVOIDED | 1 HR | .08 HRS | .08 HRS | AVOIDED |
| Maintenance Labor A/C | 1 HR | 1 HR | 1 HR | AVOIDED | 6 HRS | AVOIDED | 0.5 HRS | AVOIDED |
| System Integrity Check | — | .08 HRS | 1 HR | AVOIDED | — | AVOIDED | — | AVOIDED |
| Component Effected | NEW ELEMENTS | NEW ELEMENTS | NEW PUMP OR OVERHAUL | AVOIDED | N/A | AVOIDED | ADD FLUID | AVOIDED |
| Total Man Hours | 1 HR | 1 HR | 2.5 HRS | AVOIDED | 7 HRS | .08 HRS | 0.5 HRS | AVOIDED |
| Aircraft Downtime | 8 HRS | 1 HR | — | AVOIDED | — | AVOIDED | — | AVOIDED |
| Flight Safety | NO | NO | YES | NO | YES | NO | YES | NO |
| Costs: Man Hours | | | | | | AVOIDED | | |
| Material | | | | | | AVOIDED | | |
| Downtime | | | | | | AVOIDED | | |
| Total Costs | | | | | | | | |

Cost without the IMACS System

Cost with the IMACS System

Net Savings

# ESTIMATE OF COST EFFECTIVENESS PER INDIVIDUAL SYSTEM

## CORRECTIVE ACTION

| Maintenance Man Hour Requirements | PRIMARY Replace Filter | | SECONDARY Replace Servo-Valve & Actuator | | SECONDARY Replace Pump | | SECONDARY Decontaminate System | | SECONDARY Refill System | |
|---|---|---|---|---|---|---|---|---|---|---|
| | WITHOUT THE IMACS™ SYSTEM | WITH THE IMACS SYSTEM | WITHOUT THE IMACS SYSTEM | WITH THE IMACS SYSTEM | WITHOUT THE IMACS SYSTEM | WITH THE IMACS SYSTEM | WITHOUT THE IMACS SYSTEM | WITH THE IMACS SYSTEM | WITHOUT THE IMACS SYSTEM | WITH THE IMACS SYSTEM |
| Diagnose Problem | .08 HRS | 08 HRS | 2 HRS | — | 0.5 HRS | .08 HRS | 1 HR | — | .08 HRS | — |
| Maintenance Labor A/C | 1 HR | 1 HR | 8 HRS | AVOIDED | 1 HR | AVOIDED | 6 HRS | AVOIDED | 0.5 HRS | AVOIDED |
| System Integrity Check | — | 08 HRS | 1 HR | AVOIDED | 1 HR | AVOIDED | — | AVOIDED | — | AVOIDED |
| Component Effected | NEW ELEMENT | NEW ELEMENT | O H SERVO-ACTUATOR | AVOIDED | NEW PUMP | AVOIDED | N/A | AVOIDED | | AVOIDED |
| Total Man Hours | 1 HR | 1 HR | 11 HRS | — | 2.5 HRS | 0.8 HRS | 7 HRS | — | 0.5 HRS | — |
| Aircraft Downtime | 8 HRS | 1 HR | — | AVOIDED | — | AVOIDED | — | AVOIDED | — | AVOIDED |
| Flight Safety | YES | NO | YES | NO | YES | NO | YES | NO | YES | NO |
| Costs: Man Hours | | | | AVOIDED | | AVOIDED | | AVOIDED | | AVOIDED |
| Material | | | | AVOIDED | | AVOIDED | | AVOIDED | | AVOIDED |
| Downtime | | | | AVOIDED | | AVOIDED | | AVOIDED | | AVOIDED |
| Total Costs | | | | 0 | | 0 | | 0 | | 0 |

Cost without the IMACS System

Cost with the IMACS System

Net Savings

# ESTIMATE OF COST EFFECTIVENESS PER INDIVIDUAL SYSTEM

## CORRECTIVE ACTION

| Maintenance Man Hour Requirements | PRIMARY Replace System Relief Valve | | SECONDARY Replace Pump | | SECONDARY Replace Seal in Reservoir | | SECONDARY Overhaul Filter Manifold | |
|---|---|---|---|---|---|---|---|---|
| | WITHOUT THE IMACS™ SYSTEM | WITH THE IMACS SYSTEM | WITHOUT THE IMACS SYSTEM | WITH THE IMACS SYSTEM | WITHOUT THE IMACS SYSTEM | WITH THE IMACS SYSTEM | WITHOUT THE IMACS SYSTEM | WITH THE IMACS SYSTEM |
| **Diagnose Problem** | 1 HR | .08 HRS | 0.5 HRS | .08 HRS | 1 HR | AVOIDED | 1 HR | AVOIDED |
| **Maintenance Labor A/C** | 2 HRS | 2 HRS | 1 HR | AVOIDED | 4 HRS | AVOIDED | 1 HR | AVOIDED |
| **System Integrity Check** | 1 HR | .08 HRS | 1 HR | AVOIDED | 1 HR | AVOIDED | 1 HR | AVOIDED |
| **Component Effected** | NEW R/V | NEW R/V | O.H. PUMP | AVOIDED | O.H. RESER-VOIR | AVOIDED | OVERHAUL MANIFOLD | AVOIDED |
| **Total Man Hours** | 4 HRS | 2 HRS | 2.5 HRS | .08 HRS | 6 HRS | AVOIDED | 3 HRS | AVOIDED |
| **Aircraft Downtime** | 8 HRS | 2 HRS | — | AVOIDED | — | AVOIDED | — | AVOIDED |
| **Flight Safety** | YES | NO | YES | NO | YES | NO | YES | NO |
| **Costs: Man Hours** | | | | AVOIDED | | AVOIDED | | AVOIDED |
| **Material** | | | | AVOIDED | | AVOIDED | | AVOIDED |
| **Downtime** | | | | AVOIDED | | AVOIDED | | AVOIDED |
| **Total Costs** | | | | 0 | | 0 | | 0 |

Cost without the IMACS System

Cost with the IMACS System

Net Savings

# ESTIMATE OF COST EFFECTIVENESS PER INDIVIDUAL SYSTEM

## CORRECTIVE ACTION

| Maintenance Man Hour Requirements | PRIMARY Replace Accumulator | | SECONDARY Replace Pump | | SECONDARY Replace Lines | | SECONDARY Cycle & Bleed System | | SECONDARY Decontaminate System | |
|---|---|---|---|---|---|---|---|---|---|---|
| | WITHOUT THE IMACS™ SYSTEM | WITH THE IMACS SYSTEM | WITHOUT THE IMACS SYSTEM | WITH THE IMACS SYSTEM | WITHOUT THE IMACS SYSTEM | WITH THE IMACS SYSTEM | WITHOUT THE IMACS SYSTEM | WITH THE IMACS SYSTEM | WITHOUT THE IMACS SYSTEM | WITH THE IMACS SYSTEM |
| Diagnose Problem | 1 HR | 08 HRS | 0.5 HRS | — | 0 5 HRS | — | NO PRO-VISION | — | 1 HR | — |
| Maintenance Labor A/C | 2 HRS | 2 HRS | 1 HR | AVOIDED | 2 HRS | AVOIDED | 1 HR | AVOIDED | 6 HRS | AVOIDED |
| System Integrity Check | 1 HR | .08 HRS | 1 HR | AVOIDED | 1 HR | AVOIDED | — | AVOIDED | — | AVOIDED |
| Component Effected | O H ACCUMULATOR | | NEW PUMP | AVOIDED | INSTALL NEW LINES | AVOIDED | N/A | AVOIDED | N/A | AVOIDED |
| Total Man Hours | 4 HRS | 2 HRS | 2 5 HRS | AVOIDED | 3.5 HRS | AVOIDED | 1 HR | AVOIDED | 7 HRS | AVOIDED |
| Aircraft Downtime | 5 HRS | SCHED MAINT | — | AVOIDED | — | AVOIDED | — | AVOIDED | — | AVOIDED |
| Flight Safety | YES | NO | YES | NO | YES | NO | YES | NO | YES | NO |
| Costs: Man Hours | | | | AVOIDED | | AVOIDED | | AVOIDED | | AVOIDED |
| Material | | | | AVOIDED | | AVOIDED | | AVOIDED | | AVOIDED |
| Downtime | | | | AVOIDED | | AVOIDED | | AVOIDED | | AVOIDED |
| Total Costs | | | | 0 | | 0 | | 0 | | 0 |

Cost without the IMACS System　　　Cost with the IMACS System　　　Net Savings

# ESTIMATE OF COST EFFECTIVENESS PER INDIVIDUAL SYSTEM

## CORRECTIVE ACTION

| Maintenance Man Hour Requirements | PRIMARY Check Charge | | PRIMARY Check for Moisture | | SECONDARY Recharge Bottle | | SECONDARY Purge System of Moisture | |
|---|---|---|---|---|---|---|---|---|
| | WITHOUT THE IMACS™ SYSTEM | WITH THE IMACS SYSTEM | WITHOUT THE IMACS SYSTEM | WITH THE IMACS SYSTEM | WITHOUT THE IMACS SYSTEM | WITH THE IMACS SYSTEM | WITHOUT THE IMACS SYSTEM | WITH THE IMACS SYSTEM |
| Diagnose Problem | .08 HRS | .08 HRS | NO PRO-VISION | .08 HRS | — | — | — | — |
| Maintenance Labor A/C | — | AVOIDED | — | AVOIDED | 1 HR | 1 HR | 4 HRS | 4 HRS |
| System Integrity Check | N/A | N/A | NO PRO-VISION | AVOIDED | 24 HR DELAY REQ'D | LESS THAN 24 HR DELAY | NO PROVI-SION | .08 HRS |
| Component Effected | N/A | N/A | N/A | AVOIDED | N/A | N/A | LINES, VALVES, BOTTLES | LINES, VALVES, BOTTLES |
| Total Man Hours | — | — | — | AVOIDED | 1 HR | 1 HR | 4 HRS | 4 HRS |
| Aircraft Downtime* | 4 HRS | 4 HRS | — | N/A | — | — | — | — |
| Flight Safety | YES | YES | YES | YES | YES | YES | YES | YES |
| Costs: Man Hours | | | NO PRO-VISION | AVOIDED | | | | |
| Material | | | NO PRO-VISION | AVOIDED | | | | |
| Downtime | | | NO PRO-VISION | AVOIDED | | | | |
| Total Costs | | | NO PRO-VISION | 0 | | | | |

Cost without the IMACS System

Cost with the IMACS System

Net Savings

# ESTIMATE OF COST EFFECTIVENESS PER INDIVIDUAL SYSTEM

## CORRECTIVE ACTION

| Maintenance Man Hour Requirements | PRIMARY Replace Pump | | SECONDARY Change Filter Elements | | SECONDARY Decontaminate System | | SECONDARY Replace Fluid | |
|---|---|---|---|---|---|---|---|---|
| | WITHOUT THE IMACS™ SYSTEM | WITH THE IMACS SYSTEM | WITHOUT THE IMACS SYSTEM | WITH THE IMACS SYSTEM | WITHOUT THE IMACS SYSTEM | WITH THE IMACS SYSTEM | WITHOUT THE IMACS SYSTEM | WITH THE IMACS SYSTEM |
| Diagnose Problem | 0.5 HRS | .08 HRS | — | — | 1 HR | — | .08 HR | AVOIDED |
| Maintenance Labor A/C | 1 HR | 1 HR | 1 HR | AVOIDED | 6 HRS | AVOIDED | 0.5 HRS | AVOIDED |
| System Integrity Check | 1 HR | 1 HR | — | AVOIDED | — | AVOIDED | — | AVOIDED |
| Component Effected | NEW PUMP | O.H. PUMP | NEW ELEMENT | AVOIDED | N/A | AVOIDED | ADD FLUID | AVOIDED |
| Total Man Hours | 2.5 HRS | 1 HR | 1 HR | AVOIDED | 7 HRS | — | 0.5 HRS | AVOIDED |
| Aircraft Downtime | 8 HRS | AVOIDED SCHED. MAINT. | — | AVOIDED | — | AVOIDED | — | AVOIDED |
| Flight Safety | YES | NO | YES | NO | YES | NO | YES | NO |
| Costs: Man Hours | | | | AVOIDED | | AVOIDED | | AVOIDED |
| Material | | | | AVOIDED | | AVOIDED | | AVOIDED |
| Downtime | | | | AVOIDED | | AVOIDED | | AVOIDED |
| Total Costs | | | | 0 | | 0 | | 0 |

Cost without the IMACS System Cost with the IMACS System Net Savings

# ESTIMATE OF COST EFFECTIVENESS PER INDIVIDUAL SYSTEM

## CORRECTIVE ACTION

| Maintenance Man Hour Requirements | PRIMARY Replace Leaking Actuator | | PRIMARY Re-rig the Control Surface | |
|---|---|---|---|---|
| | WITHOUT THE IMACS™ SYSTEM | WITH THE IMACS SYSTEM | WITHOUT THE IMACS SYSTEM | WITH THE IMACS SYSTEM |
| Diagnose Problem | 2 HRS* | .08 HRS | N/A | .08 HRS |
| Maintenance Labor A/C | 8 HRS | 8 HRS | 4 HRS | 4 HRS |
| System Integrity Check | 2 HRS | .08 HRS | 1 HR | .08 HRS |
| Component Effected | OVERHAUL SERVOACTUATOR | | N/A | N/A |
| Total Man Hours | 12 HRS | 8 HRS | 5 HRS | 4 HRS |
| Aircraft Downtime | 4 HRS. | SCHED. MAINT. | — | SCHED. MAINT. |
| Flight Safety | YES | YES | YES | YES |
| Costs: Man Hours | | | | |
| Material | | | | |
| Downtime | | | | |
| Total Costs | | | | |

*Assume at least two servoactuators have to be removed and tested to find

Cost without the IMACS System

Cost with the IMACS System

Net Savings

# ESTIMATE OF COST EFFECTIVENESS PER INDIVIDUAL SYSTEM

## CORRECTIVE ACTION

| Maintenance Man Hour Requirements | PRIMARY Replace Leaking Actuator | | PRIMARY Jack A/C and Swing Main Gear | |
|---|---|---|---|---|
| | WITHOUT THE IMACS™ SYSTEM | WITH THE IMACS SYSTEM | WITHOUT THE IMACS SYSTEM | WITH THE IMACS SYSTEM |
| Diagnose Problem | 2 HRS | .08 HRS | 2 HRS | .08 HRS |
| Maintenance Labor A/C | 4 HRS | 4 HRS | 8 HRS | 8 HRS |
| System Integrity Check | 1 HR | .08 HRS | 1 HRS | .08 HRS |
| Component Effected | REPLACE ACTUATOR | REPLACE ACTUATOR | N/A | N/A |
| Total Man Hours | 7 HRS | 4 HRS | 9 HRS | 8 HRS |
| Aircraft Downtime | 4 HRS | SCHED. MAINT. | 8 HRS | SCHED. MAINT. |
| Flight Safety | YES | YES | YES | YES |
| Costs: Man Hours | | | | |
| Material | | | | |
| Downtime | | | | |
| Total Costs | | | | |

Cost without the IMACS System

Cost with the IMACS System

Net Savings

# ESTIMATE OF COST EFFECTIVENESS PER INDIVIDUAL SYSTEM

## CORRECTIVE ACTION

| Maintenance Man Hour Requirements | PRIMARY Replace Desiccant | | SECONDARY Remove & Test Servoactuators | | SECONDARY Replace Pump | | SECONDARY Decontaminate System | |
|---|---|---|---|---|---|---|---|---|
| | WITHOUT THE IMACS™ SYSTEM | WITH THE IMACS SYSTEM | WITHOUT THE IMACS SYSTEM | WITH THE IMACS SYSTEM | WITHOUT THE IMACS SYSTEM | WITH THE IMACS SYSTEM | WITHOUT THE IMACS SYSTEM | WITH THE IMACS SYSTEM |
| Diagnose Problem | 0.5 HRS | .08 HRS | 1 HR | — | 0.5 HRS | — | 1 HR | — |
| Maintenance Labor A/C | 2 HRS | 2 HRS | 4 HRS | AVOIDED | 1 HR | AVOIDED | 6 HRS | AVOIDED |
| System Integrity Check | 1 HR | .08 HRS | 1 HR | AVOIDED | 1 HR | AVOIDED | — | AVOIDED |
| Component Effected | NEW DESICCANT | NEW DESICCANT | REPLACE ACTUATOR | AVOIDED | REPLACE PUMP | AVOIDED | N/A | N/A |
| Total Man Hours | 3.5 HRS | 2 HRS | 6 HRS | — | 2.5 HRS | — | 7 HRS | — |
| Aircraft Downtime | 10 HRS | 1 HR | — | AVOIDED | — | AVOIDED | — | AVOIDED |
| Flight Safety | NO | NO | YES | NO | YES | NO | NO | NO |
| Costs: Man Hours | | | | AVOIDED | | AVOIDED | | AVOIDED |
| Material | | | | AVOIDED | | AVOIDED | | AVOIDED |
| Downtime | | | | AVOIDED | | AVOIDED | | AVOIDED |
| Total Costs | | | | 0 | | 0 | | 0 |

Cost without the IMACS System

Cost with the IMACS System

Net Savings

OPERATING EXPERIENCE WITH ADVANCED COMPUTER BASED SURVEILLANCE SYSTEM

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Abstract

For a large grass roots ethylene plant, a computer system was developed and commissioned to provide continuous surveillance to its critical unspared as well as general purpose machinery. This system was justified on the basis of minimizing unplanned shutdowns and overall improvements in equipment availability factors. Based on utilization of the system during the plant start-up and normal operation over the past year, the computer system is earning its anticipated credits. The data gathered by the computer system has been critical to (a) pinpointing the cause of events affecting machinery and (b) providing a basis for soundly engineered solutions. The following functional capabilities have been useful:

Dynamic storage of process and vibration data at once per second scan rate. When the train trips, this scanning is suspended to provide data before and after a trip for post-mortem analysis. This data helps to diagnose the cause of trips and minimize "downtime."

Monitor critical machinery parameters like vibrations, temperatures, hot alignment, horsepower, etc., provide integrated alarm annunciation to operators via CRT's in the Control Center. Current operating data are displayed and updated via user built profiles and schematics.

Alarm initiated data logging capability to capture data before and after an alarm. These data are used to analyze the cause of selected alarms.

On-line signature analysis of machinery vibrations with routine comparison of current frequency spectra with the user defined alarm and base line signatures. Vibrations with frequencies up to 100 kHz can be analyzed.

On-line monitoring of the performance of the compressors and steam turbines to evaluate performance deterioration and schedule washing as appropriate.

Monitor the high frequency signals (80 to 120 kHz) from pumps to detect incipient failures in anti-friction bearings and mechanical seals.

Predict alarms based on trends in data for incipient detection of slow deterioration. Provide alarms based on calculated variables.

Over 1500 instrument points are being monitored via three (3) satellites and four (4) multiplexers located throughout the plant. The satellites do all the input processing including limit checking for alarms. The historical data storage, data manipulation and man/machine interface requirements are handled by the host computer located in the Control Center.

# MECHANICAL AND AEROTHERMAL DIAGNOSTICS OF TURBOMACHINERY

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## ABSTRACT

To fully diagnose turbomachinery problems, both the mechanical and aerothermal parameters must be considered, since there is always a close interaction between them. The technique outlined in this paper deals with the complex aerothermal and mechanical problems in turbomachinery, and the interrelation of these parameters. Historically, the aerospace industry has been concerned with performance, while the petrochemical industry with reliability. Each of these industries looked at the two parameters exclusive of their interrelated relationships. The current trend is to use both mechanical and aerothermal parameters for monitoring and diagnostics.

This paper outlines techniques which can be used both for online diagnostics or offline problem solving. To determine deterioration in performance of a unit, it is necessary to first, determine whether the changes in performance are due to changes in ambient or process conditions, or whether these changes are due to an actual physical deterioration of the unit. Once it has been determined that the deterioration is due to physical deterioration of the system, the analysis must indicate the cause. This is done by examination of the various parameters and noting which of these parameters have caused the decay. Various groups of parameter changes indicate specific problems and these are correlated to diagnose problems.

## 1.0 INTRODUCTION

This paper describes the requirements and techniques for instrumentation, analysis and diagnostics for application on turbomachinery equipment.

To achieve effective monitoring and diagnostics of turbomachinery it is necessary to gather and analyze both the mechanical and aerothermal operating data from the machines. The instrumentation and diagnostics must also be custom tailored to suit the individual machines in the system, and also to meet the requirements of the end users. The reasons for this being that, there can be significant differences in machines of the same type or manufacturer, because of differences in installation and operation.

## 2.0 REQUIREMENTS FOR AN EFFECTIVE DIAGNOSTIC SYSTEM

The following is a list of requirements for a diagnostic system:

A. The system must produce diagnostic and failure prediction information in a timely manner before serious problems occur on the machines monitored.

B. When equipment shutdown becomes necessary, diagnostics must be precise enough to accomplish problem identification and rectification with minimal downtime.

C. The system should be useable and understood sufficiently by production personnel so that an engineer is not always necessary when urgent decisions need to be made.

D. The system should be simple and reliable and cause negligible downtime for repairs, routine calibration and checks.

E. The system must be cost effective, namely it should cost less to operate and maintain than the expenses resulting from loss of production and machinery repairs that would have resulted if the machinery was not under monitoring and predictive surveillance.

F. System flexibility to incorporate improvements in the state of the art is desirable.

G. System expansion capabilities to accept projected increases in installed machinery or increases in number of channels must be considered.

H. The use of excess capacity in a computer system available at the plant can result in considerable savings equipment costs. System components that mate with the existing computer system may, therefore, be a necessary prerequisite.

## 3.0 DIAGNOSTIC SYSTEM COMPONENTS & FUNCTIONS

A. Instrumentation and Instrumentation Mountings

B. Signal Conditioning and Amplifiers for Instrumentation

C. Data Transmission System: Cables, Telephone Linkup, or Microwave.

D. Data Integrity Checking, Data Selection, Data Normalization and Storage.

E. Baseline Generation and Comparison.

F. Problem Detection

G. Diagnostics Generation.

H. Prognosis Generation.

I. Onsite Display.

J. Systems for Curve Plotting, Documentation and Reporting

Figures 1A and 1B are schematic representations of typical systems.

## 4.0 DATA INPUTS

Obtaining good data inputs is a fundamental requirement, since any analysis system is only as good as the inputs to the system. A full audit of the various trains to be monitored must be made in order to obtain optimum instrumentation selection.

The factors that need to be considered are the instrument type, its measurement range, accuracy requirements and the operational environmental conditions. These factors must be carefully evaluated to select instruments of optimum function and cost to match the total requirements of the system. For instance, the frequency range of the vibration sensor should be adequate for monitoring and diagnostics and should match with the frequency range of analysis equipment. Sensors should be selected to operate reliably and accurately within the environmental conditions that prevail, for example, when used on high temperature turbine casings. Resistance temperature sensors with their higher accuracy and reliability as compared to thermocouples may be necessary for analysis accuracy and reliability.

Calibration of instrumentation should be conducted on a schedule established after reliability factors have been analyzed.

All data should be checked for validity and as to whether they are within reasonable limits. Data that is beyond predetermined limits should be discarded and flagged for investigation. An unreasonable result or analysis should set up a routine for identification of possible discrepant input data.

## 5.0 MONITORING/DIAGNOSTIC SYSTEM

It is essential that the instrumentation requirements be tailored to the requirements of the machine being monitored. However, the following instrumentation requirements should exist to cover the requirements for both vibration and aerothermal monitoring.

Any existing instrumentation should be used if found to be adequate. While there are advantages in the use of non-contacting sensors built into the machine for measurement of journal displacements, this instrumentation is often

HEALTH MONITORING SYSTEM

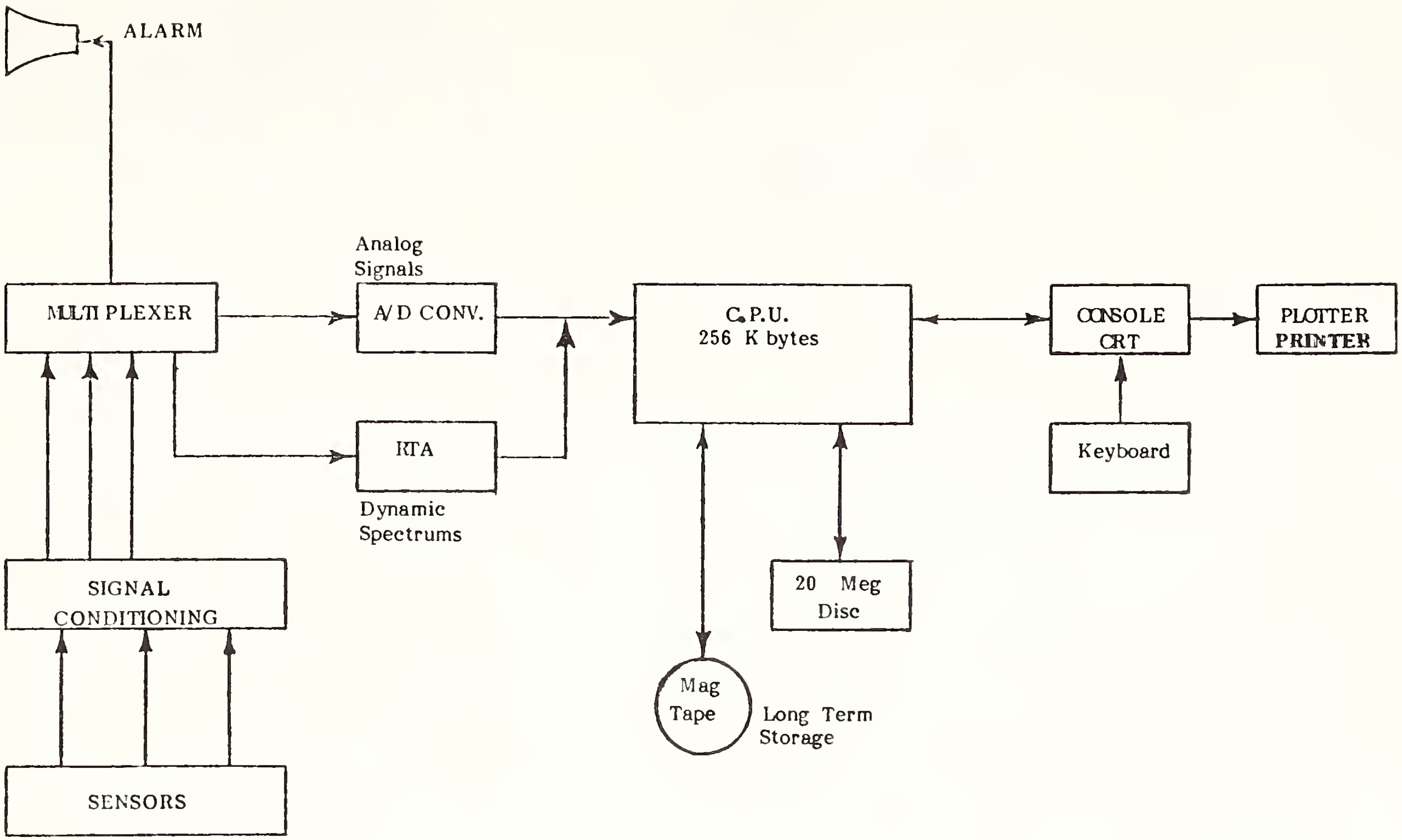



Figure 1A: Health Monitoring Systems

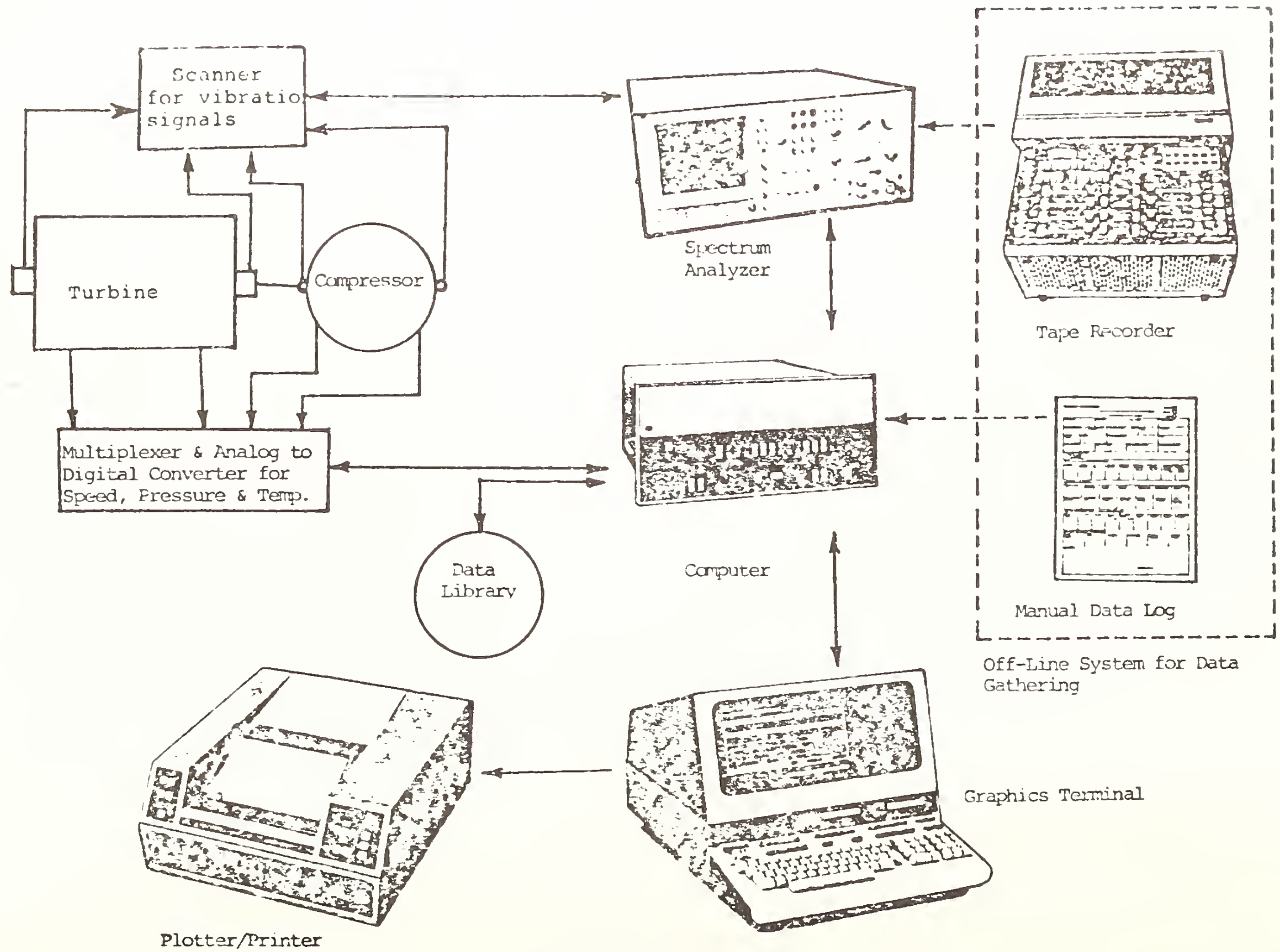



Figure 1B: A Typical Computerized System for Turbomachinery Monitoring & Diagnostics impossible to install in existing machinery. Suitably selected and located accelerometers can adequately cover the vibration monitoring requirements of machinery. Accelerometers are often an essential supplement to displacement sensors to cover the higher frequencies generated by gear mesh, blade passing, rubs and other conditions.

A. <u>Instrumentation Typical -Minimum Requirements for Each Machine</u>

Note: Locations and type of sensors would depend on the type of machine under consideration.

1. Accelerometer

   a. At machine inlet bearing case, vertical
   b. At the machine discharge bearing case, vertical
   c. At machine inlet bearing case, axial

2. Process Pressure

   a. Pressure at machine inlet
   b. Pressure at machine discharge

3. Process Temperature

   a. Temperature at machine inlet
   b. Temperature at machine discharge

4. Machine Speed

   a. Machine speed of all shafts

5. Thrust Bearing Temperature

   a. One thermocouple or resistance temperature element embedded in forward and aft thrust bearing

B. <u>Instrumentation-Desirable-Optional</u>

1. Non-contacting eddy current vibration displacement probe adjacent to:

   a. Inlet bearing, vertical
   b. Inlet bearing, horizontal
   c. Discharge bearing, vertical
   d. Discharge bearing, horizontal

2. Non-contacting eddy current gap sensing probe adjacent to:

   a. Forward face of thrust bearing collar
   b. Aft face of thrust bearing collar
   Note: The non-contacting sensor in its role of measurement of gap-D.C. voltage is sensitive to probe and driver temperature variations. Careful evaluation must hence be conducted of sensor type, its mounting and location, for this measurement.

3. Process flow measurement at inlet or discharge of machine

4. Radial bearing temperature-thermocouple or resistance temperature element embedded in each bearing, or temperature at lube oil discharge of each bearing.

5. Lube oil pressure and temperature

6. Dynamic pressure transducer at compressor discharge for indication of flow instability

Figures 2A and 2B shows possible instrument locations for an industrial gas turbine and centrifugal compressor.

## 6.0 CRITERIA FOR THE COLLECTION OF AEROTHERMAL DATA

Turbomachinery operating pressures, temperatures and speeds are very important parameters. Obtaining accurate pressures and temperatures will depend not only on the type and quality of the transducers selected, but also on their location in the gas path of the machine. These factors should be carefully evaluted. The accuracy of pressure and temperature measurements required will depend on the analysis and diagnostics that need to be performed. Figure 3 presents some criteria for selection of aerothermal instrumentation of pressure and temperature sensors for measurement of compressor efficiency. Note that the percentage accuracy requirements are more critical for temperature sensors than pressure sensors. The requirements are also dependent on the compressor pressure ratio.

## 7.0 VIBRATION INSTRUMENTATION SELECTION

The type of vibration instrumentation, its frequency ranges, its accuracy and its location within, or on the machine, must be carefully analyzed with respect to the diagnostics required to be achieved. Figure 4 presents guidelines on the selection of vibration sensors.

The displacement non-contacting eddy current sensor is most effective for monitoring and measuring vibrations near rotational and subrotational speeds.

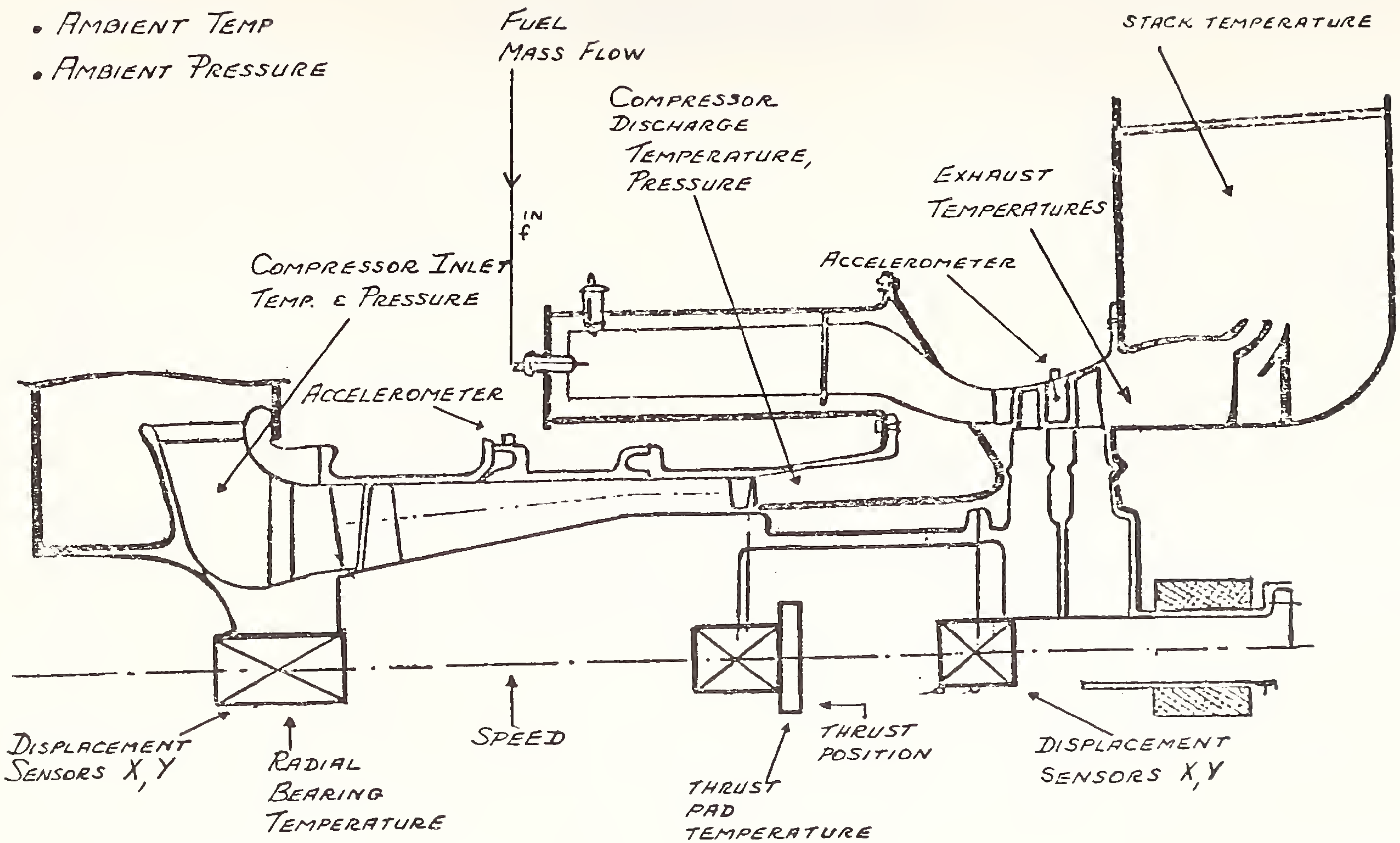



Figure 2A: INSTRUMENTATION FOR MONITORING & DIAGNOSTICS ON A GAS TURBINE ENGINE

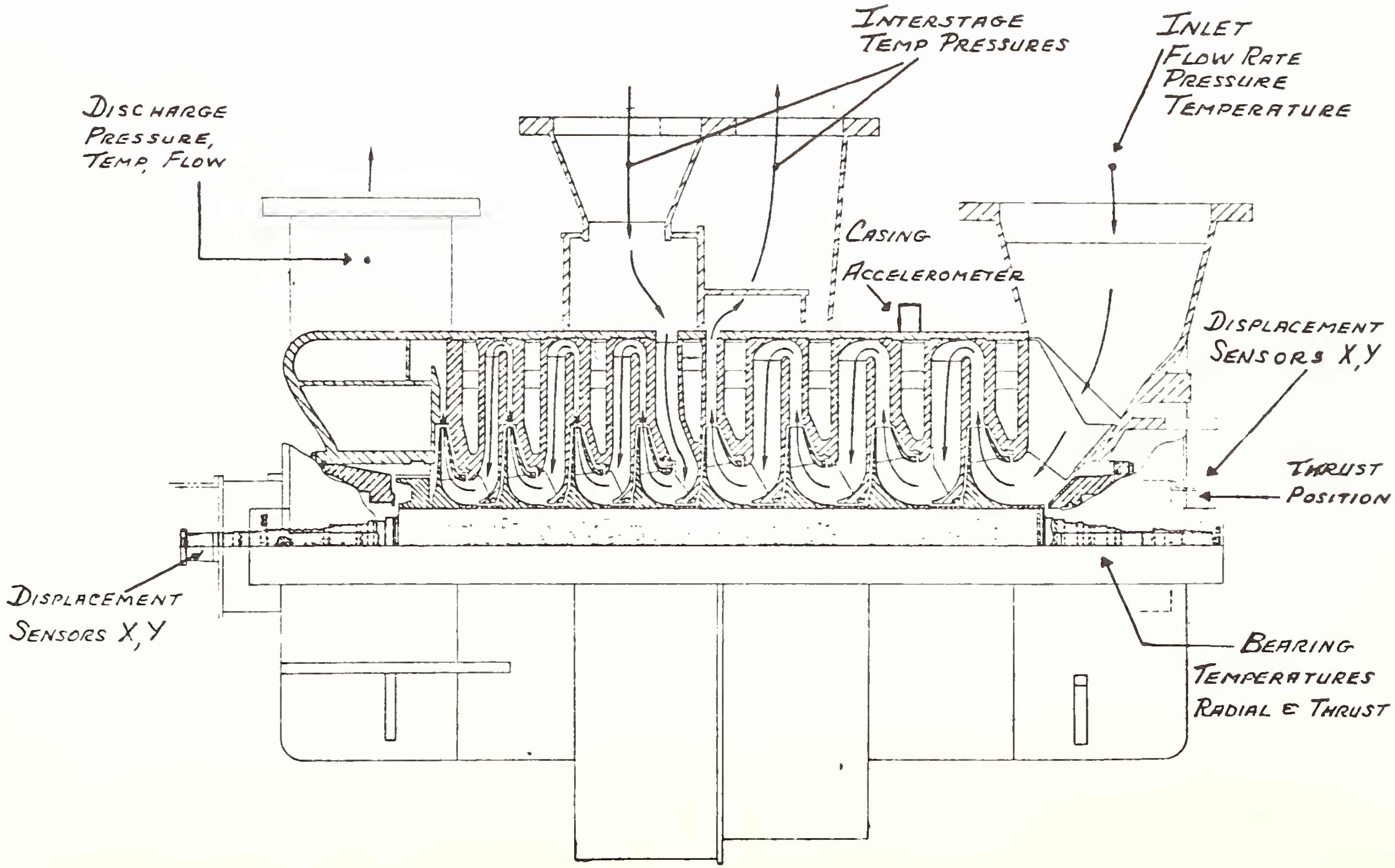



Figure 2B: INSTRUMENTATION
FOR MONITORING & DIAGNOSTICS
ON A CENTRIFUGAL COMPRESSOR

CRITERIA FOR SELECTION OF PRESSURE AND
TEMPERATURE SENSORS FOR COMPRESSOR EFFICIENCY MEASUREMENTS

| COMPRESSOR PRESSURE RATIO $P_2/P_1$ | $P_2$ SENSITIVITY PERCENT | $T_2$ SENSITIVITY PERCENT |
|---|---|---|
| 6 | .704 | .218 |
| 7 | .750 | .231 |
| 8 | .788 | .240 |
| 9 | .820 | .250 |
| 10 | .848 | .260 |
| 11 | .873 | .265 |
| 12 | .895 | .270 |
| 13 | .906 | .277 |
| 14 | .933 | .282 |
| 15 | .948 | .287 |
| 16 | .963 | .290 |

Tabulation showing percent changes in $P_2$ and $T_2$ needed to cause one-half percent change in air compressor efficiency. Ideal gas equations are used.

Figure 3: Criteria for Selection of Pressure and Temperature Sensors for Compressor Efficiency Measurements

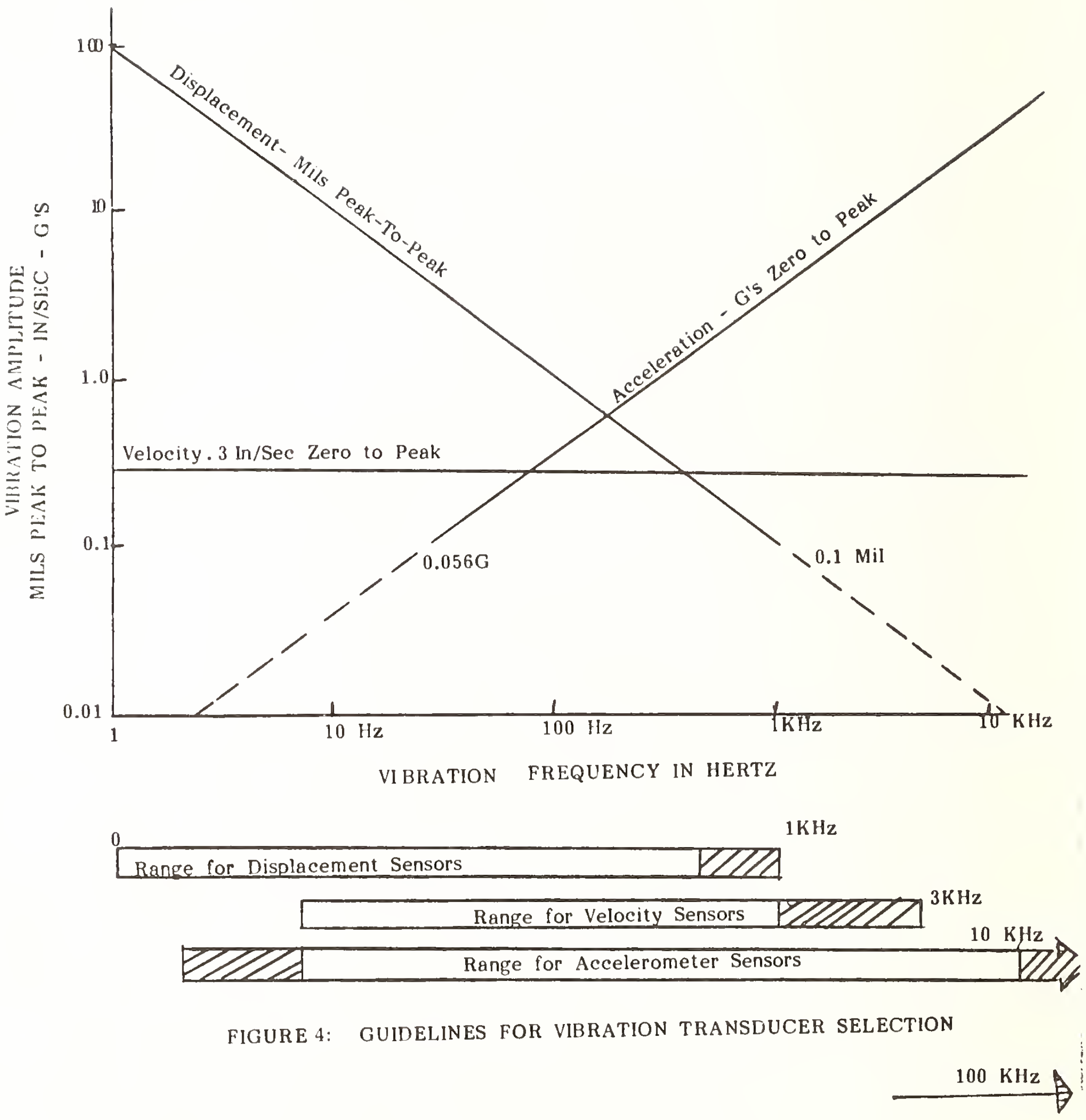



FIGURE 4: GUIDELINES FOR VIBRATION TRANSDUCER SELECTION

While the displacement sensor is capable of measuring vibration frequencies well above 2kHz, the amplitude of vibrational displacement levels that occur at frequencies above 1kHz are extremely small, and are usually lost or buried in the noise level of the readout system. The acceleration sensor is best suited for measurements at high frequencies, such as blade passing and gear meshing frequencies; however, the signals at once rotational speed are usually at low acceleration levels, and may be lost in the noise level of the measurement system monitoring. Low pass filtering and additional amplification stages may, therefore, be necessary to bring out the rotational speed signals when measurements are made with accelerometers.

Velocity sensors, because of their limited operational frequency range of usually from 10Hz to 2kHz, are not recommended for application in a diagnostic system for high speed machinery. Velocity sensors have moving elements and are subject to reliability problems at operational temperatures above 250°F. Gas turbine engine casing temperatures are usually in the 500°F level or above, hence sensor locations must be carefully examined for temperature levels. Accelerometers for these higher temperatures are more easily available than velocity sensors. At these elevated operational temperatures, high frequency accelerometers (20 kHz and above) are available from only a few selected manufacturers.

## 8.0 SELECTION OF SYSTEMS FOR ANALYSIS OF VIBRATION DATA

The overall vibration level on a machine is satisfactory for initial or rough check. However, when a machine has a seemingly acceptable overall level of vibration, there may be hidden under this level some small levels of vibrations at discrete frequencies that are known to be dangerous. An example of this is subsynchronous instabilities in a rotor system.

In the analysis of vibration data, there is most often the need to transform the data from the time domain to the frequency domain or, in other words, to obtain a spectrum analysis of the vibration. The original and inexpensive system to obtain this analysis is the tuneable swept filter analyzer. Because of inherent limitations of this system, this process, despite the use of automated sweep, is time consuming when analyzing low frequencies. When the spectrum data needs to be digitized for computer inputing,there are further limitations in capability of tuneable filter analysis systems.

Real Time Spectrum Analyzers using "Time Compression" or the "Fast Fourier Transform" (FFT) techniques, are extensively used for performing vibration spectrum analysis in computerized diagnostic systems. The FFT analyzers use digital signal processing, and hence are easier to integrate with the modern digital computer. FFT analyzers are often hybrids using micro-processors and FFT dedicated circuitry.

The FFT can be implemented in a computer using the FFT algorithm for obtaining a pure mathematical computation. While this computation is an error free process, its implementation in a digital computer can introduce several errors. To avoid these errors, it is essential to provide signal conditioning, upstream of the computer. Such signal conditioning minimizes the errors such as, aliasing and signal leakage introduced in sampling and digitizing the time domain. Such signal conditioning system will introduce considerable expense and complexity in effecting the mathematical FFT in a computer. The computerized FFT is also slower than a dedicated FFT analyzer. It also has limitations in frequency resolution. Hence, the use of a dedicated FFT analyzer is considered to be the most reliable and cost effective means for performing frequency spectrum analysis and plots in a computerized system for machinery diagnostics.

Careful analysis must be made of the type of spectrum analysis systems and the computational techniques used in vibrational analysis. There are several factors which must be considered, some of which are:

A. Frequency analysis ranges
B. Single or multi-channel analysis
C. Dynamic range
D. Accuracy of measurements necessary
E. Speed at which analysis are required to be made
F. System portability, especially if the analysis system is required for both lab and field use
G. Ease of integration with the host computer system

## 9.0 BASELINE FOR MACHINERY

### A. Mechanical Baseline

The vibration baseline for a machine can be defined as the normal or average operating condition of a machine. It can be represented on a vibration spectrum plot showing vibration frequency on the X-axis and vibration amplitude (peak-to-peak displacement, peak velocity, or peak acceleration) on the Y-axis. Since the vibration spectrum will be different at different positions, the spectrum must be associated with a specific measurement position or sensor location on the machine. When portable vibration measurement equipment is used, it is essential to ensure that the sensor is relocated at exactly the same point on the machine each time vibration readings are taken. Changes of baseline with machine speed and process conditions should be investigated and where necessary, baseline should be generated for set ranges of speeds and process conditions. When the operating vibration levels exceed the baseline levels beyond set values, an alert signal should be activated for investigation of this condition.

### B. Aerothermal Baseline

In addition to the vibration baseline spectrum, a machine also has an aerothermal performance baseline or its normal operating point on the aerothermal characteristic. Significant deviation of the operating point beyond its base point should generate alert signals.

When a compressor operates beyond its surge margin, a danger alert should be activated. A typical compressor characteristic is presented in Figure 5. Some of the other monitoring and operating outputs are loss in compressor flow, loss in pressure ratio and increase in operating fuel cost due to, for instance, operating at off design conditions or with a dirty compressor.

Since the aerothermal performance of compressors and turbines are very sensitive to inlet temperature and pressure variations, it is essential to normalize the aerothermal performance parameters such as, flow, speed, horsepower, etc. to standard day conditions. When these corrections to standard conditions are not applied, then a performance degradation may appear to occur when in fact it was a performance change resulting merely from ambient pressure and temperature changes. Some of the equations for obtaining correction to standard day conditions are given in Figure 6.

## 10.0 DATA TRENDING

The data received should first be corrected for sensing errors. This usually consists of sensor calibration correction.

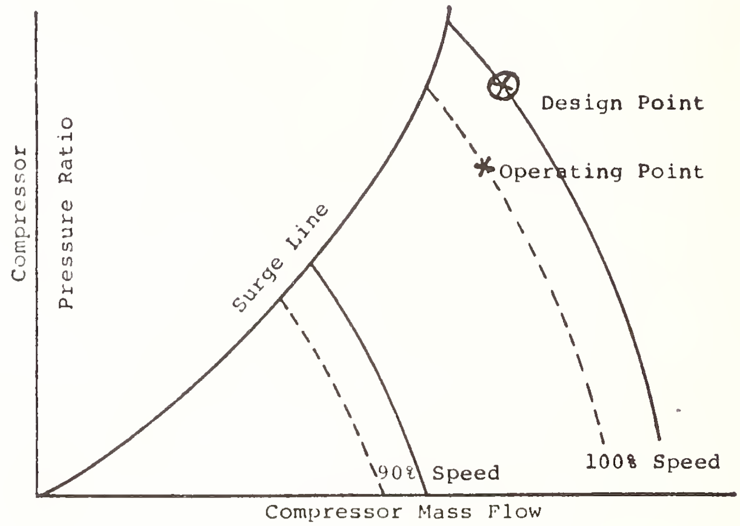
The trending technique essentially involves evaluating the slope of a curve derived from the received data. The slope of the curve is calculated for both a long-term trend, about 168 hours, and a short-term trend, based on the last 24 hours. If the short-term slope deviates from the long-term slope beyond a set limit, it means that the rate of deterioration is changed and the maintenance schedule will be affected. Thus, the program might take into account the biasing of the long-term slope by the short-term slope. Figure 7 shows a schematic of this type of trending. Numerous statistical techniques are available for trending.

Trended data is used to obtain predictions which would be helpful in the scheduling of maintenance. Referring to Figure 8, for example, it is possible to estimate when compressor cleaning will be necessary. This figure was prepared by recording the compressor exit temperature and pressure each day. These points are then joined and a dotted line is projected to predict when cleaning will be required. In this case, two parameters were monitored, but since their rates differed, the cleaning was based on the first parameter to reach the critical point. However, using a trend of both temperature and pressure provides a cross check on the validity of the diagnostics.

## 11.0 COMPRESSOR AEROTHERMAL CHARACTERISTICS & COMPRESSOR SURGE

Figure 9 shows a typical performance map for a centrifugal compressor, showing efficiency islands and constant aerodynamic speed lines. The total pressure ratio can be seen to change with flow and speed. Usually compressors are operated on a working line separated by some safety margin from the surge line.

AEROTHERMAL CONDITION MONITORING FOR COMPRESSORS




DATA INPUT

Ambient Pressure
Compressor Inlet Pressure
Compressor Discharge Pressure
Compressor Inlet Temperature
Compressor Discharge Temperature
Compressor Speed
Compressor Inlet or Discharge Flow if Available

DIAGNOSTICS OUTPUTS

Compressor Efficiency Lower than Design
Compressor Approaching Surge Conditions
Compressor Approaching CHOKE CONDITIONS
Dirty Compressor

CONDITION MONITORING OUTPUTS

Loss in Compressor Flow Through Put
Loss in Compressor Pressure Ratio
Fuel Cost Penalty
Projected Increase in Fuel Cost After One Month Operation
Surge Point Deterioration Trend and Anticipated Outage Date

Figure 5

FACTORS FOR CORRECTION TO STANDARD DAY TEMPERATURE & PRESSURE CONDITIONS

Assumed Standard Day Pressure ———— 14.7 psia
Assumed Standard Day Temperature ———— $60^{o}F$ ($520^{o}R$)

Conditions of Test

Inlet Temperature ———— $T_i$ $^{o}R$
Inlet Pressure ———— $P_i$ psia

| | | |
|---|---|---|
| Corrected Temperature | = | (Observed Temperature) $(520/T_1)$ |
| Corrected Pressure | = | (Observed Pressure) $(14.7/P_i)$ |
| Corrected Speed | = | (Observed Speed) $\sqrt{520/T_i}$ |
| Corrected Air Flow | = | (Observed Flow) $(14.7/P_i)$ $\sqrt{T_i/520}$ |
| Corrected Horsepower | = | (Observed Power) $(14.7/P_i)$ $\sqrt{520/T_i}$ |

Figure 6: Gas Turbine Aerothermal Performance Equations for Correction to Standard Day Conditions.

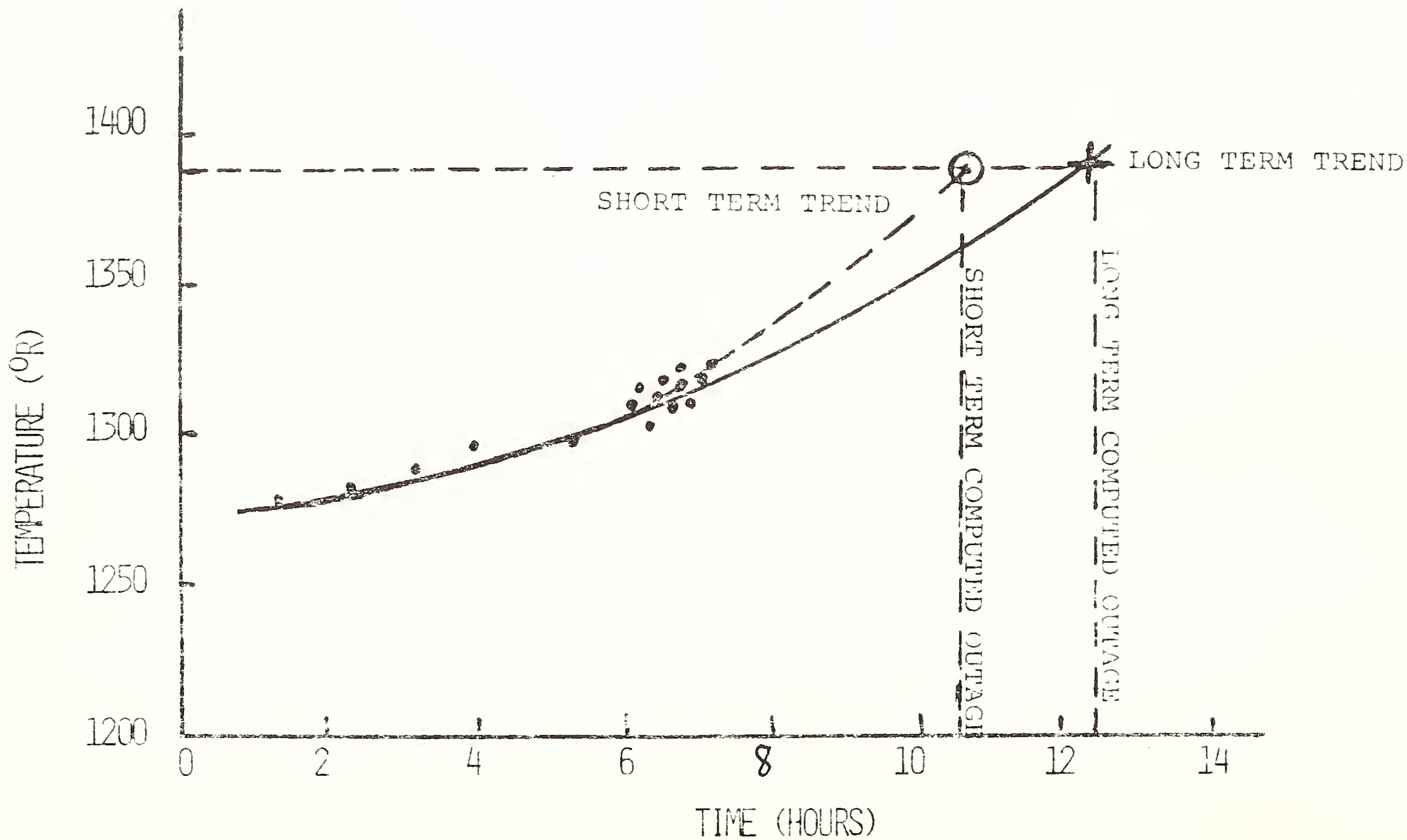



Figure 7: TEMPERATURE VERSUS EXPECTED OUTAGE TIME

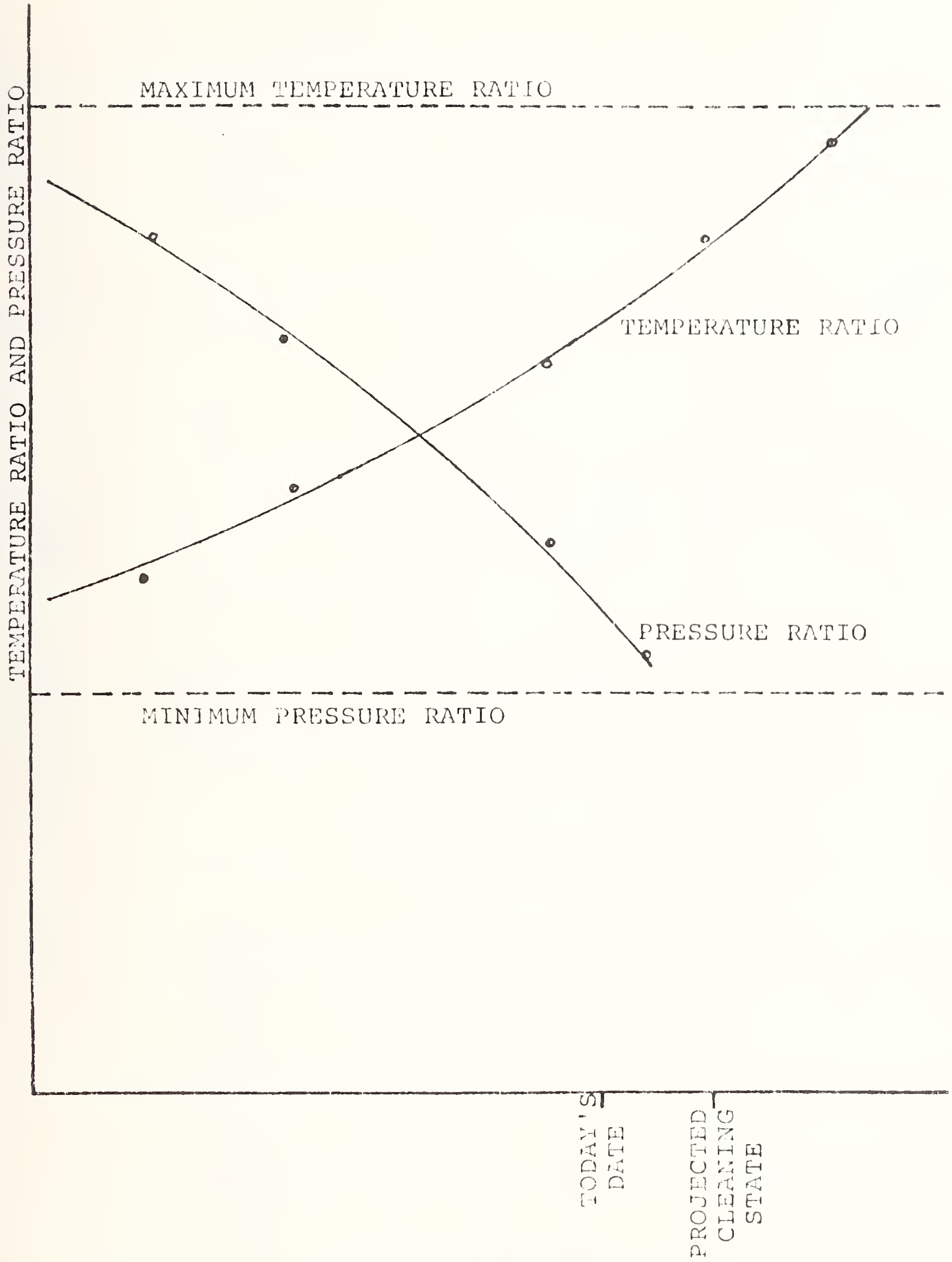



Figure 8: DATA TRENDING TO PREDICT MAINTENANCE SCHEDULES

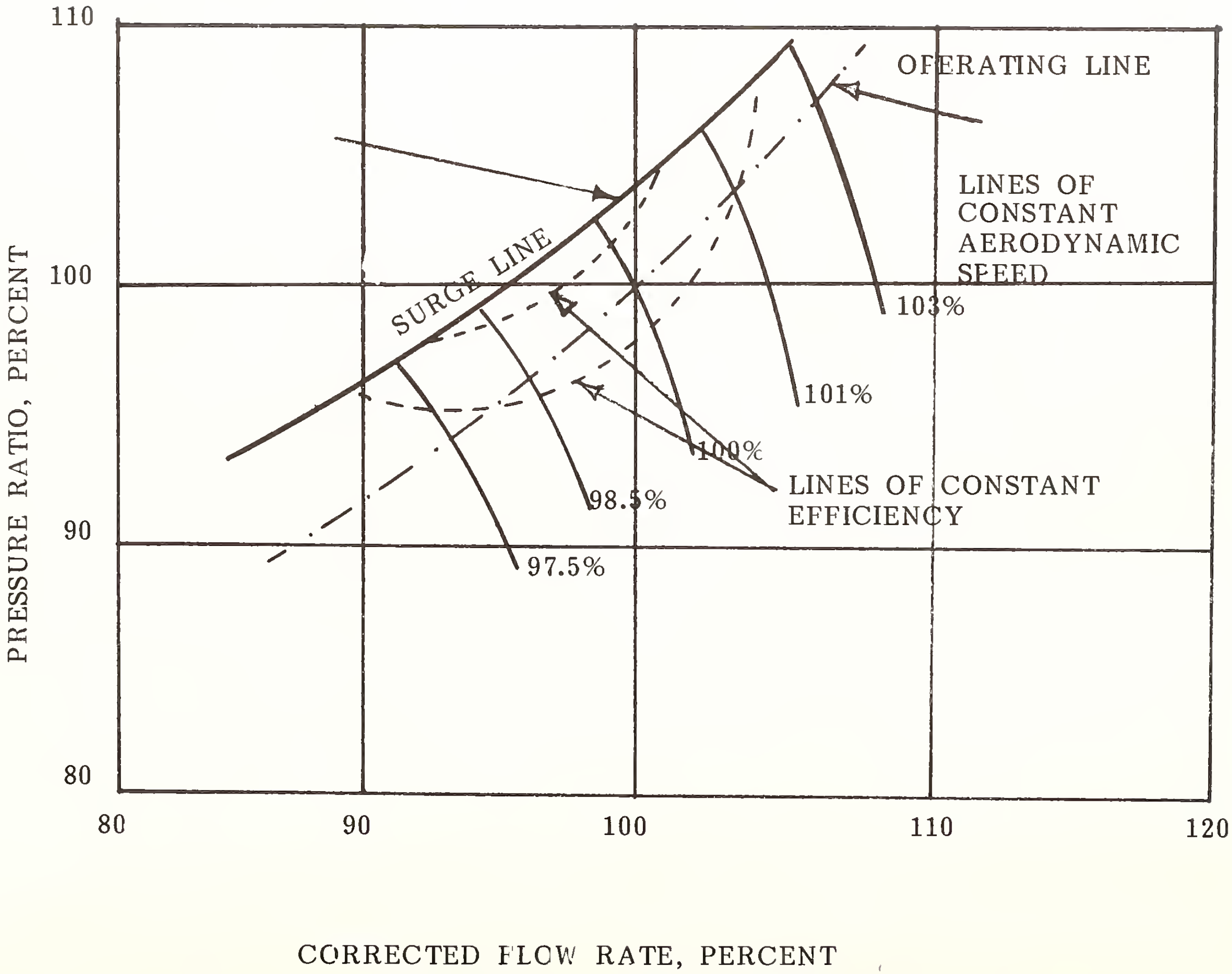



Figure 9. Typical Compressor Map

Compressor surge is essentially a situation of unstable operation and should, therefore, be avoided in both design and operation. Surge has been traditionally defined as the lower limit of stable operation of a compressor and involves the reversal of flow. This reversal of flow occurs because of some kind of aerodynamic instability within the system. Usually it is a part of the compressor that is the cause of the aerodynamic instability though it is possible that the system arrangement could be capable of magnifying this instability.

Usually, surge is linked with excessive vibration and an audible sound; yet, there have been cases in which surge problems which are not audible have caused failures.

## 12.0 DIAGNOSTICS

Problem evaluation in turbomachinery is complex, but with the aid of performance and mechanical signals, solutions can be found to diagnose various types of failures. This is done by using several inputs and a matrix. A sample of some of the problems are given in the next few sections.

### A. Compressor Analysis

Compressor analysis is done by monitoring the inlet and exit pressures and temperatures, the ambient pressure, vibration at each bearing, and the pressure and temperature of the lubrication system. Table I shows the effect various parameters have on some of the major problems encountered in a compressor. Monitoring these parameters allows the detection of the following problems:

1. Clogged air filter- A clogged air filter may be detected by noting an increase in the pressure drop through the filter

2. Compressor surging-Surge may be detected by noting a rapid increase in shaft vibration, along with a discharge pressure instability. If more than one stage is present, the probes located within the bleed air chambers are useful in locating the problem stage by checking for pressure fluctuations

3. Compressor Fouling-This is indicated by a decrease in pressure ratio and flow accompanied by an increase of exit temperature with time. The change in the temperature and pressure ratio tend to show a decrease in efficiency. If a change in vibration has occurred, the fouling is critical, since it indicates excessive build up of deposits on the rotor

4. Bearing failure-Symptoms of bearing trouble include a loss of lubrication pressure, an increase in the temperature difference across the bearing, and an increase in vibration. If oil whirl or other bearing instabilities are present, there will be a vibration at subsynchronous frequency

TABLE I

COMPRESSOR DIAGNOSTICS

| | $\eta_c$ | $P_2/P_1$ | $T_2/T_1$ | Compressor Fluid Mass Flow | Vibration | $\Delta T$ Bearing | Bearing Pressure | Bleed Chamber Pressure |
|---|---|---|---|---|---|---|---|---|
| Clogged Filter | — | ↓ | — | ↓ | — | — | — | — |
| Surge | ↓ | Variable | — | ↓ | Highly Fluctuating | ↑ | ↑ | Highly Fluctuating |
| Fouling | ↓ | ↓ | ↑ | ↓ | ↑ | — | — | — |
| Damaged Blade | ↓ | ↓ | ↑ | ↓ | ↑ | — | — | Highly Fluctuating |
| Bearing Failure | — | — | — | — | ↑ | ↑ | ↓ | — |

B. Combustor Analysis

In the combustor, the only two parameters which can be measured are fuel pressure and evenness of combustion noise. Turbine inlet temperatures are not usually meassured, due to very high temperatures and limited probe life. Table II shows the effect of various parameters on important functions of the combustor.

1. Plugged Nozzle-This is indicated by an increase in fuel pressure in conjuction with increased combustion uneveness. This is a common problem when residual fuels are used

2. Cracked or Detached Liner-This is indicated by an increase in an acoustic meter reading and a large spread in exhaust temperature

3. Combustor Inspection or Overhaul-This is based on equivalent engine hours which are based on number of starts, fuel and temperature. Figure 10 shows the effect of these parameters on the life of the unit. Note the strong effect that fuel and number of starts has on the life.

C. Turbine Analysis

To analyze a turbine, it is necessary to measure pressures and temperatures across the turbine, shaft vibration, and the temperature and pressure of the lubrication system. Table III shows the effect various parameters have on important functions of the turbines. Analysis of these parameters will aid in the prediction of the following:

1. Turbine Fouling-This is indicated by an increase in turbine exhaust temperature. Change in vibration amplitude will occur when fouling is excessive and causes rotor imbalance.

2. Damaged Turbine Blades-This results in a large vibration increase accompanied by an increase in the exhaust temperature.

3. Bowed Nozzle-The exhaust temperature will increase, and there may be an increase in turbine vibration.

4. Bearing Failure-The symptoms of bearing problems for a turbine are the same as for a compressor

5. Cooling Air Failure-Problems associated with the blade cooling system may be detected by an increase in the pressure drop in the cooling line.

TABLE II

COMBUSTOR DIAGNOSTICS

| | Fuel Pressure | Uneveness of Combustion sound | Exhaust Temperature Spread |
|---|---|---|---|
| Clogging of Fuel Nozzle | ↑ | ↑ | ↑ |
| Combustor Fouling | — | ↑ | ↓ |
| Crossover Tube Failure | — | — | — |
| Detached or Cracked Liner | — | ↑ | ↑ |

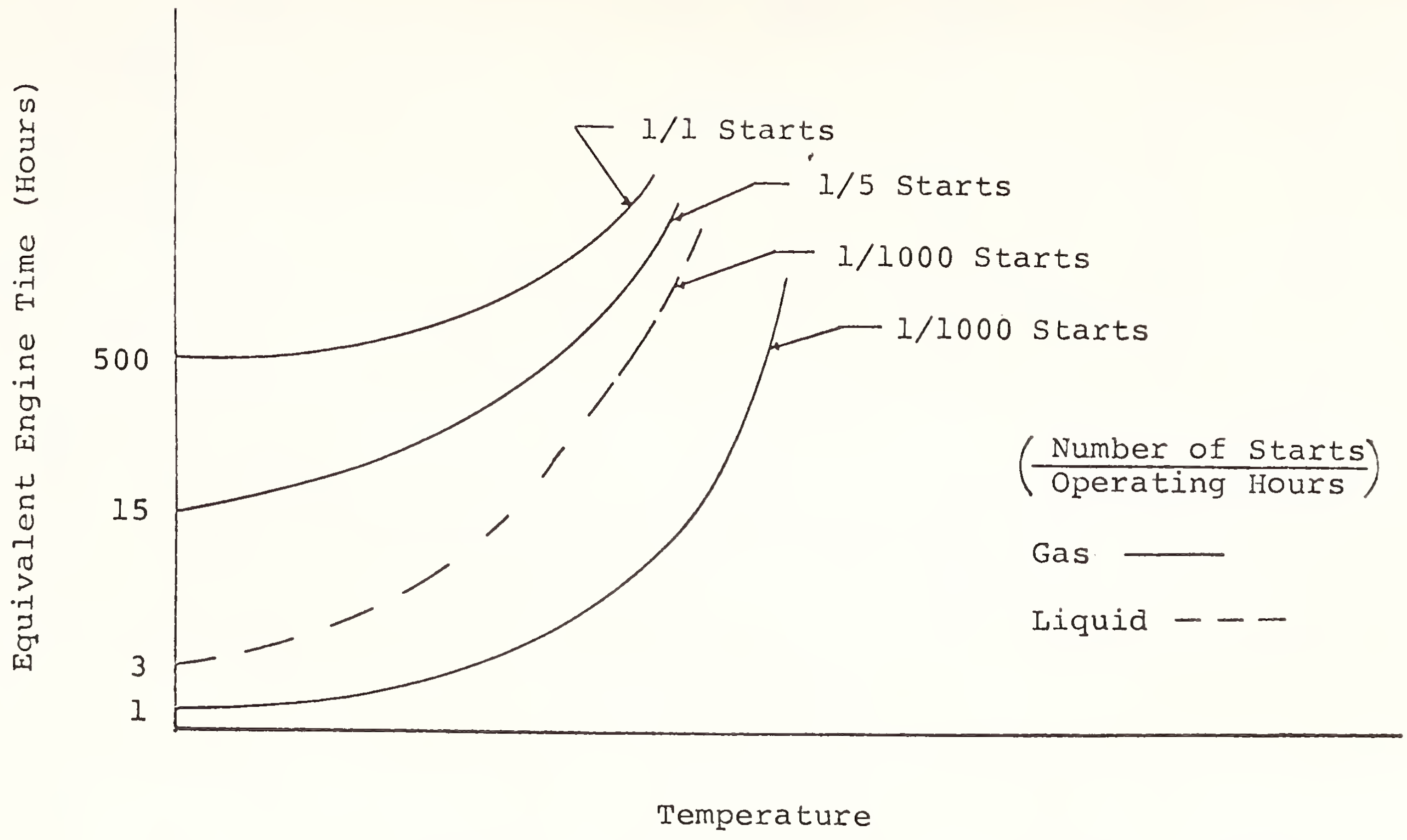



Figure 10 Equivalent Engine Time in the Combustor Section

TABLE III

TURBINE DIAGNOSIS

| | $\eta_t$ | $P_3/P_4$ | $T_3/T_4$ | Vibration | $\Delta T$ Bearing | Cooling Air Pressure | Wheel Space Temperature | Bearing Pressure |
|---|---|---|---|---|---|---|---|---|
| Fouling | ↓ | — | ↓ | ↑ | — | — | ↑ | — |
| Damaged Blade | ↓ | — | ↓ | ↑ | — | — | — | — |
| Bowed Nozzle | ↓ | ↓ | ↓ | ↑ | — | — | ↑ | — |
| Bearing Failure | — | — | — | ↑ | ↑ | — | — | ↓ |
| Cooling Air Failure | — | — | — | — | ↑ | ↓ | ↑ | — |

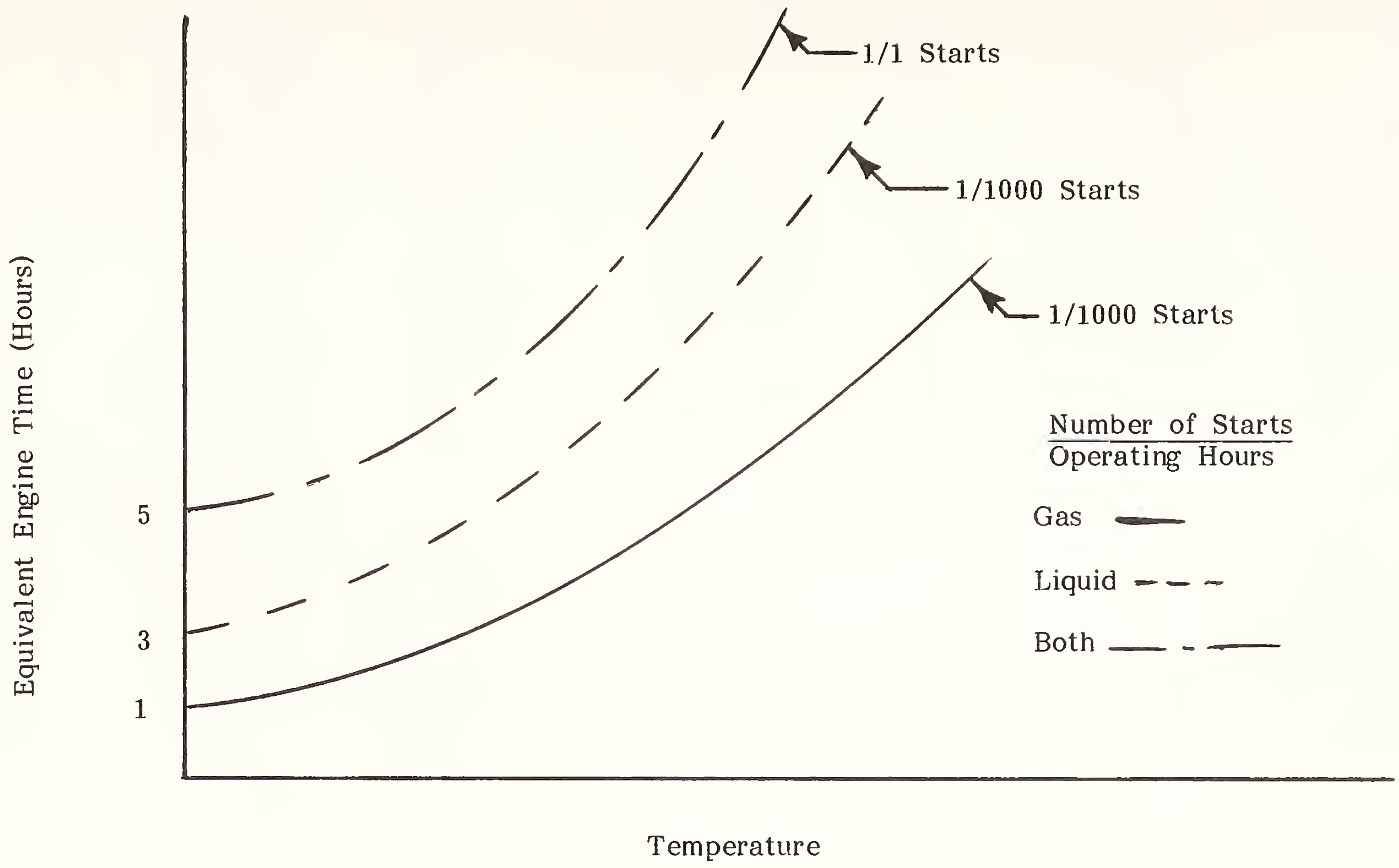



Figure 11: Equivalent Engine Time in the Turbine Section

6. Turbine Maintenance-This should be based on "Equivalent Engine Time" which is the function of temperature, fuel used, and number of starts and fuel is substantial. Figure 11 shows the correction that can be applied to running hours for intermittent duty units with high start/stop operation.

D. Turbine Efficiency

1. With the current high costs of fuel, very significant savings can be achieved by monitoring equipment operating efficiencies and correcting for operational inefficiencies. Some of these operational inefficiencies may be very simple to correct such as wash or clean of the compressor on a gas turbine unit. In other cases, it may be necessary to develop a load distribution program that achieves maximum overall efficiency of the plant equipment for a given load demand.

2. Figure 12 shows the significant dollar cost penalties that occur when operating a turbine at a very small percentage efficiency degradation.

3. Figure 13 shows a load distribution program for a 87.5 MW power station comprising of steam turbines and gas turbines. The selection of equipment and their loading for the most efficient operation can be programmed when the efficiency of individual units are monitored. The program selects the units which should be operated to provide the power load demand at the maximum overall efficiency of the combination of units.

## 13.0 MECHANICAL PROBLEM DIAGNOSTICS

The advent of new, more reliable and sensitive vibration instrumentation such as, the eddy current sensor and the accelerometer coupled with modern technology analysis equipment such as the real time vibration spectrum analyzer and low cost computers gives the mechanical engineer very powerful aids in achieving machinery diagnostics.

A chart for vibration diagnosis is presented in Figure 14. While this is a general criteria or rough guideline for diagnosis of mechanical problems, it can be developed into a very powerful diagnostic system when specific problems and their associated frequency domain vibration spectrums are logged and correlated in a computerized system. With the extensive memory capability of the computer system, case histories can be recalled and efficient diagnostics achieved.

## 14.0 DATA RETRIEVAL

In addition to being valuable as a diagnostic and analysis tool, a Data Retrieval program would also provide an extremely flexible method of data

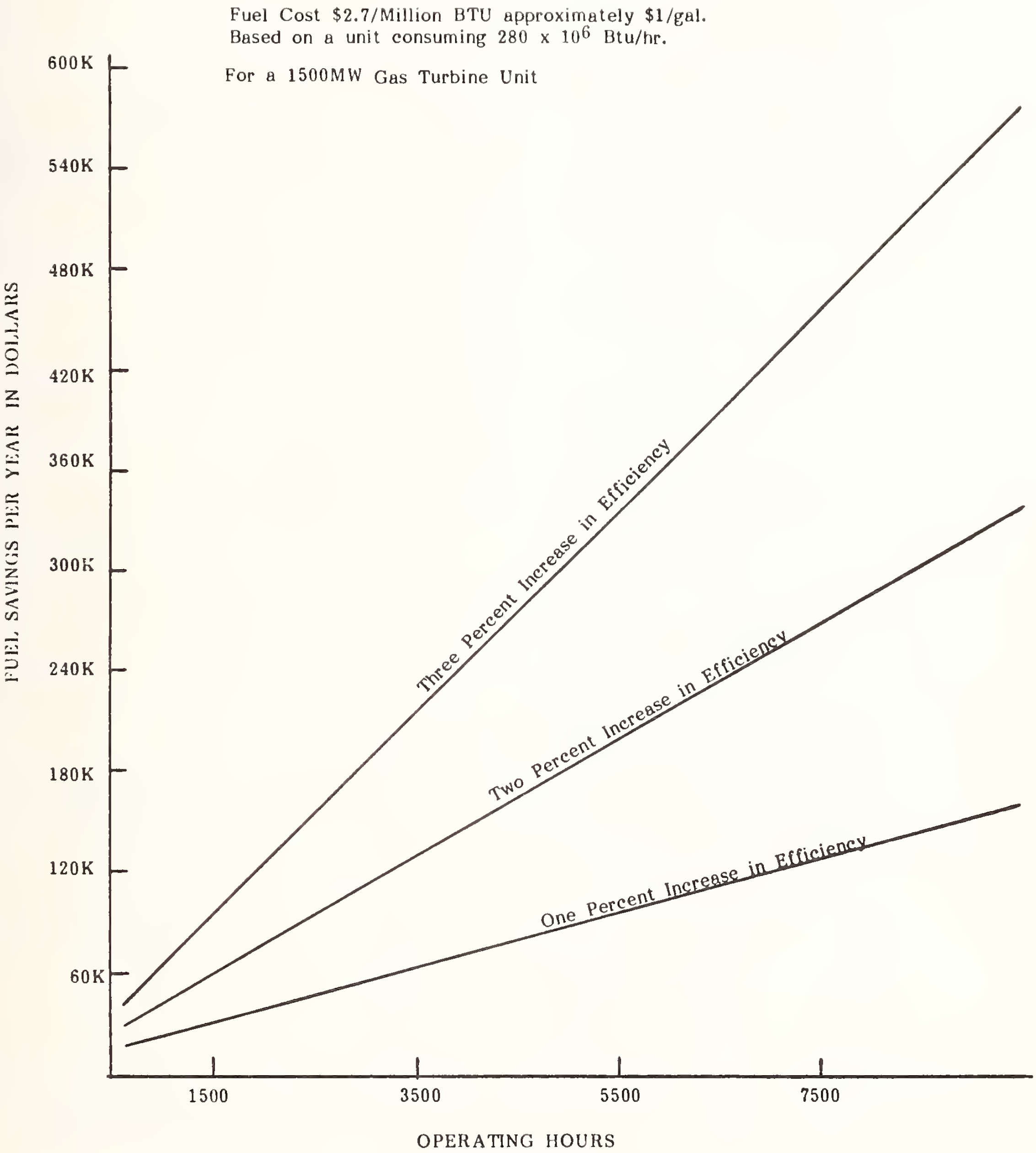



Figure 12: Savings Vs. Efficiency

DESCRIPTION OF UTILITY PLANTS UNITS

| UNIT# | DESIGN MW | TURBINE TYPE | EFFICIENCY AT DESIGN OUTPUT POINT |
|---|---|---|---|
| 1 | 2.5 | Steam | 22 |
| 2 | 2.5 | Steam | 22 |
| 3 | 5.0 | Steam | 24 |
| 4 | 5.0 | Steam | 24 |
| 5 | 5.0 | Steam | 24 |
| 6 | 7.5 | Steam | 25 |
| 7 | 15 | Steam | 30 |
| 8 | 15 | Steam | 23 |
| 9 | 15 | Gas | 21 |
| 10 | 15 | Gas | 21 |

COMBINATION OF UNITS TO YIELD EFFICIENT
POWER LOAD DISTRIBUTION FOR DIFFERENT DEMAND LOADS

UNITS NOT WORKING IN ASCENDING ORDER

1,4,9

EFFICIENT POWER LOAD DISTRIBUTION PROGRAM

| TOTAL DEMAND= | 30.00 | MW | TOTAL DEMAND= | 50.00 | MW |
|---|---|---|---|---|---|
| TOTAL OUTPUT SUPPLIED= | 30.00 | MW | TOTAL OUTPUT SUPPLIED | 50.00 | MW |

| | | | | | |
|---|---|---|---|---|---|
| Units Not Working= | | 1 4 9 0 | Units Not Working= | | 1 4 0 0 |
| Unit 1= | 0.00 | 0.00 | Unit 1= | 0.00 | 0.00 |
| Unit 2= | 0.00 | 0.00 | Unit 2 | 2.50 | 22.01 |
| Unit 3= | 2.50 | 21.00 | Unit 3= | 5.00 | 24.50 |
| Unit 4= | 0.00 | 0.00 | Unit 4= | 0.00 | 0.00 |
| Unit 5= | 5.00 | 24.50 | Unit 5= | 5.00 | 24.50 |
| Unit 6= | 7.50 | 25.19 | Unit 6= | 7.50 | 25.19 |
| Unit 7= | 15.00 | 29.91 | Unit 7= | 15.00 | 29.81 |
| Unit 8= | 0.00 | 0.00 | Unit 8 | 0.00 | 0.00 |
| Unit 9= | 0.00 | 0.00 | Unit 9= | 0.00 | 0.00 |
| Unit 10= | 0.00 | 0.00 | Unit 10= | 15.00 | 21.00 |
| MAXIMUM OVERALL EFFICIENCY=27.04 | | | MAXIMUM OVERALL EFFICIENCY=25.02 | | |

Power Demands = MW(Max.Demand=87.5)

Figure 13: Load Sharing Program

| Usual Predominant Frequency* | Cause of Vibration |
| --- | --- |
| 0-40% Running Frequency | Loose assembly of bearing liner, bearing casing, or casing and Support<br>Loose rotor shrink fits<br>Friction Induced Whirl<br>Thrust Bearing Damage |
| 40-50% Running Frequency | Bearing Support Excitation<br>Loose Assembly of Bearing Liner, Bearing Case, or Casing and Support<br>Oil Whirl<br>Resonant Whirl<br>Clearance Induced Vibration |
| Running Frequency | Initial Unbalance<br>Rotor Bow<br>Lost Rotor Parts<br>Casing Distortion<br>Foundation Distortion<br>Misalignment<br>Piping Forces<br>Journal & Bearing Eccentricity<br>Bearing Damage<br>Rotor Bearing System Critical<br>Coupling Critical<br>Structural Resonance<br>Thrust Bearing Damage |
| Odd Frequency | Loose Casing and Support<br>Pressure Pulsations<br>Vibration Transmission |
| Very High Frequency | Gear Inaccuracy<br>Valve Vibration<br>Dry Whirl<br>Blade Passage |

*Occurs in most cases predominantly at this frequency, harmonics may or may not agree.

Figure 14: Vibration Diagnosis

storage and recovery. By careful design of a health monitoring system, an engineer or technician could compare the present operation of this unit with the operation of the same machine, or of another machine, under similar conditions in the past. This could be done by selecting one, or several limiting parameters and defining the other parameters which are to be displayed when the limiting parameters are met. This would eliminate the necessity of sifting through large amounts of data. A few examples of how this system would be used are:

A. Retrieval by Time-In this mode, the computer would retrieve data taken during a specified time period, thus enabling the user to evaluate the period of interest

B. Retrieval by Ambient Temperature-The failure of the gas turbine may occur during an unusually hot or cold period, and the operator may wish to determine how his unit has functioned at this temperature in the past

C. Retrieval by Turbine Exhaust Temperature-The exhaust temperature can be an important parameter in failure investigations. An analysis of this parameter can verify the existence of a problem with either the combustor or turbine

D. Retrieval by Vibration Levels-Inspection of data provided by this mode can be useful in determining compressor fouling, compressor or turbine blade failure, nozzle bowing, uneven combustion and bearing problems.

E. Retrieval by Output Power-In this mode, the user should input the output power range of interest and would thus obtain only data applying to that particular power setting. In this manner, he would only have to consider the pertinent data to pinpoint the problem areas.

F. Retrieval by Two or More Limiting Parameters-By retrieving data with limits on several parameters, the data can be evaluated and will be even further reduced. Diagnostic criteria can then be developed.

## 15.0 CONCLUSIONS

A. The monitoring of mechanical characteristics of turbomachinery, such as vibrations, has been extensively applied this past decade. The advent of the accelerometer and the real time vibration spectrum analyzer has required a computer to match and utilize the extensive analysis and diagnostic capability of these instruments.

B. The high cost for machinery replacements and downtime makes machinery operational reliability very important, however with the currently prevailing and projected futher increases in fuel costs, aerothermal monitoring has become very important. Aerothermal monitoring can provide not merely increased operational efficiency for turbomachinery, but when combined with mechanical monitoring, provides an overall, more effective system than one that monitors only the mechanical functions or aerothermal functions.

C. While there had been concern on the reliability of computer systems, they are currently receiving wide acceptance and are fast replacing analog systems.

D. The systematized application of modern technology instrumentation, both mechanical and aerothermal, low cost computers and turbomachinery engineering experience will result in the development and application of cost effective systems.

# SESSION V

## OPPORTUNITIES FOR DETECTION, DIAGNOSIS, AND PROGNOSIS IN THE ENERGY FIELD

Chairmen: R. Hohenberg, Mechanical Technology, Inc.

J. L. Frarey, Shaker Research Corporation

# DATA ACQUISITION AND ANALYSIS IN THE DOE/NASA WIND ENERGY PROGRAM

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Abstract: The Lewis Research Center of NASA manages for the Department of Energy, the technology and engineering development of all large horizontal axis wind turbines. In support of this activity each wind turbine has various data systems used to acquire, process and analyze data. This report will identify four categories of data systems, each responding to a distinct information need. The categories are: Control, Technology, Engineering and Performance.

The focus of this report is on the Technology data system which consists of the following elements: (1) sensors which measure critical parameters such as wind speed and direction, output power, blade loads and strains, and tower vibrations; (2) remote multiplexing units (RMUs) mounted on each wind turbine which frequency modulate, multiplex and transmit sensor outputs; (3) the instrumentation available to record, process and display these signals; and (4) centralized computer analysis of data at the NASA-Lewis Research Center in Cleveland, Ohio.

RMU characteristics and multiplexing techniques are presented. Data processing is illustrated by following a typical signal through instruments such as the analog tape recorder, analog-to-digital converter, data compressor, digital tape recorder, video (CRT) display, and strip chart recorder. Sample output data from the 200 kW Mod-OA wind turbine at Clayton, New Mexico, are presented.

Key words: Wind energy; wind turbine generators; wind turbine performance; horizontal axis wind turbines; wind turbine data systems.

## I. Introduction

The U.S. Government has established a Wind Energy Program within the Department of Energy (DOE) to encourage the development and promote commercialization of wind energy systems. One phase of this program is being managed by the NASA Lewis Research Center (LeRC). An agreement with DOE stipulates that LeRC shall manage both the

Technology and Engineering Development for all large ( > 100 kW) horizontal axis wind turbines.[1] Four wind turbine projects, designated the Mod-0[2], Mod-0A[3], Mod-1[4], and Mod-2[5], are part of the current development program. In addition to these projects, efforts aimed at achieving lower machine costs have been initiated These include an advanced 1,000 kW-class wind turbine project (Mod-5) and an advanced 200 to 500 kW wind turbine project (Mod-6). The four existing models are shown in figure 1 and the major features of all these machines are summarized in Table 1.

With regard to all these machines, LeRC maintains the continuing capability to monitor, analyze, understand and report on their performance. Despite the diversity in Wind Turbine Generator characteristics amongst the machines listed in Table I, we can nonetheless identify four distinct information/user categories that are common to all wind turbines. Namely: operations/the wind turbine itself; technology/field operations personnel; engineering/system and component designers; performance/utility or program manager. The requirements for each category are sufficiently unique that we have developed a separate data system to meet each need. At one extreme, with the highest sampling rates, are the computer based control systems which govern the routine operation of each wind turbine generator. The data portion of the control system provides information regarding adequacy of the wind, status of all critical systems, machine alignment with the wind, etc., and often monitors over 100 sensors. In the case of critical controls it must be capable of responding within milliseconds. This system is considered to be an integral part of the wind turbine and varies significantly from one design to the next. At the other extreme, with the lowest sampling rate, would be a Performance Data System to provide data for evaluation of wind turbines in terms of availability, reliability and energy production. These data requirements are generally limited to meteorological and electrical parameters with time scales from an hour to the lifetime of the machine. The two remaining information systems, namely Technology and Engineering are discussed in greater detail in the remainder of this report. The next two sections deal with the signal conditioning acquisition and data display. The fourth section discusses the subsequent statistical analysis.

## II. Technology Information Systems

The Technology Data Acquisition/Display System has three functionally (and spatially) distinct components. As one follows the data signals from the sensors through the system, these are: signal conditioning, acquisition/display, and post processing for statistical analysis. Physically these functions occur: on the wind turbine, at or near the base of the wind turbine tower and at LeRC, respectively. As mentioned earlier in connection with Table I, the wind turbines display considerable variability as to source, location, blade composition and design. Despite, and to some extent because of this variability, it was decided that all data system implementations of must have the same (or functionally equivalent) hardware and software.

## Signal Conditioning

Signal conditioning is performed by a Remote Multiplexing Unit (RMU). As input, an RMU can accept up to 32 low-level or high-level data signals from a variety of transducers. Each RMU contains reference junctions for thermocouples as well as the necessary electronics for excitation and bridge completion of strain gauges. As a specific example, Table II contains a list of all the transducers monitored by the Technology Information System during the initial start-up of the Mod-1 at Boone, North Carolina. Each wind turbine has one RMU located in the hub, another in the nacelle and (with the exception of Mod-2) a third unit at the base of the tower in the controls room.

After a signal is received at the RMU it is conditioned (scale and/or offset; amplification or attenuation) to a common range and frequency multiplexed for output. Each RMU can generate two multiplex groups. Each multiplex group consists of up to 16 FM subcarriers ($\pm$ 125 Hz centered at 500 Hz intervals from 1000 Hz thru 8500 Hz) plus a precise reference tone at 9500 Hz. Other significant features of the RMU include a 4-pole active Butterworth low pass filter and an end-to-end system calibration capability (upon command from an external source).

## Acquisition/Display

LeRC has two nearly identical Technology Acquisition/Display Facilities. One is installed at the LeRC Plum Brook station in Sandusky, Ohio and is used by the LeRC engineering staff to conduct the Supporting Research and Technology program based on the Mod-0 machine. The other is installed in a large van.[6] This latter relocatable system is used to support field engineers thru assembly, check-out and initial operation of the first units of each new wind turbine design.

A schematic representation of the electronic data processing capability of this facility is shown in figure 2. All RMU generated FM multiplexes entering this facility are, with the addition of a time code, recorded in direct analog form. This recording can be performed independent of any other equipment or processing activity within the facility. Simultaneously, the data can also be routed thru a set of 6 banks of 16 discriminators which de-multiplex the signals and generate analog ( $\pm$5V) signals. Any or all of these 96 analog signals can be digitized and routed thru the mini-computer. From there it can be processed for real-time digital display on a CRT and/or for transmittal on digital tape to the LeRC main-frame computers for further analysis. In addition, any 24 of the 96 analog signals may be selected for display on strip charts and any single analog signal may be routed to a spectrum analyzer for frequency content evaluation. All the components shown with a gray stipling in figure 2, can be set up and run under computer control at the discretion of the facility operator via the operator's console. The facility in the van has sufficient capacity to simultaneously support up to three wind turbines at a single site.

## STATISTICAL ANALYSIS

Large volumes of data are of little value in their raw form. Even after processing, they may well be of negligible value if the end product is overwhelmingly voluminous, inadequately disseminated, or excessively delayed. To preclude these occurrences we routinely perform statistical analyses of both the technology and the engineering data. Condensed summaries are provided in a timely fashion in both graphic and tabular form, using microfiche as the distribution medium.

### Pre-Processing

Digital magnetic tapes are generated at one of the Technology Acquisition/ Display facilities. The digital data consist of 11 readings per parameter per (nominal) revolution and are stored as a tightly packed, randomly sequenced record. Each datum is accompanied by an identifying tag. Time markers (to nearest millisecond) are merged with the data.

As the first step, these data tapes are transferred to disk for short term (i.e. days) storage on the LeRC main frame system. During the transfer process, the internal representation is transformed (in software) from ASCII to EBCDIC. The next step in the processing compacts this initially large dataset ($\sim 5\text{x}10^6$ data values + $5\text{x}10^6$ tags + $1\text{x}10^6$ time markers) into a more manageable form, as follows: The rotor shaft position is used to mark the start of each rotation. These markers are then combined with the associated time markers and processed to give rotor speed (rpm) as a function of time. Then, the data from approximately 30 sensors of general interest are screened to yield maximum and minimum values for each parameter for each revolution of the rotor. This smaller data set ($\sim 5\times10^5$ data values) is stored on disk and is the data base for all further processing.

## Standard Analysis

In the final step, this latter data set is processed onto a microfiche containing the time history and statistical summary of each parameter of general interest. While the specific set of sensors and their associated scale factors will vary from machine to machine, the same presentation format is, nevertheless, applied to all data from all machines. This entire process is shown schematically in Figure 3.

The data, which have been stored as maximum and minimum values for each revolution, are combined to represent the midpoint and cyclic values for each revolution. The transformation equations are:

$$\text{midpoint} = (\text{maximum} + \text{minimum}) / 2$$

and

$$\text{cyclic} = (\text{maximum} - \text{minimum}) / 2) \times (1 + f(\text{rpm}))$$

where,

$$f(\text{rpm}) = 2\times10^{-5} (\text{rpm})^2.$$

The correction function, f(rpm), is introduced to compensate for the consistent underestimation of cyclic values resulting from the data sampling rate of 11 per (nominal) revolution.

The results of the analysis of each sensor are displayed as two frames, one graphical and one tabular on the microfiche card. The first of the two frames (see fig. 4) for each sensor contains three graphs. On each graph there are two plots, one of the midpoint values with circles as symbols and the other of the cyclic values with diamonds as symbols. The three graphs are:

1. Time history. This graph summarizes the information in the continuous trace associated with a strip chart recorder. One plot is of the average, over 30 second intervals, of the midpoint values. The other plot is of the corresponding cyclic values.

2. Partitioned distributions.[7] The abscissa for this graph is the wind speed midpoint value as measured at the nacelle. These wind speed values are used as the basis for sorting corresponding data values of the sensor of interest. The data values for the sensor of interest are grouped into subsets such

that for each subset all the sensor data values were obtained at approximately the same measured wind speed. Then the data values within each subset are separately ranked in ascending order. We find the 16th and 84th percentile for each such sequenced subset and display these percentiles as horizontal tabs at the end of a vertical bar. We also estimate the confidence interval (at the 0.95 level) for significant differences of the median[7] and display it as an interval (denoted by the occurrences of a circle or diamond) on the same vertical bar. This entire process is performed separately for the midpoint (circle) and cyclic (diamond) data values.

3. Cumulative distribution. This graph corresponds to a normal distribution , i.e. the abscissa is in units of normalized standard deviations and is segmented with tick marks and labelled by percentiles. Such a graph has the characteristic that if the plotted data have a normal (i.e. gaussian) distribution, the plot will appear linear. For this graph the entire set of all midpoint values is sequenced and plotted by percentile. This process is repeated, separately, for the cyclic values.

The second frame (see fig.5) for each sensor presents two tables listing all the plotted data points from both distributional plots. Some additional related but non-plotted, data are also tabulated. Because some of these present extreme values (i.e. maxima and minima), these tabulated values must be addressed with caution as they might represent spurious noise.

## III. Engineering Acquisitional/Display System

After the initial check-out, LeRC retains the responsibility to monitor, analyze, understand and report on all the wind turbines under our supervision . Since the Technology Acquisition/Display System described above is too elaborate and expensive for long term monitoring at each site we have identified a subset which we call the Engineering Acquisition/Display System. This system is installed in the control area of each wind turbine. This latter facility provides on-line analog display on strip chart (8 channels per wind turbine) and continuously records 48 signals (as 3 FM multiplexes) plus time on 4 tracks of analog magnetic tape. The analog magnetic tape record operates in either of two modes, depending on local conditions and requirements. At some sites they record until the tape is full (32 hours) and then automatically rewind (10 minutes) and restart, erasing old data as new data are recorded. At other sites the recorders operate for 96 hours (by making 3 passes thru the tape using a total of 12 tracks) and then automatically turn off. They remain off until the tape is replaced and the unit is manually restarted. These analog tapes can be played back onto strip charts at the site, and they can also be sent to one of the Technology Acquisition/ Display facilities for further processing of the data.


## ACKNOWLEDGEMENTS

The author wishes to especially acknowledge the understanding of FM data systems obtained from many conversations with Robert Wolf, who technically managed the procurement and operation of the first of the Signal Conditioning and the Technology Data Acquisition/Display Systems described in this report. I also wish to acknowledge the insights into wind turbine data interpretation provided by numerous colleagues with particular thanks for the patience shown by Timothy Richards, Richard Shaltens and Dr. David Spera.

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Table I - DOE/NASA Large Horizontal Axis Wind Turbines

| WTG | Rated Power, kW | Rotor Features | Rotor Diameter | Blade Material | Location |
|---|---|---|---|---|---|
| Mod-0 | 100 | variable | 38.5M | Aluminum, Steel, Wood | LeRC Plum Brook |
| Mod-0A 1 | 200 | downwind | | " | Clayton, NM |
| Mod-0A 2 | 200 | downwind | | " | Culebra, PR |
| Mod-0A 3 | 200 | downwind | | " | Block Island, RI |
| Mod-1 | 2000 | downwind | 61.5M | Steel Fiberglass | Boone, NC |
| Mod-0A 4 | 200 | downwind | | Wood | Kahuku, HI |
| Mod-2 | 2500 | upwind teetered | 92.3M | Steel | Goldendale, WA |
| SVU | 4000 | downwind teetered | 78.5M | Steel | Medicine Bow, WY |
| Mod-5A | 4000(P) | ? | 107.7M | ? | ? |
| Mod-5B | 4000(P) | upwind | 38.5M | Steel/Wood(P) | ? |
| Mod-6H | 500(P) | ? | ? | ? | ? |

(P) = Preliminary / ? = Not determined yet

TABLE II - INSTRUMENTATION LIST MOD-1 BOONE, NC AS OF 5 AUGUST 1980

| PARAMETER | SIGNAL *UNITS | SIGNAL LOWER | SIGNAL UPPER | ENGINEERING *UNITS | ENGINEERING LOWER | ENGINEERING UPPER |
|---|---|---|---|---|---|---|
| ROTOR BNG IN RACE | *MV | +0.391 | +5.281 | * DEG F | +50 | +250 |
| PITCH CHANGE BNG | *MV | -0.674 | +3.967 | * DEG F | 0 | +200 |
| SPAN STRAIN 469 | *MV | -20 | +20 | *FT-LBS | -4.728 E5 | 4.728 E5 |
| SPAN STRAIN 390 | *MV | -10 | +10 | * PSI | -2.8409E4 | 2.8409E4 |
| CB 907.5 | *MV | -10 | +10 | *FT-LBS | | |
| CHORD STRAIN 390 | *MV | - 5 | + 5 | * PSI | -5.6818E4 | 5.6818E4 |
| DIAG STRAIN 469 | *MV | - 5 | + 5 | * PSI | -5.6818E4 | 5.6818E4 |
| FB 907.5 | *MV | -50 | +50 | *FT-LBS | | |
| SPAN STRAIN 482 | *MV | -20 | +20 | * PSI | -1.1364E5 | 1.1364E5 |
| CB 469 | *MV | -20 | +20 | *FT-LBS | -2.1561E6 | 2.1561E6 |
| HUB OUTSIDE BRL | *MV | -20 | +20 | * PSI | -1.1364E5 | 1.1364E5 |
| HUB INSIDE BRL | *MV | -20 | +20 | * PSI | -1.1364E5 | 1.1364E5 |
| HUB TAIL X BEND | *MV | -20 | +20 | *FT-LBS | | |
| HUB TAIL Y BEND | *MV | -20 | +20 | *FT-LBS | | |
| PITCH ROD #1(T/C) | *MV | -20 | +20 | *IN/IN | -2.5 E4 | 2.5 E4 |
| NOT USED | * | -- | -- | * | -- | -- |
| FB 117 | *MV | -50.0 | +50.0 | *FT-LBS | -1.3967E7 | 1.3967E7 |
| CB 117 | *MV | -20 | +20 | *FT-LBS | -6.4008E6 | 6.4008E6 |
| FB 469 | *MV | -50 | +50 | * PSI | -1.1364E5 | 1.1364E5 |
| FB 469 | *MV | -50 | +50 | *FT-LBS | -2.7444E6 | 2.7444E6 |
| SPAN STRAIN 482 | *MV | - 5 | + 5 | * PSI | -2.8409E4 | 2.8409E4 |
| FB 117 | *MV | -- | -- | *FT-LBS | | |
| FB 907.5 | *MV | -50 | +50 | *FT-LBS | -5.4356E6 | 5.4356E6 |
| SPAN STRAIN 46 | *MV | -10 | +10 | * PSI | -5.6818E4 | 5.6818E4 |
| CB 907.5 | *MV | -10 | +10 | *FT-LBS | -2.2338E6 | 2.2338E6 |
| SPAN STRAIN 299 | *MV | -10 | +10 | * PSI | -1.1364E5 | 1.1364E5 |
| CB 117 | *MV | -20 | +20 | *FT-LBS | | |
| CB 469 | *MV | -20 | +20 | * PSI | -2.8409E4 | 2.8409E4 |
| HUB INSIDE BRL | *MV | -20 | +20 | * PSI | -1.1364E5 | 1.1364E5 |
| HUB OUTSIDE TAIL | *MV | -20 | +20 | * PSI | -1.1364E5 | 1.1364E5 |
| SHAFT TORSION | *MV | -20 | +20 | *IN/IN | -1.0 E3 | 1.0 E3 |
| PITCH ROD #2(T/C) | *MV | - 5 | + 5 | *IN/IN | -2.5 E4 | 2.5 E4 |
| INLET OIL TEMP | *MV | -.674 | +5.281 | * DEG F | 0 | + 250 |
| SERVO OUT (TP 1) | * V | 0 | +10 | * | -- | -- |
| RTR BNG OUT RACE | *MV | -3.91 | +5.281 | * DEG F | +50 | +250.5 |
| TRANS LUBE TEMP | *MV | -.674 | +5.281 | * DEG F | 0 | + 250 |

| | | | | | | |
|---|---|---|---|---|---|---|
| INLET OIL TEMP | *MV | .391 | +5.281* | DEG F | 50 | + 250 |
| GEARBOX FLANGE | *MV | -20 | +20 | *IN/IN | -9.9E-4 | +9.9E-4 |
| AIR TEMP (OUT) | *MV | -1.667 | +3.967* | DEG F | -50 | +2001 |
| PITCH ROD DEC P | * V | 0 | + 5 | * PSI | 0 | +4000 |
| GEN BNG SHAFT END | * V | 0 | +10 | * DEG F | -30 | +300 |
| GEN WINDING TEMP | * V | 0 | +10 | * DEG F | -30 | +300 |
| AFT HORIZONTAL | *MV | -20 | +20 | *IN/IN | -9.9E-4 | +9.9E-4 |
| GEN SHAFT SPEED | * V | 0 | +10 | * RPM | 0 | 39.5 |
| PITCH ROD INC P | * V | 0 | + 5 | * PSI | 0 | +4000 |
| RTR BED VIB (Z) | *MA | -3.325 | +3.325* | G | -2.5 | +2.5 |
| RTR BED VIB (Y) | *MA | -3.325 | +3.325* | G | -2.5 | +2.5 |
| RTR BED VIB (X) | *MA | -3.325 | +3.325* | G | -2.5 | +2.5 |
| BLADE PITCH ANGLE | * V | +6.54 | +9.11 * | DEG | +24.0 | -4.2 |
| RTR BED VIB (Z) | *MA | -3.325 | +3.325* | G | -2.5 | +2.5 |
| FWD BED VIB (Y) | *MA | -3.325 | +3.325* | G | -2.5 | +2.5 |
| SERVO OUT (TP 4) | * V | 0 | +10 | * | -- | -- |
| YAW TORQUE CCW | * V | 0 | + 5 | * PSI | 0 | +4000 |
| YAW ERROR | * V | 0 | +10.0 | * DEG | -270 | +270 |
| BLADE PITCH ANGLE | * V | 0 | 10 | * DEG | +96.0 | -14.0 |
| WIND SPEED(NAC) | * V | +0.25 | +10.0 | * MPH | 2.5 | 100 |
| COVER FLANGE | *MV | -20 | +20 | *IN/IN | -9.9E-4 | +9.9E-4 |
| GEARBOX WEB | *MV | -20 | +20 | *IN/IN | -9.9E-4 | +9.9E-4 |
| RTR SHAFT POS | * V | 0 | +10 | * DEG F | 0 | +360 |
| AFT 45° | *MV | -20 | +20 | *IN/IN | -9.9E-4 | +9.9E-4 |
| YAW TORQUE CW | * V | 0 | + 5 | * PSI | 0 | +4000 |
| AFT VERTICAL | *MV | -20 | +20 | *IN/IN | -9.9E-4 | +9.9E-4 |
| YAW POSITION | * V | 0 | +10 | * DEG | 0 | + 360 |
| SERVO CURRENT | * V | 0 | +1.5 | * | -- | -- |
| SOUND (5/27/80) | * V | -2.0 | +2.0 | * PSF/db | -- | -- |
| SOUND (5/27/80) | * V | -2.0 | +2.0 | * PSF/db | -- | -- |
| SOUND (5/27/80) | * V | -2.0 | +2.0 | * PSF/db | -- | -- |
| UTILITY VOLT EXP | * V | +7.2 | +8.8 | * KVOLT | 3.78 | 4.62 |
| GEN CURRENT A | * V | 0 | +10.0 | * AMPS | 0 | 600 |
| GEN CURRENT B | * V | 0 | +10.0 | * AMPS | 0 | 600 |
| GEN CURRENT C | * V | 0 | +10.0 | * AMPS | 0 | 600 |
| TOW VIB SITE#1(X) | *MA | -3.325 | +3.325* | G | -2.5 | +2.5 |
| TOW VIB SITE#2(X) | *MA | -3.325 | +3.325* | G | -2.5 | +2.5 |
| EXCITER FLD CURR | * V | 0 | 2.5 | * AMPS | 0 | 10 |
| NOT USED | * | -- | -- | * | -- | -- |
| NOT USED | * | -- | -- | * | -- | -- |
| UTILITY VOLT | * V | 0 | +10.0 | * KVOLT | 0 | 5.25 |
| NOT USED | * | -- | -- | * | -- | -- |
| NOT USED | * | -- | -- | * | -- | -- |
| NOT USED | * | -- | -- | * | -- | -- |
| TOW VIB SITE#1(Y) | *MA | -3.325 | +3.325* | G | -2.5 | +2.5 |
| GEN PWR KVARS | * V | -10 | +10 | * KVARS | -4200 | 4200 |
| GEN PWR WATTS | * V | -2.38 | +10 | * KW | -1050 | 4200 |
| -- | | | | | | |

| | | | | | | | | |
|---|---|---|---|---|---|---|---|---|
| TOW VIB SITE #1(Z) | * | MA | -3.325 | +3.325 | * | G | -2.5 | +2.5 |
| WIND SPEED 250' | * | V | 0 | 5 | * | MPH | 0 | 100 |
| TOW VIB SITE #2(Z) | * | MA | -3.325 | +3.325 | * | G | -2.5 | +2.5 |
| ROTOR SHAFT SPEED | * | V | -10 | +10 | * | RPM | 20.0 | 40.0 |
| NOT USED | * | | -- | -- | * | | -- | -- |
| NOT USED | * | | -- | -- | * | | -- | -- |
| AIR TEMP 150' | * | V | 0 | 5 | * | DEG F | -30 | 110 |
| WIND DIRECT 250' | * | V | 0 | 5 | * | DEG | 0 | 360 |
| WIND DIRECT 150' | * | V | 0 | 5 | * | DEG | 0 | 360 |
| WIND SPEED 60' | * | V | 0 | 5 | * | MPH | 0 | 100 |
| TOW VIB SITE #2(Y) | * | MA | -3.325 | +3.325 | * | G | -2.5 | +2.5 |
| WIND DIRECT 60' | * | V | 0 | 5 | * | DEG | 0 | 360 |
| WIND SPEED 150' | * | V | 0 | 5 | * | MPH | 0 | 100 |
| GEN PWR WATTS | * | V | -10 | +10 | * | KW | -3600 | +3600 |
| STRAIN 669 | * | | -- | -- | * | | -- | -- |
| STRAIN 664 | * | | -- | -- | * | | -- | -- |
| WHITE GEN NOISE | * | V | -5.0 | +5.0 | * | V | -5.0 | +5.0 |
| CHORD STRAIN 469 | * | MV | -10 | +10 | * | FT-LBS | -5.1768E6 | 5.1768E6 |
| DIAG STRAIN 390 | * | MV | - 5 | + 5 | * | PSI | -2.8409E4 | 2.8409E4 |
| SPAN STRAIN 299 | * | MV | -10 | +10 | * | PSI | -5.6818E4 | 5.6818E4 |
| FWD BED VIB (X) | * | MA | -3.325 | +3.325 | * | G | -2.5 | +2.5 |

*************************************************************************

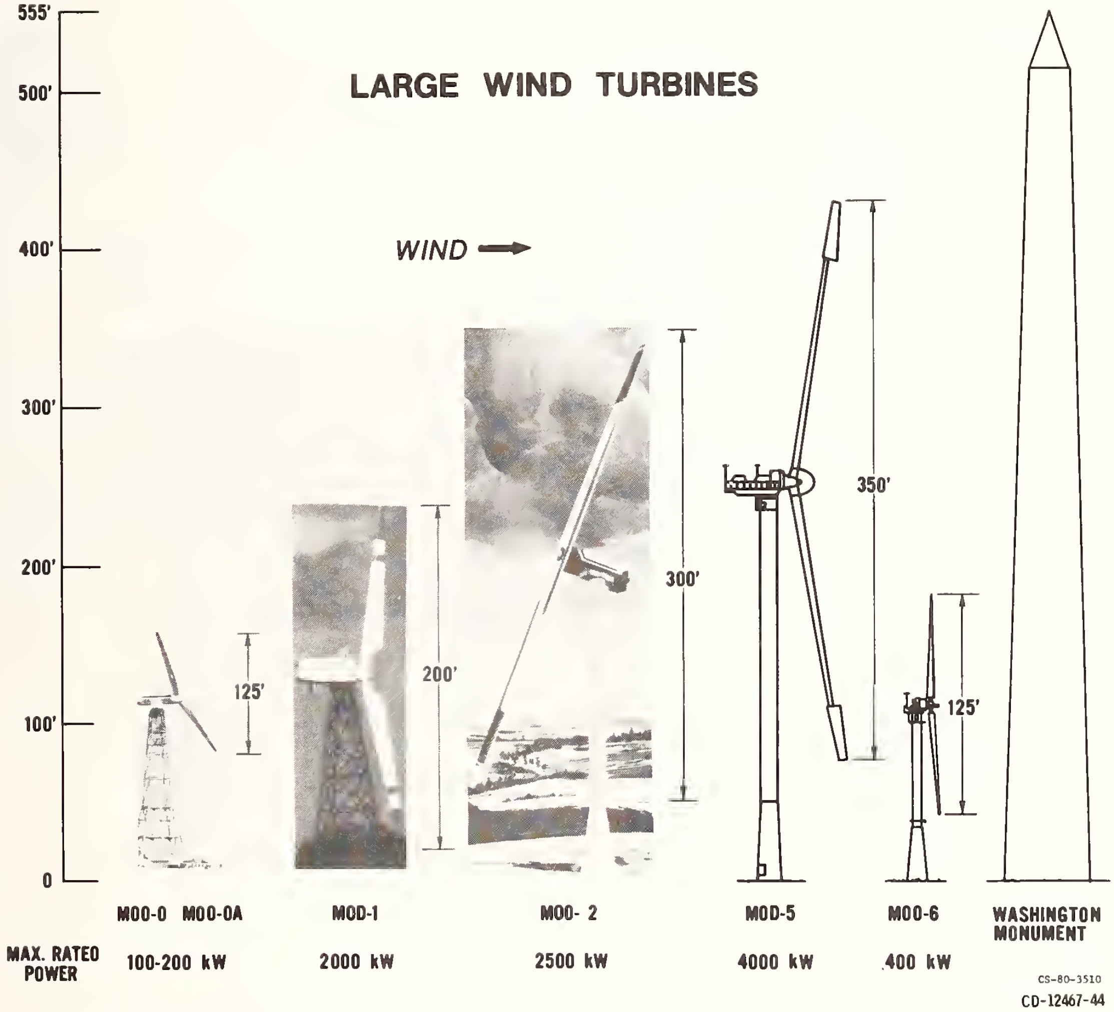

555'
500'
400'
300'
200'
100'
0
LARGE WIND TURBINES
WIND
125'
200'
300'
350'
125'
MOD-0 MOD-0A
MOD-1
MOD- 2
MOD-5
MOD-6
WASHINGTON MONUMENT
MAX. RATED POWER
100-200 kW
2000 kW
2500 kW
4000 kW
400 kW
CS-80-3510
CD-12467-44

Figure 1.

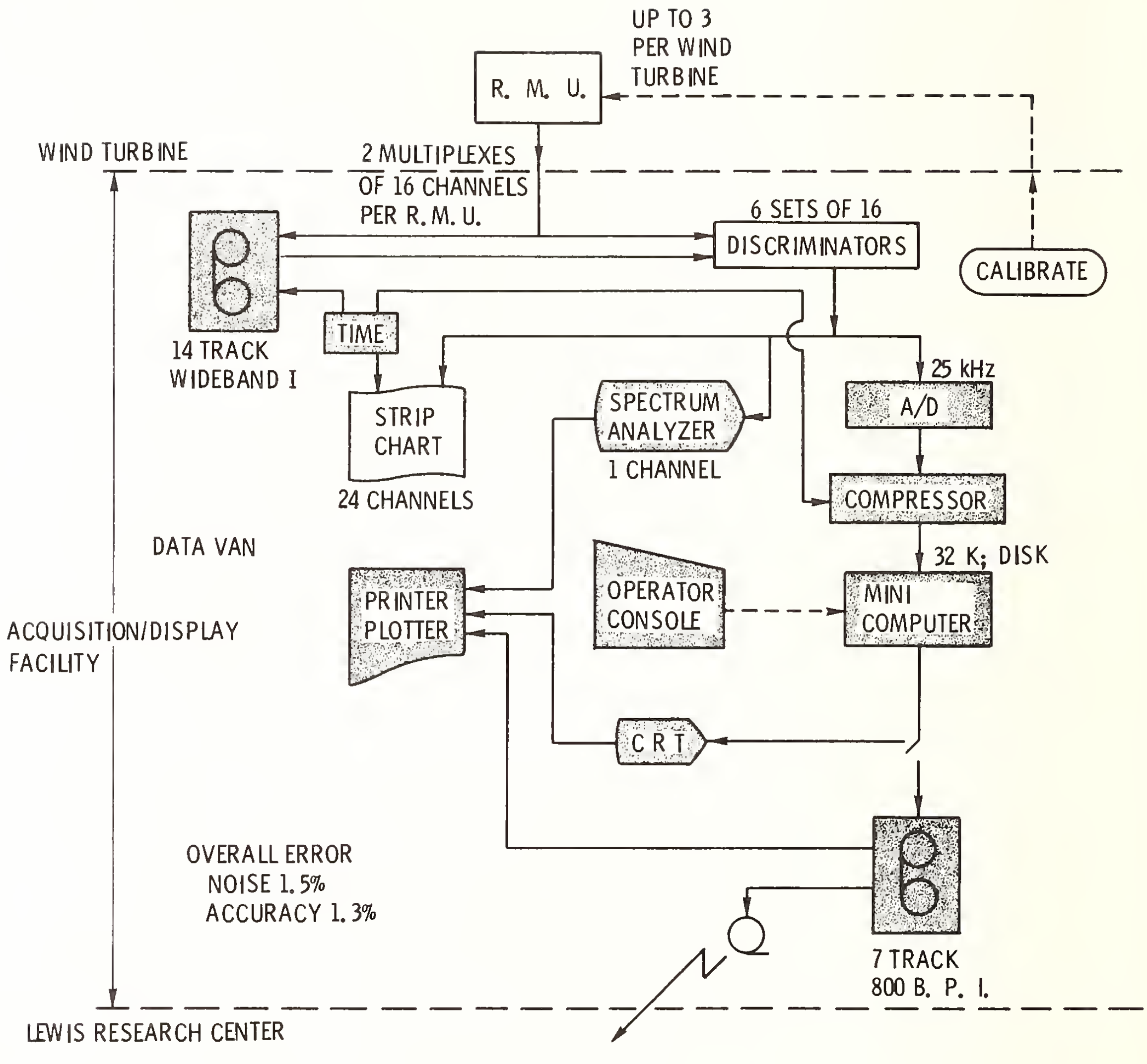



Figure 2. - Schematic representation of the electronic data processing in the Technology Acquisition/Display Facility. Every device shown with gray stipling can be controlled by the operator vis the computer console.

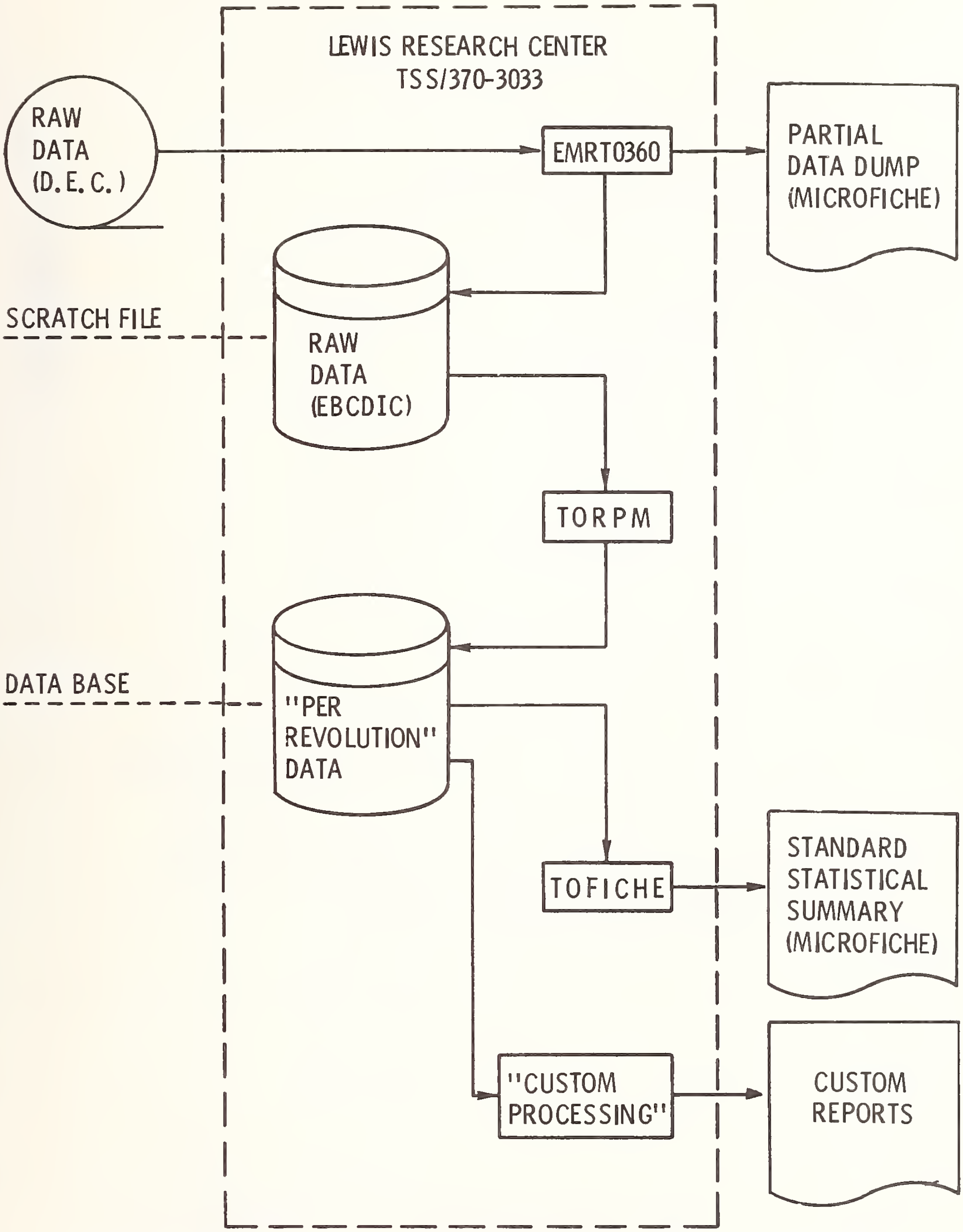
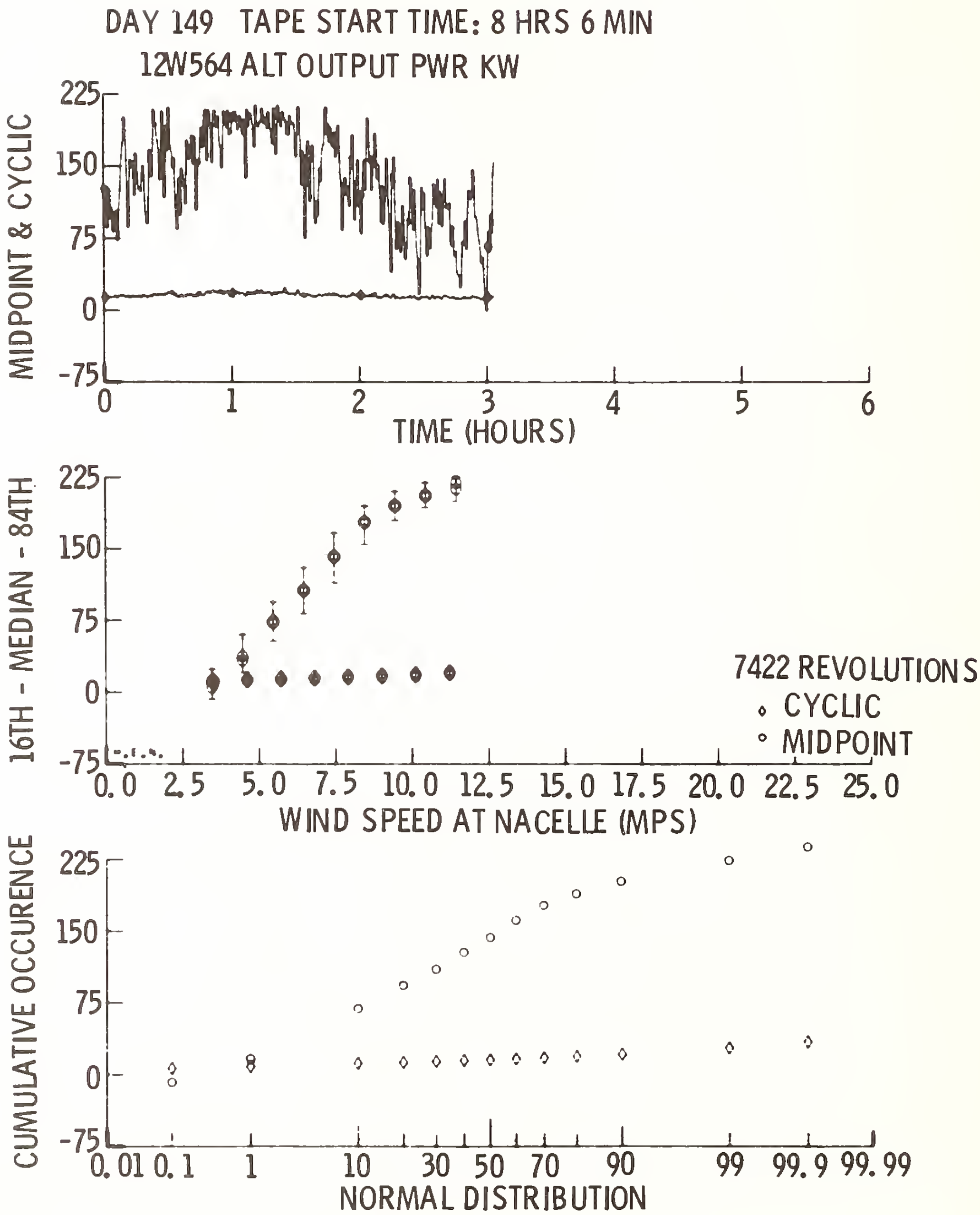



Figure 3. - Schematic representation of the data processing at the LeRC main computer facility.

CLAYTON, NEW MEXICO MACHINE: MOD OA1 1980



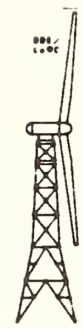



Figure 4. - Sample frame of the statistical data summary routinely available on microfiche.

CLAYTON,NEW MEXICO MACHINE : MOD OA1 1980
DAY 149 TAPE START TIME: 8HRS 6MIN
12W564 ALT OUTPUT PWR KW

---------- SEGMENTED DISTRIBUTIONS ----------

| | M I D P O I N T | | | | | | C Y C L I C | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| WIND | MINIMUM | 16TH PCTL | MEDIAN | 84TH PCTL | MAXIMUM | WIND | MINIMUM | 16TH PCTL | MEDIAN | 84TH PCTL | MAXIMUM |
| 3.5 | -1.4E 01 | -6.0E 00 | 9.4E 00 | 2.5E 01 | 4.6E 01 | 3.5 | 5.9E 00 | 8.4E 00 | 1.1E 01 | 1.5E 01 | 1.9E 01 |
| 4.5 | 1.9E 00 | 2.1E 01 | 3.6E 01 | 6.1E 01 | 9.2E 01 | 4.6 | 5.7E 00 | 1.0E 01 | 1.3E 01 | 1.6E 01 | 2.3E 01 |
| 5.5 | 1.0E 01 | 5.5E 01 | 7.4E 01 | 9.5E 01 | 1.4E 02 | 5.7 | 5.7E 00 | 1.1E 01 | 1.4E 01 | 1.7E 01 | 2.9E 01 |
| 6.5 | 2.4E 01 | 8.3E 01 | 1.1E 02 | 1.3E 02 | 1.8E 02 | 6.8 | 6.2E 00 | 1.1E 01 | 1.4E 01 | 1.8E 01 | 3.5E 01 |
| 7.5 | 5.0E 01 | 1.2E 02 | 1.4E 02 | 1.7E 02 | 2.0E 02 | 7.9 | 6.0E 00 | 1.3E 01 | 1.6E 01 | 2.0E 01 | 4.0E 01 |
| 8.5 | 6.0E 01 | 1.6E 02 | 1.8E 02 | 2.0E 02 | 2.3E 02 | 9.0 | 6.8E 00 | 1.4E 01 | 1.7E 01 | 2.1E 01 | 3.6E 01 |
| 9.5 | 1.4E 02 | 1.8E 02 | 2.0E 02 | 2.1E 02 | 2.5E 02 | 10.1 | 8.9E 00 | 1.5E 01 | 1.8E 01 | 2.2E 01 | 3.5E 01 |
| 10.5 | 1.5E 02 | 1.9E 02 | 2.1E 02 | 2.2E 02 | 2.5E 02 | 11.2 | 1.2E 01 | 1.6E 01 | 2.0E 01 | 2.3E 01 | 3.2E 01 |
| 11.5 | 1.9E 02 | 2.0E 02 | 2.2E 02 | 2.3E 02 | 2.4E 02 | 12.3 | 0.0 | 0.0 | 0.0 | 0.0 | 0.0 |

--------- CUMULATIVE DISTRIBUTION ----------

| FREQUENCY | MIDPOINT | CYCLIC | MAXIMUM | MINIMUM |
|---|---|---|---|---|
| 0.0010 | -7.1 | 6.1 | 1.3 | -16.9 |
| 0.0100 | 17.7 | 8.2 | 29.2 | 5.2 |
| 0.1000 | 69.7 | 11.0 | 83.1 | 56.4 |
| 0.2000 | 93.9 | 12.4 | 108.1 | 79.5 |
| 0.3000 | 110.7 | 13.5 | 125.3 | 96.6 |
| 0.4000 | 128.6 | 14.4 | 144.2 | 113.1 |
| 0.5000 | 144.6 | 15.4 | 160.5 | 128.9 |
| 0.6000 | 162.2 | 16.3 | 178.6 | 145.7 |
| 0.7000 | 177.8 | 17.5 | 194.5 | 161.1 |
| 0.8000 | 190.1 | 19.1 | 207.6 | 172.3 |
| 0.9000 | 203.0 | 21.2 | 221.5 | 185.2 |
| 0.9900 | 225.5 | 27.6 | 246.0 | 207.0 |
| 0.9990 | 239.3 | 33.6 | 262.7 | 222.1 |

Figure 5. - Tabular data produced in conjunction with the plots shown in figure 4.

# PERIODIC VIBRATION MONITORING AT THE SYNTHANE COAL-GASIFICATION PILOT PLANT

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Abstract: Periodic vibration monitoring of mechanical equipment has proven to be an effective early-detection method for identifying and scheduling preventive maintenance. As a means of demonstrating this approach to the emerging U.S. synthetic fuel production industry, vibration monitoring was performed at the SYNTHANE pilot plant, one of several coal-gasification facilities sponsored by the U.S. Department of Energy (DOE). The procedures used in obtaining, analyzing and interpreting vibration data are described in this paper and, to provide a cost basis, the level of effort required to obtain, process, and report the data is given. The utility of the data as an aid to maintenance management in the pilot plant is evaluated and extrapolated to predict the general usefulness of periodic vibration monitoring as a data source for the management and operation of future, large, coal-conversion plants.

Key Words: Vibration monitoring; mechanical components; coal conversion; SYNTHANE pilot plant.

Introduction: In order to assess the feasibility of various coal-conversion process technologies, several pilot plants have been developed under U.S. DOE sponsorship. This paper discusses the data obtained by monitoring the vibrations of the mechanical equipment in one of these facilities, the SYNTHANE pilot plant. The transition from bench-scale apparatus to a virtually complete pilot plant has been made for the SYNTHANE process; and, even though the pilot plant has been placed on mothball status since January 1979, many of the initial complex uncertainties that existed during the design of the plant in 1972 have been resolved and recorded.

In early 1978, a DOE-sponsored program was undertaken to provide mechanical data for future large plants by evaluating the equipment used in the SYNTHANE pilot plant. As part of this program, vibration measurements were made on all major rotating and reciprocating equipment. The general program coupled the analysis of vibration data with an analysis of maintenance history and a systems analysis of the entire plant [1]*.

*Numbers in brackets designate references at the end of this paper.

This recorded information now constitutes a valuable part of the data base that is available from the U.S. Coal Conversion Program. The coal program is important to the U.S. because it deals with a domestic resource that is critical to meeting national energy needs. The coal program is also important to the Mechanical Failures Prevention Group for two reasons: 1) implementation of the proposed Federal Synthetic Fuels Program will require large numbers of mechanical components in many commercial plants that are still to be built, and 2) the failure rate of mechanical components operated in the predecessor pilot plant program was alarmingly high.

Pilot Plant Background: SYNTHANE (Synthetic Methane) is a high-Btu, high-pressure, coal-gasification pilot plant, owned by the U.S. Government and located in South Park Township, Allegheny County, Pennsylvania. SYNTHANE's unique feature is its potential to gasify caking coals found in the eastern section of the U.S. The heart of the process is the pretreater-gasifier, operating at about 1000 psia. The gasifier is an oxygen-blown, single-stage, fluidized-bed reactor which is preceded by a high-pressure, in-line, oxygen-fed, fluidized-bed pretreater.

The SYNTHANE process was developed by bench-scale experiments conducted at the Pittsburgh Energy Research Center in 1965. The preliminary design for the 72-tons-of-coal-per-day pilot plant was made by M.W. Kellogg Company in 1970, and the bid package was subsequently prepared by the C-E Lummus Company. Rust Engineering Company constructed the plant, which was turned over to C-E Lummus in August 1974 for start-up and operation.

Operation of the SYNTHANE gasification pilot plant was officially inaugurated by Test Directive No. 1, issued from ERDA (now DOE) on June 9, 1976. Figure 1 shows a schematic of the overall SYNTHANE process. Figure 2 shows the test periods, the equipment operating time per month and the period during which vibration monitoring was performed.

An explosion occurred in the high-pressure boiler on September 21, 1978, and, in January 1979, DOE directed the operating contractor to place the plant in mothball status. As a consequence, equipment in functional blocks downstream of the gas scrubbing and cooling section were never operated in the integrated mode.

Monitoring Objectives: SYNTHANE was the first U.S. coal-conversion plant to investigate the plantwide use of vibration monitoring as a potential data source for managing plant operations. The principal objective was to demonstrate if vibration monitoring produced useful data in a cost-effective manner, while contending with the uncertain schedule of the pilot-plant test program. Specific objectives of the monitoring program were to:

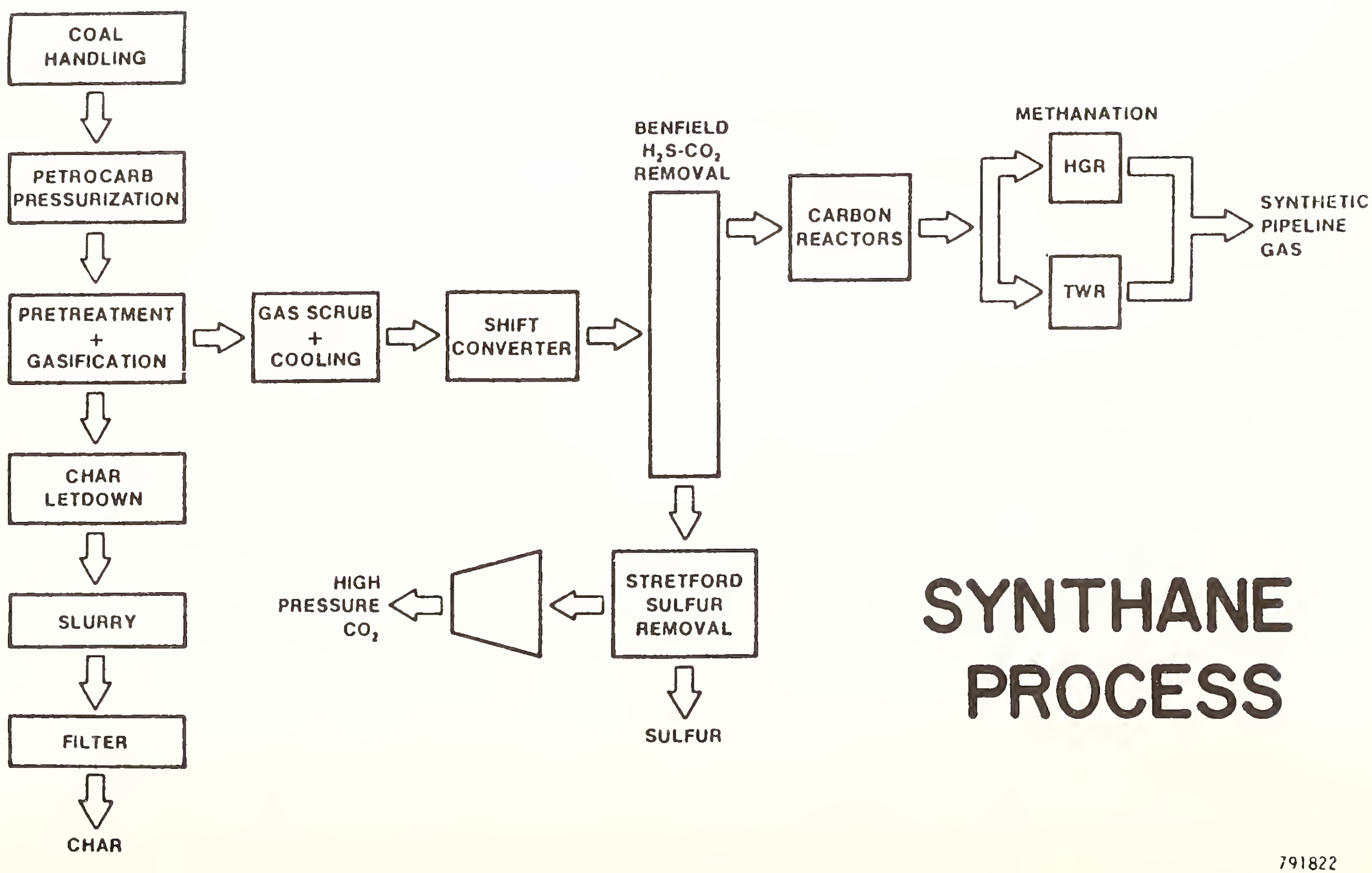


Figure 1  Synthane Process Flow Sheet [1]

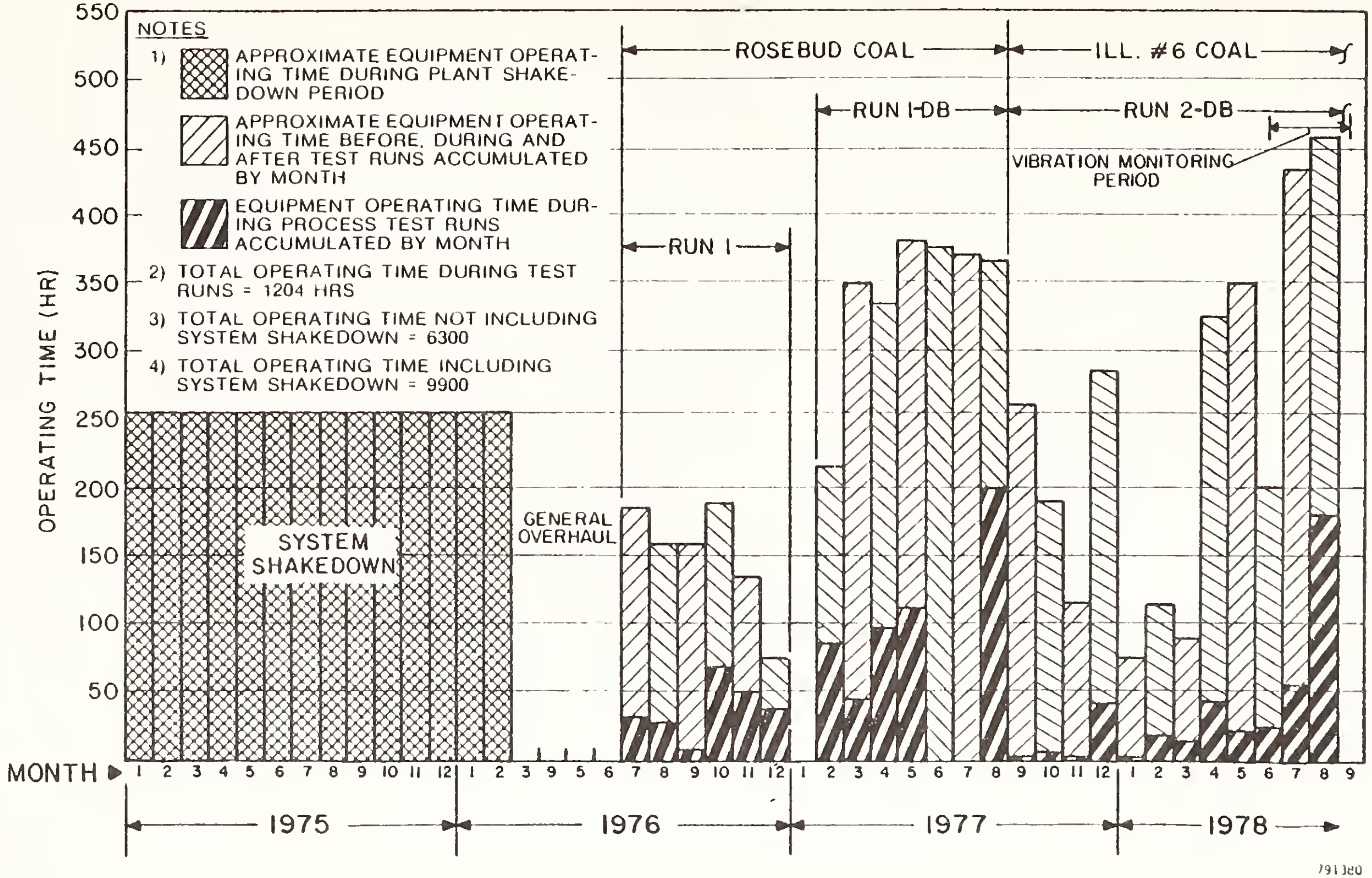


Figure 2 Estimated Operating Time for Mechanical Components [1]

- Establish initial tolerance tables relating vibration amplitudes to machine conditions.
- Demonstrate that an analysis of sequence of periodic vibration readings, referenced to the tolerance tables, provides an accurate estimate of the present and future condition of the equipment.
- Demonstrate that acquiring useful vibration data does not necessitate access to the machine internals, but is possible from sensors mounted on the housing.

The Monitoring Procedure: Selection of the equipment to be monitored was based on criticality to the process, size, and frequency of use. Table 1 lists the 16 equipment trains selected for vibration monitoring and provides an overall assessment of their repair history.

Metal discs for mounting sensors were cemented to the machine housing and adjacent piping; vertical, horizontal and axial locations were chosen near key elements, such as bearings, couplings and seals, and depending upon accessibility and need. Vibration signals from both low-frequency and high-frequency portable transducers were recorded on magnetic tape for later analysis. The low-frequency transducers were the seismic type where the voltage output was linearly proportional to frequency in the range 15 to 1700 Hz. The high-frequency transducers were the piezoelectric type where the voltage output was linearly proportional to frequency in the range 1 to 10 kHz. In addition to these transducers, a large number of unusual problems with the two carbon dioxide compressors prompted plans to install the special diagnostic equipment, as listed on Table 2. However, because the plant was unexpectedly shut down, this instrumentation was not installed.

A two-man team conducted surveys at monthly intervals while the equipment was operating under normal load. The base-line signatures were recorded in two days with subsequent data recorded at the rate of 30 to 40 data points per hour. Each survey collected a total of 138 data points, and was executed by the team in less than one day.

Data Analysis: Two types of analysis were performed on the collected data. The first was a severity analysis that provided the basic frequency/amplitude data to assess the present condition of the equipment. The second was a trend analysis that focused on changes in vibration levels, as compared with the initial base-line measurements and subsequent readings.

Based on past experience with similar equipment, maintenance data and the base-line signatures, tolerance tables for each machine were formulated. Tables 3A, 3B and 3C represent a sample tolerance table for a specific machine. Table 3A lists the conditions that must be

TABLE 1

LIST OF EQUIPMENT SELECTED FOR VIBRATION MONITORING

| Section | Description | Item Number | Assessment of Repair History | | |
|---|---|---|---|---|---|
| | | | Major Repair | Frequent Repair | Normal Maintenance |
| Coal Handling | Drying Air Blower | GB-101 | X | | |
| | Drying Fan | GB-102 | X | | |
| Char System | Wilson - Snyder Pump | GA-401S | | | X |
| | Filter Feed Pump | GA-406 | X | X | |
| | Filter Feed Pump Spare | GA-406S | X | X | |
| | Vacuum Pump | GB-401X | X | | |
| | Vacuum Pump Spare | GB-401XS | X | | |
| Gas Scrubbing and Cooling | Venturi Scrubber Recycle Pump | GA-207 | X | X | |
| | Venturi Scrubber Recycle Pump Spare | GA-207S | X | X | |
| | Scrubber Tower Recycle Pump | GA-201 | X | X | |
| Utilities | Air Compressor | GB-304 | | X | |
| | Air Compressor Spare | GB-304S | | X | |
| | BFW High-Pressure Pump | GA-303 | X | | X |
| | BFW High-Pressure Pump Spare | GA-303S | | | X |
| | $CO_2$ Compressor | GB-301 | X | X | |
| | $CO_2$ Compressor Spare | GB-301S | X | X | |

TABLE 2

PROPOSED MONITORING INSTRUMENTATION FOR $CO_2$ COMPRESSORS

| Item | Instrumentation | Purpose |
|---|---|---|
| 1 | Pressure transducers: head and crank end of each cylinder; motion transducer on crankshaft | • Produce recorded pV diagram<br>• Measure valve performance<br>• Measure pressure ratio<br>• Infer ring clearances |
| 2 | Temperature transducer: head and crank end each cylinder | • Measure gas temperature<br>• Infer presence of abnormal amount of liquid in head |
| 3 | Accelerometers: two on housing of each cylinder in axial and transverse directions | • Infer clearances in wrist pin, crosshead and crankshaft bearings |
| 4 | Microphone | • Detect gasket leakage |
| 5 | Elapsed time meter | • Record running time |

NOTES:
1. Portable pressure and temperature transducers can be accommodated by permanently installed indicator valves in each cylinder. All cylinders do not have to be recorded simultaneously.
2. Portable accelerometers and microphones can be accommodated by permanently installed, tapped pads. All cylinders do not have to be recorded simultaneously.

TABLE 3A

SAMPLE TOLERANCE TABLE

Conditions To Be Satisfied Before Data Table Is Printed

| | | Code |
|---|---|---|
| 1. | All 1X & 2X Synchronous Data | |
| | • Frequency equals shaft speed | 2 |
| 2. | Maximum Amplitudes | |
| | • Amplitude is ≥ lowest severity desired | 8 |
| | • Acceleration ≥ lowest desired | 16 |
| 3. | Trends | |
| | • New amplitude/BL*Amplitude > 0.5 and BL amplitude ≥ 0.0785 in./sec | 32 |
| | • New amplitude/BL amplitude ≥ 3.0 and BL amplitude ≥ 0.0196 in./sec | 64 |
| | • New amplitude/previous amplitude ≥ 0.7 and previous amplitude ≥ 0.0785 in./sec | 128 |
| | • New amplitude/previous amplitude ≥ 2.0 and previous amplitude ≥ 0.0196 in./sec | 256 |

LOWEST SEVERITY DESIRED = 0.0392 in./sec
LOWEST ACCELERATION DESIRED = 1 g

Printed Data

- Sensor position
- Frequency
- Shaft speed
- Mils, peak-to-peak
- Velocity, in./sec
- Acceleration, g's
- Transducer type
- Trigger code
- Date of last observation

*BL = Base line; first readings of all sensors used as reference level for measuring changes and rate of change.

TABLE 3B

SAMPLE TOLERANCE TABLE

Relationship Between Velocity Transducer Reading and Running Condition of Machine

| Index | Maximum Velocity Less than, in./sec | Running Condition |
|---|---|---|
| 1 | 0.0049 | Extremely smooth |
| 2 | 0.0098 | Very smooth |
| 3 | 0.0196 | Smooth |
| 4 | 0.0392 | Very good |
| 5 | 0.0785 | Good |
| 6 | 0.1570 | Fair |
| 7 | 0.3140 | Slightly rough |
| 8 | 0.6280 | Rough |
| 9 | 1.2560 | Very rough |
| 10 | >1.2560 | Extremely rough |

TABLE 3C

SAMPLE TOLERANCE TABLE

Relationship Between Accelerometer Transducer Reading and Running Condition of Machine

| Index | Maximum Acceleration (g's) | Running Condition |
|---|---|---|
| 4 | <0.6 | Very good |
| 5 | >0.6 | Good |
| 6 | >3.0 | Fair |
| 7 | >6.5 | Slightly rough |
| 8 | >10.0 | Rough |
| 9 | >25.0 | Very rough |
| 10 | >40.0 | Extremely rough |

satisfied before a data table is generated. Table 3B presents the relationship between the reduced velocity transducer signal and the machine condition; Table 3C shows that same relationship for acceleration. A computer code was used to search the data tape for amplitudes exceeding the preset limits of the tolerance table. This code, together with equipment drawings and repair records, was used to analyze present problems and to predict future conditions. Table 4 illustrates a generally accepted correlation [3] between frequency ranges and vibration phenomena; the correlation is often used as a guide in this type of cause-and effect analysis. Although vibration data acquisition and processing are key elements, the real value of the results is heavily dependent upon the skill and experience of the data analyst.

Reporting: After the field data were analyzed and recommendations were formulated, the following reporting steps were taken:

1. If the analysis indicated there was a reasonable probability that any machine was close to a catastrophic failure, the plant manager was notified by telephone and a confirmation telegram was transmitted.
2. A monthly report was issued containing:
   - A one-page executive summary of problem machinery, status, and recommended corrective action.
   - Tables of severity and trend data for all machines satisfying similar conditions to those given in Table 3A.
   - Computer-generated multicolor pen plots of selected machines showing time-spaced trends in amplitude versus frequency.

Results: During the initial plant survey in July 1978, sensor mounting pads were installed and preliminary vibration measurements were made to assess the initial status of the equipment. Taped vibration data for analysis were taken in August. In September, one week before plant operations were terminated by DOE, a second set of data was recorded.

Vibration monitoring results are presented in three forms: charts for quick status assessment, specific recommendations for maintenance actions, and reduced measured data giving quantitative status and trends. Based on vibration data taken on September 13, 1978, Figure 3 shows the relative need for maintenance by dividing the equipment into three categories:

- Components that are running satisfactorily
- Components that show a trend toward failure and are expected to need maintenance soon

TABLE 4

CORRELATION OF FREQUENCY RANGES WITH VIBRATION PHENOMENA

| Frequency | Cause | Amplitude |
|---|---|---|
| Less than 1 x running speed | Flexible critical speed | Often severe |
| | Oil whirl | Often severe |
| | Structural resonances from impact loading | Often severe |
| Running speed | Unbalance | Proportional to unbalance |
| | Rotor-stator rubs under some conditions | Proportional to unbalance |
| 2 x running speed | Mechanical looseness | Often severe |
| | Bent shaft | Often severe |
| | Misaligned coupling | Often severe |
| | Nonlinearities | Small |
| | Drive belts | May be 3 to 4 x running speed |
| 60, 120, 250, 360 Hz | Electrical | Generally small |
| High multiples of running speed | Gear tooth passing frequency | Generally small |
| | Blade pass frequency | Often severe |
| 2,000 to 10,000 Hz random noise | Gas or liquid turbulence | Aerodynamic noise due to high Reynolds No. |
| 2,000 to 50,000 Hz discrete frequencies, often amplitude modulated | Rubs, damaged rolling-element bearings | Often severe |

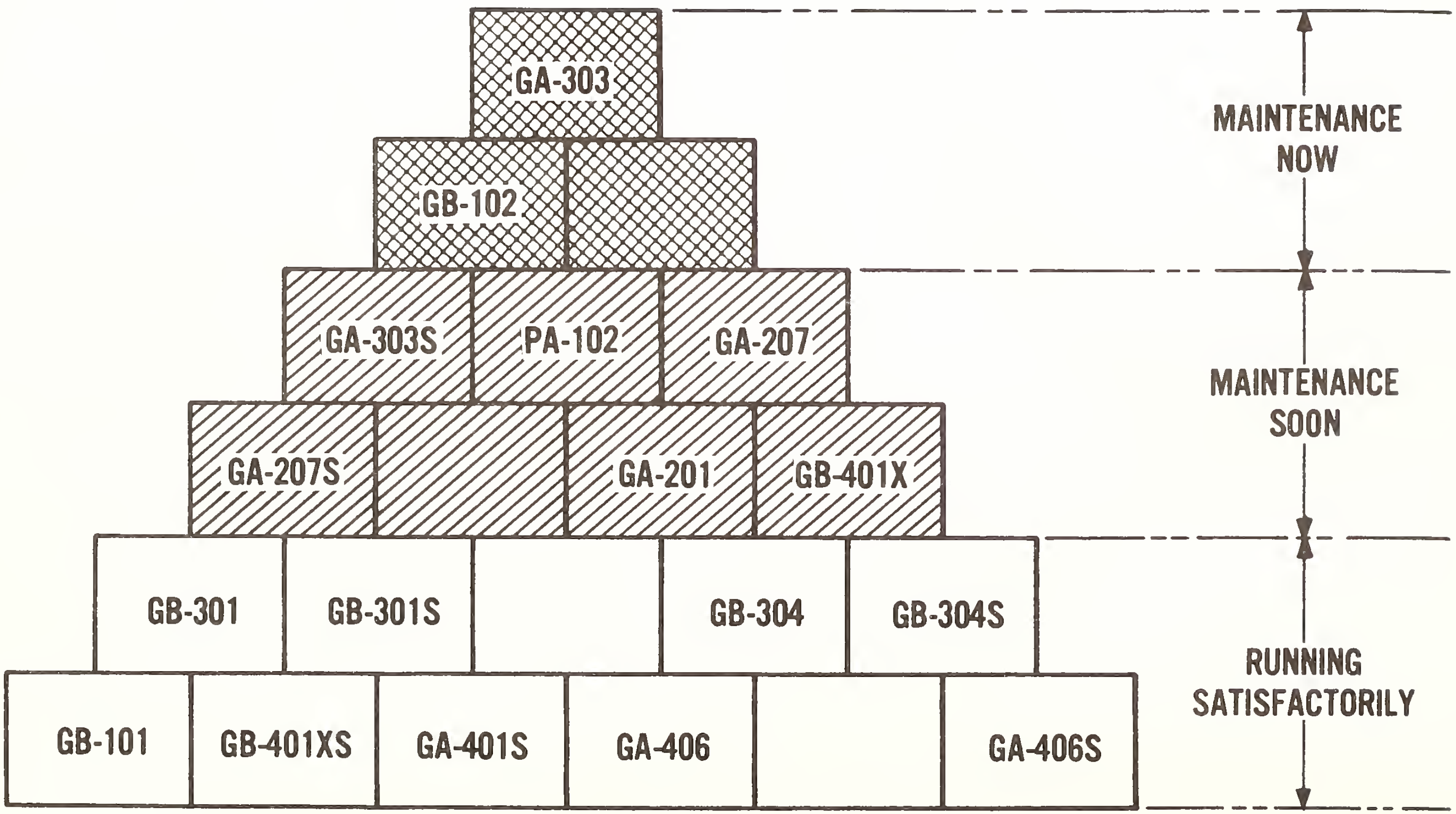


Figure 3 Results of Vibration Data Survey

- Components that show a critical problem and need maintenance now.

In Figure 3, the components are designated by the item numbers given in Table 1. Machines that are running satisfactorily are at the base of the pyramid, while the machine with the highest maintenance priority is in the block at the top of the pyramid. The pyramid format is used to permit quick review by plant management. Thus, components that reappear at the top of these monthly charts would be highlighted as troublesome items so as to alert management of the need to seek corrective action. Of the 17 components surveyed, Figure 3 shows that 9 components, or 53%, were running satisfactorily, 6 components, or 35%, were in need of scheduled maintenance and 2 components, or 12%, were in need of immediate attention.

In addition to qualitative data for plant management, analysis of the vibration data (given in detail in Reference [2]) produced recommendations for specific maintenance actions. These recommendations are repeated here because they are believed to be representative of the type of guidance that can be expected from periodic vibration monitoring.

1. <u>Components Believed to Warrant Immediate Corrective Action</u>:

   - Boiler feedwater pump GA-303 near catastrophic failure; recommend immediate shutdown. Mass unbalance of high-speed rotors, bearing damage and seal deterioration suspected.*

   - Coal drying system fan GB-102 has excessive unbalance; recommend balancing as soon as possible. Replacement of drive belts and realignment of sheaves recommended.**

2. <u>Components Believed to Warrant Scheduled Maintenance</u>

   - Boiler feedwater pump GA-303S motor rotor out of balance; recommend cleaning and balancing.

   - Hammer mill PA-102 motor either misaligned with mill sheave or mounting bolts loose; recommend inspection of belt/sheave alignment and check mounting bolt torque.

3. <u>Components Believed to Have Developing Problems; However, Corrective Action Is not Currently Recommended</u>

   - Venturi Scrubber Water Recycle Pump GA-207 believed to have unbalanced impeller. Some degradation of pump bearing and misalignment of motor to pump suspected.

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*Inspection showed broken gear tooth and bearing damage.

**Inspection showed fan blades bent from water accumulation in fan housing.

- Venturi Scrubber Water Recycle Pump GA-207S and Scrubber Tower Water Recycle Pump GA-201 believed to have unbalanced impellers.

The above results imply that the role of vibration monitoring in a pilot plant clearly can be a significant benefit in maintaining mechanical equipment. As a decision-making tool for management, vibration monitoring is a simple and cost-effective source of quantitative data on the condition of machines.

Vibration Monitoring for the Future Synthetic Fuels Program: The development of new processes in the petrochemical industry often proceeds through trials that address a succession of key technical and economic issues. The scope of these trials might include bench-scale tests, followed by small pilot-plant tests, and, in turn, large pilot-plant tests. At this point, if test data meet preset goals, a pioneer plant is built that completes the economic forecast and demonstrates any remaining question of scale-up. The equivalent DOE program steps generally include bench-scale testing, pilot-plant testing and demonstration-plant testing. At the end of this sequence, DOE, through various economic incentives, expects to entice industry to commercialize the technology. During all phases of plant testing, effort should be focused on the mechanical equipment so as to produce the data needed to formulate a detailed mechanical failure prevention plan for the commercial plant.

The SYNTHANE process was evaluated through bench-scale and pilot-plant tests. Sufficient data were collected to permit DOE to conclude that the SYNTHANE process was not as promising as other process options; consequently, DOE currently does not plan to scale-up this process. However, the magnitude of managing future equipment problems can be sensed from other processes that are in advanced development stages. One of these, the Solvent Refined Coal Process (SRC II) demo project, is directed at converting coal to liquid; it is one of five different coal-conversion technology demonstration plants being funded simultaneously by DOE. The 6000-ton-per-day (TPD) SCR II demo plant is expected to cost $1.4 billion when full operation is reached in late 1984. The current SCR II project will demonstrate one full-size train of a planned commercial plant that will consist of five identical trains. The five-train commercial SCR II plant is designed to process 30,000 TPD of coal and produce 100,000 barrels of oil equivalent per day (BOE/D). Since the current National Energy Plan calls for producing about 2 million BOE/D from coal by 1990, the equivalent of 20 SRC II commercial plants will be required if this goal is to be met. Placing this goal in a machinery perspective, consider the following preliminary estimate of the compressor needed for a single SCR II train:

COMPRESSORS FOR ONE TRAIN OF SCR II PLANT

| Gas | Flow (scfm) | Suction Pressure (psia) | Discharge Pressure (psia) | Horsepower |
|---|---|---|---|---|
| Air | 220,000 | 15 | 110 | 44,000 |
| Hydrogen | 80,000 | 1,100 | 2,500 | 8,500 |
| Hydrogen Recycle | 180,000 | 2,300 | 2,500 | 1,200 |
| Oxygen | 40,000 | 15 | 1,500 | 22,000 |
| Nitrogen | 20,000 | 5 | 300 | 7,000 |

In addition to the fact that all of these compressors will be large and costly, the oxygen compressor presents a unique problem because there is a lack of commercial experience at these high-pressure and high-flow rates.

In some respects, the magnitude of the equipment requirements for an SRC II commercial plant is similar to the SASOL 2 coal-conversion plant in South Africa, which cost $3 billion to design and construct. This plant, which occupies an area 2.5 km by 3.5 km, consumes coal at the rate of 40,000 TPD and produces 58,000 bbl/D of transportation fuels. In constructing this plant, 186,000 tons of steel were erected, 120 km of underground pipe were laid, 66,000 valves were fitted and 6100 items of equipment were installed [4]. As an example of the size of some of the machinery, the 15,000-TPD, 500-psia oxygen plant uses six 200-ton axial-centrifugal air compressors; each is driven by a 73-ton synchronous motor rated at 35,500 kW. Oxygen from the air separation plant is compressed to the delivery pressure by six 27-ton centrifugal compressors; each is driven by a 21-ton induction motor rated at 13,700 kW. As one measure of similarity between SASOL 2 and the proposed U.S. SRC II commercial plant, both plants require 192-MW electrical input to drive the air compressors for their oxygen plants.

SASOL I, the first coal-conversion plant built in South Africa, has accumulated more than 20 years of commercial operating experience. During the five-year period following construction of this plant, the process was refined and many pieces of equipment were modified. Operating and maintenance procedures were established and SASOL I became the pioneer plant for SASOL 2 and SASOL 3. The U.S. is about to embark on a similar venture, probably using several different conversion technologies, some of which may be at high pressure compared to SASOL. If the U.S. Synthetic Fuels Program plant proceeds on schedule, plant availability assurance is certain to be a priority topic. As a part of this topic, vibration monitoring can help to provide a controlling handle on the complex problem of dealing with large numbers of components. This emerging field offers the Mechanical Failures Prevention Group a unique opportunity to contribute toward organizing and executing the plans that will be necessary to make this undertaking successful.

Summary: The analysis of vibration data can often clarify the present conditions of a dynamic component. Each piece of equipment has unique signature frequencies; changes in amplitude and slight shifts in frequencies can frequently be correlated with the onset of specific, identifiable problems. Early trouble signs can often be detected by periodic measurements made by transducers mounted on the machine housing; because the majority of mechanical problems develop slowly, adequate time is usually available to plan corrections. Electronic hardware and computer codes for analyzing the cause of trouble signs are readily available, and the scope of their capabilities is increasing steadily. The cost of vibration monitoring and data analysis is low, and this procedure is an effective means of acquiring quantitative data to manage the maintenance of dynamic equipment.

The vibration monitoring performed on the equipment used in the SYNTHANE plant produced useful and important information that otherwise would not have been available to plant personnel. Analysis of the data produced a concise list of machine problems, weighted according to relative urgency and accompanied by specific recommendations for corrective action.

The U.S. Synthetic Fuels Program will involve plants that are large in comparison to the SYNTHANE pilot plant, and these plants will require effective maintenance if a reasonable overall plant availability goal is to be met. An attractive attribute of the vibration monitoring program demonstrated in the SYNTHANE pilot plant is that this approach can be scaled-up to meet the demands of the Synthetic Fuels Program without significant loss in simplicity or utility. While most components can be monitored satisfactorily on a periodic basis, some equipment may require dedicated diagnostic systems until sufficient data can be gathered to determine how to make these special components more serviceable. In any case, the monitoring results demonstrated at SYNTHANE indicate that the burden of operating the new generation of coal-conversion plants in a cost-effective manner will be greatly enhanced by incorporating a vibration monitoring program in the initial operating plan for these facilities.

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2. Rickert, B., "Periodic Vibration Monitoring of Mechanical Equipment at the SYNTHANE Pilot Plant," MTI Interim Report 2615-6-1, Contract EF-77-C-01-2615, September 1978.

3. Machinery Vibration Seminar Course Notes by Mechanical Technology Incorporated, June 1980.

4. "Synthetic Fuel the SASOL Way," Engineering and Mining Journal, December 1979, pp. 54-60.

# DESCRIPTION OF "LA RANCE" TIDAL POWER PLANT

. USE OF THE PLANT

. ANTICIPATION, DETECTION AND IDENTIFICATION OF ELECTROMECHANICAL EQUIPMENT FAULTS

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Electricité de France
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75008 PARIS - FRANCE

## 1. - DESCRIPTION OF PLANT

### 1.1 - Introduction to the plant

Before showing you a film in English on the Rance plant, and especially its design features, I must point out that it has become very difficult to say anything really original on the subject. People have been writing about tidal machinery since the year 1737 (a French Army engineer called BELIDOR), about tidal electricity production since 1918 (MAYNARD) and about the La Rance site in particular since 1944 (GIBRAT). Also, La Rance is still the only tidal power plant in the world, at least for the time being.

For additional general design details, therefore. I will refer to an extensive bibliography, part of which is in English. For particularly interesting technical features, I will quote from articles by or conversations with Mr. ROUX, Mr.JULLIARD and other successive managers of the plant, Mr. LEFRANCOIS and Mr.HILLAIRET, chief power production executives for Brittany, and Mr. ANDRE, Mr. COTILLON and Mr. BANAL and technicians of equipment manufacturers and E.D.F. who have been most active in circulating full information on the technology tried out at La Rance in English.

### 1.2 - Other bulb units

The bulb units at La Rance are derived from similar units which have been operating satisfactorily elsewhere, both before and since La Rance. Table 1 shows some statistics for these units.

I also thought you might be interested in seeing a descriptive film on the technology and installation of the turbines at one of these plants (Pierre Benite on the river Rhone) at a later session.

## 2 - USE OF "LA RANCE" PLANT

### 2.1 - Principles

A) The two following major principles should be borne in mind :

a) Electricity cannot be stored permanently. Producers have to adapt the use of their production facilities to consumer's requirements.

b) Tides are not governed by the solar cycle on which most human activities depend, but follow the lunar cycle. In France, two tides at approximately 12 1/2 hour intervals are observed. The peak equinoctial spring tide amplitude at La Rance is 13.5 m.

B) The basic principle of harnessing tidal energy consists in damming-off one or more areas from the sea to form storage ponds and thus obtain a difference in water levels across a barrage containing the power plant.

The diagrams in Table 2 illustrate the one-way and two-way cycles - with and without pumping - to which tidal power plant operates. La Rance, which has a single storage basin, is designed for use of all cycles.

C) The next consideration is to match the available facilities as closely as possible to power system requirements (cost per marginal kW) and tidal potential.

For this, a computer program is prepared from tide table data, the energy cost histogram and both operating conditions and limitations (environment, plant and equipment, power system). It operates by dynamic programming on a CII 9040 computer, optimizing the power unit and gate-operating conditions in 10-minute steps, and of course, over several tides (usually a week's tides).

The results are fed into an automatic plant-operating unit (PDP 8) which applies all marginal corrections according to the actual tide, or if limitations or requirements vary. The operator in charge takes over manually if such variations become excessive. In this connection, you may be interested to know that the simulations show pumping to be a more attractive proposition than two-way operation.

A further point to note is that, assuming constant cost, total availability of plant and equipment and no pumping limitations, tidal output may vary in proportions of 1 to 18, for a neap tide and an equinoctial spring tide respectively, which is within conventional hydrology dispersion limits.

D) Just as a reminder, the plant is equipped with twenty-four 10 MW bulb units each capable of discharging 250 $M^3/s$, with 5.35 m-diameter runners at 93.75 r.p.m. Peak natural flow is 16,000 $M^3/s$.

## 2.2 - Operating results

A) Water-power plant operation can be organized to maximalize one of the following :

a) Production

b) Scope for emergency compensating action for power system failures.

c) Income (a function of varying production cost), by determining the maximum value for :

$$\int_{t_1}^{t_2} n \cdot \eta \cdot Q \cdot H \cdot p \cdot dt$$

where : $n$ = number of units in service

$\eta$ = efficiency (see efficiency curves) : f (Q,H)

$Q$ = rate of flow through each unit : f (H)

$H$ = level difference

$p$ = cost of energy at time t : f (t)

It was decided to run La Rance for maximum profit, which explains why -as we are frequently asked- the plant has been generally operating at below its full production capacity (the estimated deficit is roughly 60 GWh).

I will now illustrate the operating results with a few tables.

B) Gross and net production (Table 3)

The difference is accounted for by the pumping operation I mentioned earlier on, which was suspended to avoid premature ageing of the generators (see para. 3.2) and gradually resumed in step with preventive maintenance.

C) Availability of units (Table 4)

Note the excellent, gradually increasing availability rate before preventive renovation of the equipment. A rate of 95 % corresponds to average annual availability of twenty-three of the twenty-four units. Since 1976, this average rate has been reduced to twenty units out of the twenty-four ( three units under renovation plus one under maintenance).

D) Number of boats through the lock (Table 5)

This is not an operating result, but together with the amount of automobile traffic over the bridge, it provides a convincing illustration of how well the plant fits in with its environment. A marina opened in 1976 temporarily slowed-down the upward trend of boat transits.

Incidentally, half the annual transits take place in July and August.

E) Costs (Table 6)

These figures are accountancy data corrected by application of the gross internal product index.

Note the increase in operating costs since preventive maintenance and its low value (23 %) in relation to the total cost, a major proportion of which is accounted for by financial costs.

## 3. - ANTICIPATION, DETECTION AND IDENTIFICATION OF FAULTS

### 3.1 - General remarks

We now come to a brief description of a few problems experienced at La Rance, with some examples of our Service's electrical and mechanical equipment maintenance policy.

The La Rance plant is one of four hundred and forty-nine water-power plants run by E.D.F. Though certainly one of the most famous of these, it is only the sixteenth in order of out put. Its total output is 1,800 MW, with an annual productivity of roughly 58,000 GWh. My purpose in stating this is not so much to emphasize that we are the biggest water-power company in the world (of which we are of course very proud !), but rather to demonstrate the degree of interaction of the system.

Whenever a major problem arises or is solved at one of the 448 other power plants, someone starts thinking about how it might apply to La Rance, that is to say in the form of a risk, an improvement, or possible profit for example. Whenever there is a major problem at La Rance, or one is solved, the information is transmitted in the reverse direction. No single power plant maintenance system can be considered independently of the others.

As a second preliminary comment, we have developed a planned inspection and maintenance procedure (MECEP) which is based on the philosophy that overall preventive inspection aimed at anticipating the probable behaviour of plant and equipment so as to be able to prepare for maintenance at an acceptable time is wiser than having to deal with unexpected trouble. In other words, we would rather pay for preventive medicine for water-power plant equipment and structures than for remedial surgery. In this, I believe we differ considerably from our colleagues in this country.

My third and final remark before the examples is that dismantling conditions at La Rance (and for submerged horizontal units in general) differ appreciably from those for conventional low-head vertical units. While the major turbine components - e.g. blades, faired nose-cap - are more conveniently accessible (in roughly one-third of the time) on bulb units and subject to a few precautions, all medium-size components can be removed through the access shaft. The generator stator on the other hand is much more difficult to get at and takes five times as long to dismantle.

To fully appreciate the importance of this remark, one must have witnessed a scheduled inspection and seen how plant operators react if a single drop of seawater has found its way inside a unit : very much like a competent mechanic seeing a drop of oil leaking from a powerful motor he has just reassembled.

### 3.2 - Special study of problems experienced with the electromechanical equipment at La Rance

A) Before going any further, I think we should bear in mind that the equipment at La Rance has behaved very satisfactorily on the whole, as shown by the operating results quoted in para. 2.2. Standing before this 'assembled court' of technicians, however, I feel that an objective description of some of the problems we have had to deal with on various items of equipment can hardly be out of place.

B) Marine corrosion

Considering that we are just by the ocean I do not think I need waste words in explaining our misgivings as regards corrosive seawater action on the plant equipment.

In our efforts to reduce this to a minimum, we relied on careful design, a suitable choice of materials, effective coatings and linings and provision of an applied-current cathodic protection system. Here, I will simply say that the latter was remarkably designed and has been eminently successful in operation.

Here are a few figures :

Each unit is protected by 36 anodes, each gate by 4 anodes and the lock by 6 anodes. Tantalum anodes are provided throughout, with a 50 $\mu$m platinum surface coating (for a consumption rate of 10 $\mu$m in 30 years). Potential is maintained at roughly 300 to 1,200 mV, for a current density of 170 $mA/m^2$ of protected surface area. The overall annual energy requirement for the system is 130,000 kWh and a 1 $cm^2$ anode area protects an unpainted metal surface area of 1.5 $m^2$.

C) Gates

With paint as their only protection, the gates initially suffered a certain amount of corrosion, especially as the paintwork was vulnerable to damage by flotsam. Installation of the cathodic protection system was completed in 1972, since then all we can say about the gates is "nothing to report".

D) Turbines

D.1 Guide-vanes

Overtightness of the self-lubricated plastic bearing shells on initial assembly caused a few problems, but these were very soon put right.

D.2 Resistance to cavitation

No significant indications of cavitation were observed on the throat ring and blades after 100,000 hours under water, including 80,000 hours in operation.

The absence of blade damage can no doubt be attributed to the use of cupro-aluminium and 17/4 (17 % Cr, 4 % Ni) stainless-steel for these components.

D.3 Runner blade leakage

Serious runner blade problems were experienced, with water penetrating through the seals and mixing with the turbine oil after a certain amount of seal wear. The trouble was remedied by increasing the internal turbine oil pressure and fitting annular or herringbone-pattern seals of a carefully-selected material.

D.4 Labyrinths

The labyrinths, wetted with seawater from which all sediment has been settled out, are in excellent condition, which is quite surprising after fourteen years of service. They do not show any signs of abrasion, cavitation damage or corrosion ; the only signs of service are a few tiny mussels clinging fast at the bottom of cavities, which probably helps to reduce leakage. A good thing for production !

D.5 Shaft seals

Some trouble was experienced initially with the shaft seals, in the form of abnormal wear of the fourth sealing ring (a carbon ring like two of the others ; the last ring is of high-resistance polyethylene). Decisive modifications consisted in improving lubrification and increasing the seal bearing pressure against the wearing component. Thirty-two different schemes were tried out.

From 1976 onwards, signs of severe corrosion damage were observed on operating components inadequately covered by the cathodic system. We are trying to solve the problem by improving the cathodic system layout.

#### D.6 Couplings

These will be discussed in para. 3.3. later on as they come under the general category of high-capacity bolted assemblies which have been the subject of very detailed study by our Service.

### E) Generators

#### E.1 Iron circuit

On January 1st 1975, a serious incident caused the stator of a unit to foul the rotor. Subsequent inspection of the unit and a thorough 'post-mortem' showed a design fault to have been responsible, a minor component having been insufficiently dimensioned for the winding stresses and repeated loads on the iron circuit caused by 500 startups and 7500 asynchronous couplings. To prevent further similar trouble, the most severe cycles were eliminated and a preventive renovation schedule was put into effect, covering three machines yearly. It was intended to apply this procedure from 1975 to 1983, but thanks to satisfactory services from certain manufacturers, we will now be able to end it by the beginning of 1982. There was also some trouble with the iron circuit clamping and securing arrangements.

#### E.2 Stator

Insulation measurements at regular intervals occasionally produced comparatively low values after several days at a standstill. This was attributed to the fact that the only access to the turbine runner is through the generator. Appropriate shielding has now been installed to protect the insulation from 'micro-damage', but for our new designs, we will try to provide direct access to the runner from the downstream end.

#### E.3 Damping bars

The first case of rotor damping-bar damage, in the form of severe electro-erosion, was observed at one of our other power stations. Inspection of the units at La Rance showed several poles to have been affected. This trouble was remedied by modifying the bar distribution, reducing tolerances, and by argon-welding and caulking the bars secure in position.

#### E.4 Brushgear

Slipring and brush performance was unsatisfactory at the start, with wear as high as 10 mm per 1000 hours in some cases. Many different types and makes of brush were tried, but the drastic improvement was obtained by enclosing the sliprings in a separate compartment accessible for inspection and under depression to eliminate dust. The brushes which were finally adopted are charged with silver.

The units at La Rance unfortunately could not be provided with static excitation as this type of system did not come into use in France until three years after the plant was commissioned.

E.5 Pressurization system

Pressurization at 2 bars absolute assists the heat-exchange processes taking place in the unit (heat-exchange being proportional to the specific mass of air). An output of 10 MW is obtained, whereas it would be limited to 7 MW in atmospheric air. No problems have been experienced with this equipment.

F) Preliminary conclusions

The so-called 'series effect' does not apply to water-power equipment. La Rance must be one of the very few examples in the world of a plant equipped with more than ten similar units. Successful results obtained from one unit can therefore be generalized for twenty-three units, but unfortunately, the same also applies to unfortunate incidents on a single unit.

3.3 - Examples of anticipation, detection and identification experience for mechanical equipment at EDF water-power plants

A) As Table 7 shows, most of the effects observed at La Rance can be generalized to apply to the 220 bulb units we are operating. Many of these effects can also be extrapolated to the 1,500 units of all types operating at EDF water-power stations. However, reverting to your particular subject of interest, I will now explain our maintenance policies for a few examples of unitary, similar and strictly repetitive equipment.

B) Strictly repetitive equipment

B.1 Circuit-breakers

Many circuit-breakers operate in power plant which is frequently put into and out of service in order to follow the system load curve, but are only required to deal with low current. Thus, although the breaker chamber is operating well within design limits, the operating mechanism and mechanical components in general have to withstand very severe conditions.

The policy of EDF, therefore, has been to define a preventive accelerated-ageing test which, of course, includes critical breaking conditions, but also a series of 10,000 breaker operations. Each new breaker design has to pass this test before it can be accepted for use in the EDF power system.

B.2 Hydraulic operating controls

These are mainly used for gate and turbine blading operation. Here again, and in addition to strict dimensioning rules, we specify preventive approval tests. Solenoid valves for example are tested especially for their ability to operate after prolonged service and without voltage. For safety operations, systematic use is made of electric distributors with without-voltage operated pilot valves.

C) Similar equipment

Potentially-destructive prototype tests cannot be envisaged for this type of equipment.

C.1 Generator iron circuits

From our unfortunate iron circuit trouble experience and with the assistance of our usual suppliers of equipment, we have established a few rules which we consider to be essential. They apply to design (monolithic character, clamping and securing arrangements), and in addition, to manufacture (cutting-out and deburring of laminations, insulation) and to assembly and installation (intermediate stack-compression, acceptance tests).

C.2 Pelton turbines

Pelton buckets are subjected to superimposed vibration and impact stresses, which can only be determined by telemeasurements on the actual unit. Knowing these stresses and the maximum safe number of impacts without trouble from fatigue or fatigue failure, and since tests in air have been found to be representative, the following action was possible in each case :

a) Preventive maintenance on runners in service

b) Correct dimensioning of new runners.

Specially-designed, closely-adjusted vibration detectors are provided to give warning of abnormal operation.

## D) Unitary equipment

Such items of equipment all differ from each other. Experience can yield no more than design principles, to which one can also add general supervision methods.

### D.1 Couplings

This brings us back to a problem experienced at La Rance and with bulb units in general.

Heavy rotating components overhung on a horizontal shaft require assembling by means of studs and nuts 70 mm to 100 m in diameter and of short length compared to their diameter. The studs are subjected to severe cyclic stresses during rotation. Correct stud tightness is achieved by the following :

a) Machining the nut bearing surfaces, ensuring they are truly perpendicular to the stud centerlines.

b) selecting more elastic steel for the stud than for the nut.

c) Tapered extruded threads radiused at the bottom.

d) Maximum possible shank length between threaded portions, thinner than the latter with a transition radius to the thread diameter.

e) Tightening by controlled elongation of the stud under heat or using a jack.

A very rigourous method as been made up with help of makers.

Here again, the vibration pickups give warning of abnormal conditions.

### D.2 Transformers

The transformers are monitored during operation by periodic cinematography of dissolved gas in the oil. Faults are identifiable by the type of gas they produce. By this preventive measure the consequences of transformer faults at French water-power stations were reduced to roughly one-fifth in eight years. Single major faults are located by a system using ultrasonic waves emitted by partial discharges.

E) Preliminary conclusions

I hope I have not bored you with this long enumeration, but as any specialist will agree, there are many more items that I could have mentioned, such as dielectric strength measurements on generators for example.

In giving you these examples, I wanted to show you our general policy for the generally heterogeneous equipment that we operate day in, day out for 24 hours on end (or 25 hours if possible !) for 365 days of the year (sometimes 366 or 367 days).

Instead of 'detection, identification, anticipation', we prefer a policy in the following order :

1) Anticipation or prevention based on experience gradually acquired with closely-related, similar or identical machines, relating to design, acceptance tests, and checks required in service.

2) Detection with the aid of carefully adjusted overall protection systems checked at regular intervals and each designed to respond to a variety of causes.

3) Identification (and repairs), also based on experience acquired by plant operators and in repair shops.

3.4 - Final conclusions

A) Regarding La Rance and tidal plant in general, we consider that, with present experience, it should be possible to avoid the hazards normally associated with new forms of energy, but that one should also be careful not to expect too much from tidal energy at the present stage.

There is a limit to the amount of power that can be produced from tides, the structures at a tidal power site are of a size which will require very substantial capital investment, and their geographical extension is such that their effect on the environment will concern far more people than is the case for other forms of power plant.

Even though it is by no means certain that present technological progress is sufficient to ensure immediate cost-effectiveness of tidal energy in France, it seems highly probable that it will compare favorably with other forms of modulable energy in the future. The cost price for assumed implementation in 1980 of a project by methods applied in 1960 would amount to $1.45 \times 10^9$ French Francs for a 240 MW out put ($\$ 355 \times 10^6$), i.e. 6000 F/kW ($ 1470/kW). Since, as we have seen, operating costs (excluding major maintenance) amount to 4 c/kWh, for $540 \times 10^9$ kWh, the total cost calculated by EDF rules would amount to 33 c/kWh (81 mill/kWh), which is comparable with the figure for fuel power plant and well below the cost for other sources of energy (e.g. wind and solar energy) being developed at the present time.

Finally, from our own experience, we would unhesitatingly opt for bulb units (as we do for river plant) for a potential new French power plant for which we are presently engaged in effective preliminary development studies.

In particular, one should compare the cost of the additional complication of a two-way unit also operating as a pump with the additional return obtainable from this type of operation. If the comparison showed a one-way unit to be preferable, one would no doubt be well advised to set the unit at a higher elevation.

B) As I have already mentioned, we prefer, 'anticipation, detection, identification' to 'detection, diagnosis prevention'. However, I also said that our system is based on an exchange of experience and information. To conclude, therefore, I would like to add that this system includes the following :

a) A 'non-availability' card-index system compiled by local plant operators, the data of which is processed at a central station, the results being used for information, locating similar cases and later on, compiling statistics.

b) Generalized personal contacts and meetings to discuss and analyze points of individual concern and problems, thus providing information from which a few specialists can establish a 'doctrine' of instructions and principles.

   The main effects of this are as follows :

   - Repetitive equipment : Systematization of preliminary and representative prototype tests.

   - Unitary equipment : Availability of a list of attempted solutions. Practically systematic location of similar or closely-related cases.

- Orientation of technical or technological action and justification for investment : Subsequent application of statistics based on operating results from a large number of power stations.

## TABLE 1

## STATISTICAL DATA FOR BULB UNITS OPERATING IN FRANCE

1. 'Micro' bulb units (up to 1.5 MW)

| | |
|---|---|
| Number in operation | : about 120 |
| Individual output | : 38 kW to 1,400 kW |
| Average output | : 490 kW |
| Head | : 1.5 m to 17.4 m |
| Rated discharge | : 1.9 $m^3/s$ to 39 $m^3/s$ |
| Runner diameter | : 630 mm to 3,000 mm |
| When commissioned | : between 1954 and 1979 |

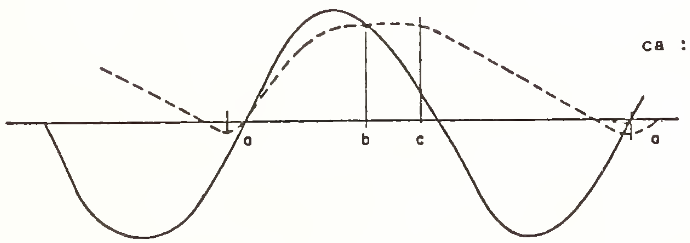
2. Other bulb units (over 2 MW)

| | |
|---|---|
| Number in operation | : about 100 |
| Individual output | : 5 MW to 40 MW |
| Average output | : 21 MW |
| Head | : 3.4 m to 16.5 m |
| Rated discharge | : 38 $m^3/s$ to 410 $m^3/s$ |
| Runner diameter | : 2,500 mm to 6,900 mm |
| When commissioned | : between 1957 and 1979 |

Note for reference :

Rock Island bulb units (1978) :

| | |
|---|---|
| output | : 54 MW |
| diameter | : 7.4 m |
| head | : 12.1 m |

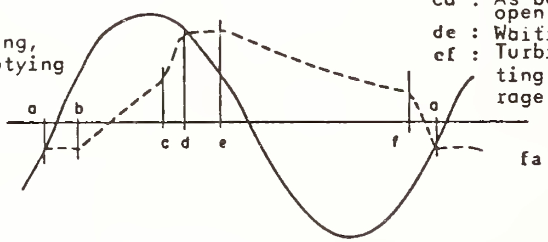



ab : Gates open, storage pond filling-up

bc : Everything closed, waiting period

ca : Turbines operating, storage pond emptying

1) ONE-WAY (EMPTYING) CYCLE




ab : Waiting period

bc : Turbines operating from sea to storage pond

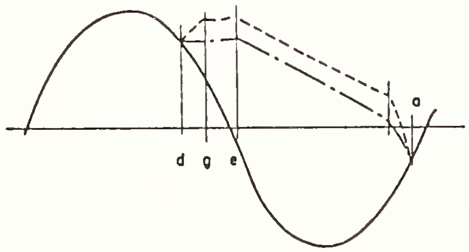
cd : As bc + gates open

de : Waiting period.

ef : Turbines operating from storage pond to sea

fa : As ef + gates open

2) TWO-WAY CYCLE



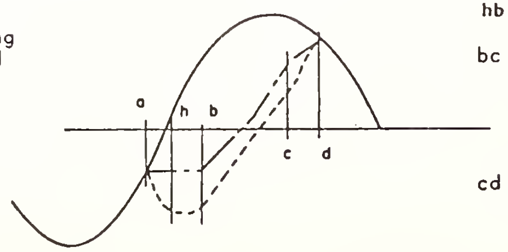

dg : Pumping

ge : Waiting period

ef : Turbines operating from storage pond to sea

fa : As ef + gates open

3) PUMPING WITH ADDITIONAL FILLING OF STORAGE POND




ah : Pumping

hb : Waiting period

bc : Turbines operating from sea to storage pond

cd : As bc + gates open

4) PUMPING WITH ADDITIONAL EMPTYING OF STORAGE POND

TABLE 2

——— Sea level

- - - - - Storage pond level

TABLE 3

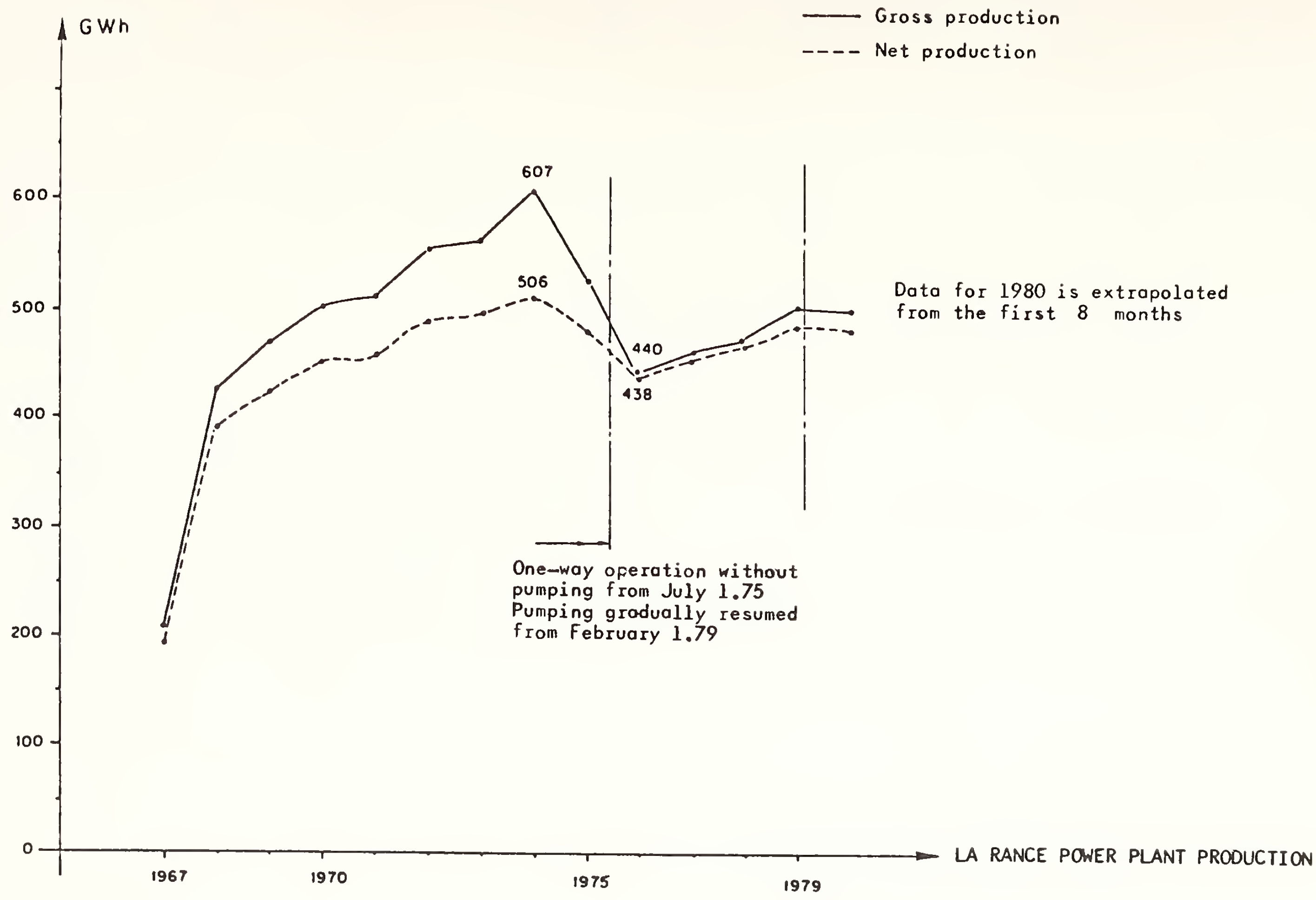

GWh
Gross production
Net production
607
506
440
438
Data for 1980 is extrapolated from the first 8 months
One-way operation without pumping from July 1.75
Pumping gradually resumed from February 1.79
600
500
400
300
200
100
0
1967
1970
1975
1979
LA RANCE POWER PLANT PRODUCTION

TABLE 4

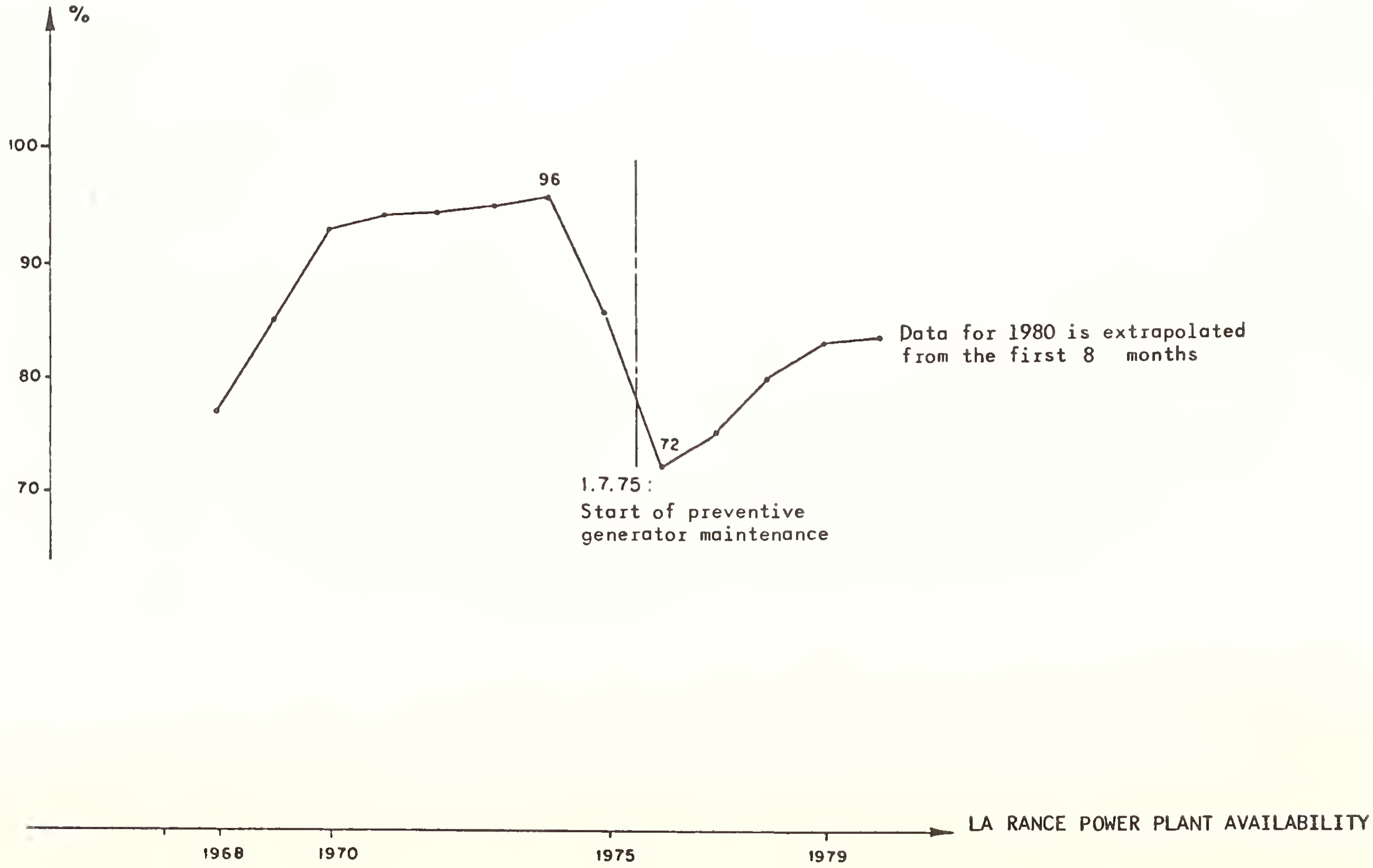
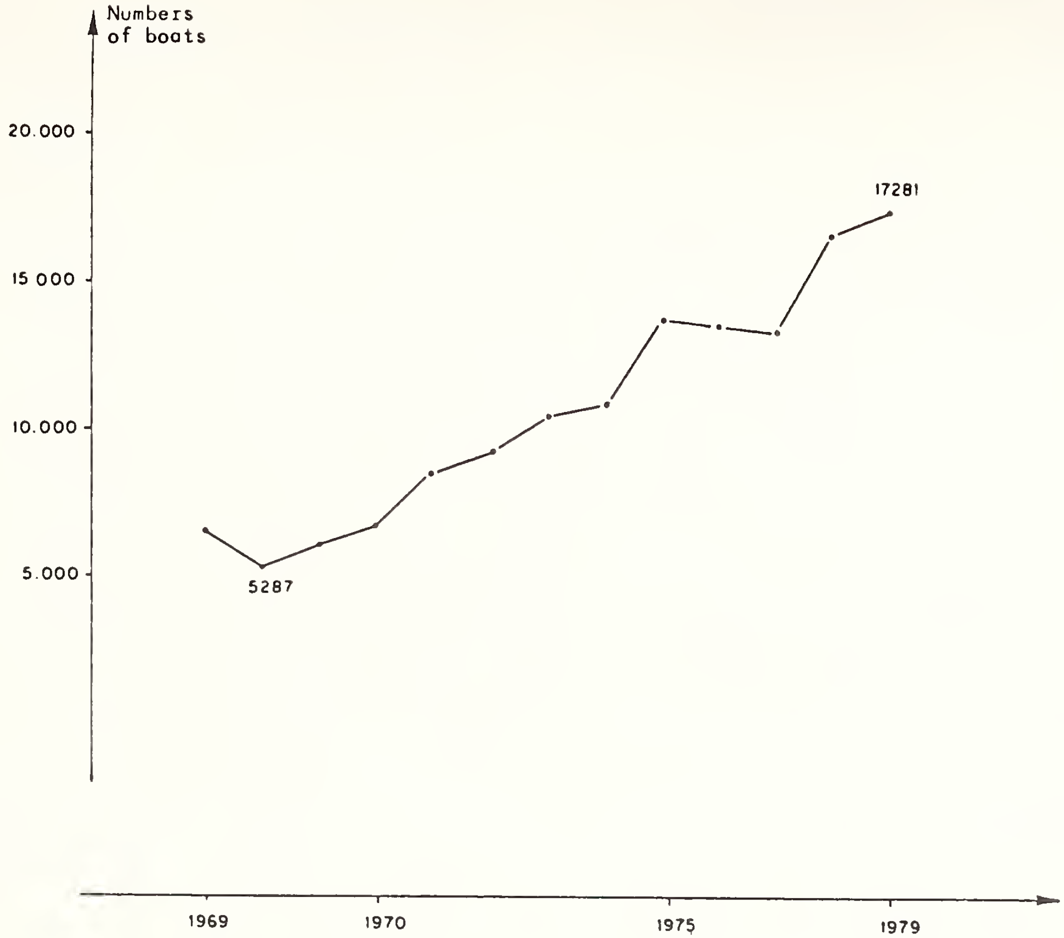
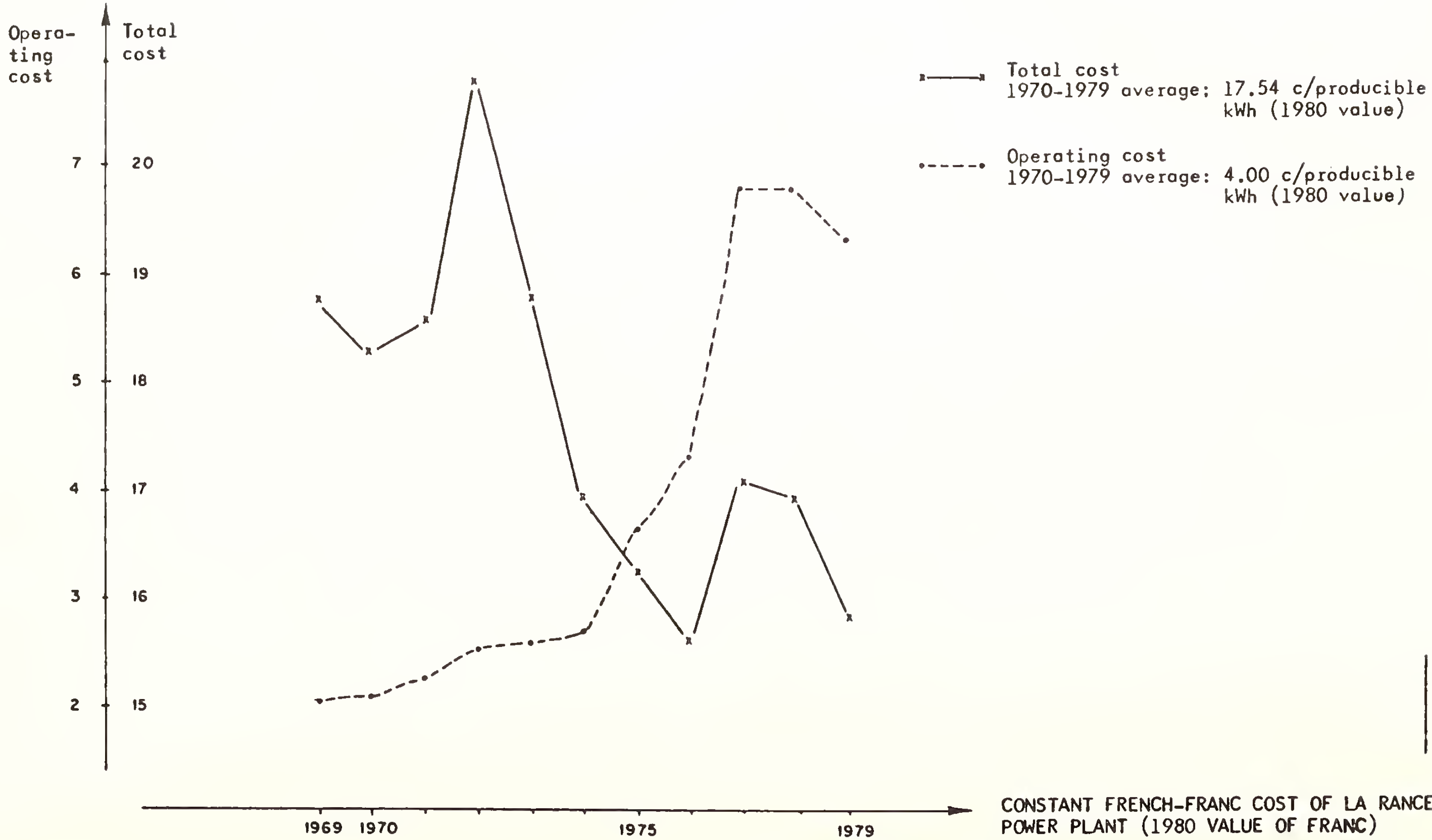

%
100
90
80
70
96
72
1.7.75 :
Start of preventive
generator maintenance
Data for 1980 is extrapolated
from the first 8 months
1968
1970
1975
1979
LA RANCE POWER PLANT AVAILABILITY

TABLE 5




NUMBER OF BOATS THROUGH LA RANCE
PLANT LOCK

TABLE 6




CONSTANT FRENCH-FRANC COST OF LA RANCE POWER PLANT (1980 VALUE OF FRANC)

Nota : Data corrected by gross internal product index
1 centime ≃ 2.45 mill.

TABLE 7

GENERALIZATION OF PROBLEMS EXPERIENCED WITH ELECTRICAL AND MECHANICAL EQUIPMENT AT LA RANCE

| Affected equipment | Problem specific to La Rance | Problem specific to bulb units | Problem generally associated with water-power equipment |
|---|---|---|---|
| Corrosion | X (none) | | |
| Gates | X (none) | X | X |
| Turbines : | | | |
| Guide vanes | X (initially) | X | X |
| Cavitation | X (none) | X | X |
| Blade leaks | X (initially) | X | X |
| Labyrinths | X (none) | X | X |
| Shaft seals | X (yes) | X | X |
| Couplings | X (initially) | X | X |
| Generators : | | | |
| Iron circuits | X (yes) | X | |
| Stator | X (initially) | X | X |
| Damping bars | X (initially) | X | |
| Brushgear | X (yes) | X | X |
| Pressurization system | X (none) | X | |

Other equipment not mentioned above : no major problems.

# DETECTION, DIAGNOSIS AND PROGNOSIS IN GEOTHERMAL WELL TECHNOLOGY

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Abstract: For successful and safe operation of a geothermal well, the condition of the casing and cement must be accurately determined. Measurements on casing wall thickness, corrosion damage, holes, cracks, splits, etc., are needed to assess casing integrity. Cement bond logs are needed to detect channels or water pockets in cement behind pipe and to determine the state of the cement bond to the pipe and formation. Instrumentation for making such measurements is limited by the temperature capabilities (<175°C) of existing logging equipment developed for the oil and gas industry. This paper reviews the instruments that are needed for geothermal casing and cementing inspection, identifies the principal deficiencies in their high temperature use, and describes Sandia's upgrade research program on multi-arm caliper and acoustic cement bond logging tool. The key electronic section in a multi-arm caliper will consist of 275°C circuits designed by Sandia. In an acoustic cement bond logging tool, a simple circuit with possibilities of using commercially available components for high temperature operation is being developed. These new tools will be field tested for operation at a minimum temperature of 275°C and pressure of 7000 psi for up to 1000 hours.

Key words: Casing inspection; cement bond logging tool; geothermal technology.

Introduction: Downhole conditions very often cause casing and cementing problems in geothermal wells as illustrated in Figure 1. Proper engineering development and production of a geothermal field requires periodic inspection and evaluation of the wellbore's casing and cementation. According to a recent study by Knutson and Boardman (1978), the most common tools for assessing casing integrity are spinner surveys, radioactive tracer surveys, electromagnetic surveys, and calipers. The logs commonly used to evaluate cement bond are temperature logs, nuclear cement logs, noise, and acoustic cement bond logs (CBL).

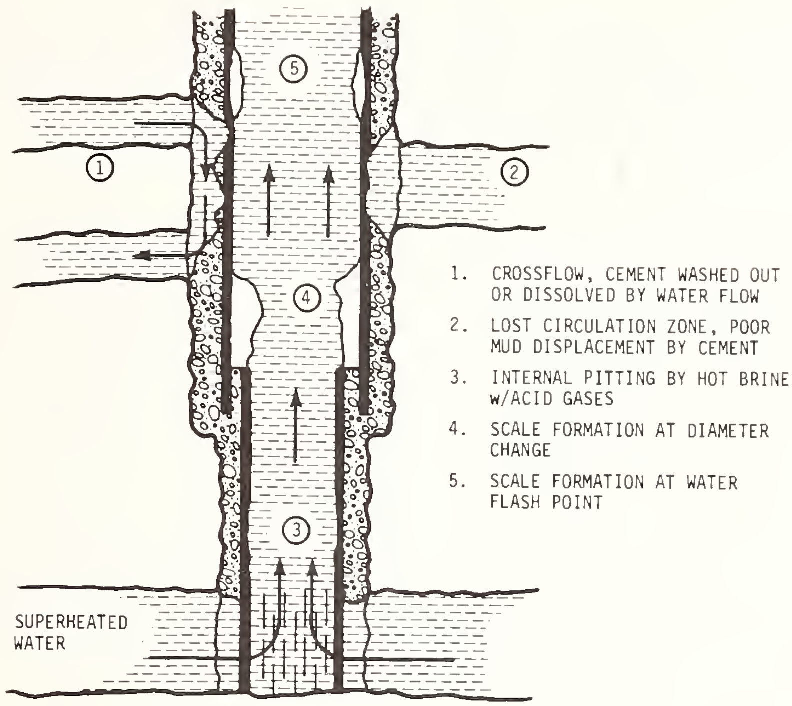


Figure 1. Casing and Cementing Problems in a Geothermal Well

A spinner-type flowmeter responds to and locates fluid flow through holes in the casing. Fluid leakage sometimes also causes temperature anomalies. A differential temperature log describes the slope of the absolute temperature curve and is capable of pinpointing these anomalies even though they may be minute. Sandia National Laboratories has developed prototypes of the flow tool and the temperature tool for operation up to 275°C. (Veneruso, 1979). All the component parts for these tools will soon become commercially available.

Radioactive tracer surveys are useful in locating casing leaks by logging the movement of injected tracers. These surveys are very useful in detecting low flow rate leaks.

Commercially available downhole electromagnetic surveys utilize a coil for generating a magnetic field. The phase changes detected by the receiving coil are related to the casing wall thickness. If the inner diameter is known (this can be measured by a caliper), then the electromagnetic surveys will reveal the outer diameter's condition and thereby indicate the extent of pipe wear and corrosion.

Caliper tools have multiple expanding arms that make contact with the casing walls. The diamter recorded is that of a circle described by the tips of the arms. The caliper log is useful in analyzing corrosion damage, scale buildup, collapsed and parted casing, and casing breaks.

In oil and gas wells, a temperature log is also used to detect the cement top when the hydrating cement undergoes an exothermic reaction that provides a temperature anomaly. Unfortunately, temperature anomalies are common in geothermal wells, and therefore, the interpretation of cement tops is ambiguous. Also, this log must be run while the cement is reacting because afterward thermal equalibrium is achieved and no information can be obtained about the position of cured cement behind the pipe.

The nuclear cement log functions as a modified density log. If there is a density contrast between the cement and the mud or fluid being displaced by the cement, the log provides an estimate of the amount of cement fill at any position in the well by indicating the material density.

The noise log is a display of high frequency sound being produced at any vertical location in the wellbore. An evaluation of this log can pinpoint areas in a well where fluid flow channels occur behind the pipe.

The acoustic CBL tool determines the quality and extent of the physical bond between the casing pipe and the surrounding cement sheath, and between the cement and formation. This log measures the amplitude of the acoustic signals from the casing pipe and the amplitude of a later arrival which reflects the cement bound to the formation.

There are currently no commercially available high temperature cement bond and casing integrity logging systems for geothermal wells with maximum temperatures in excess of 275°C. Most commercial logging tools become unreliable above 175°C in actual field tests. As a consequence, operators must cool the well in order to run surveys. The thermal shock in cooling could cause well damage. Also, loss in production time is expensive. Therefore, high temperature logging capabilities are needed for rapid development of geothermal energy.

Sandia's R&D objective is to develop and commercialize high temperature tools for operation up to 275°C. The technology must be adaptable to the industry to stimulate the development of a large enough market such that commercial suppliers and logging companies can provide the required hardware services on a routine basis. Although the tools described above all provide information on casing and cementing conditions, the multi-arm caliper and acoustic CBL tool are the most versatile and widely used tools in the field. For this reason, we select them as our first set of tools for upgrading.

Table 1 summarizes casing and cementing inspection tools, their limitations in geothermal applications, and the recommended upgrading procedures. The following section will describe the upgrade procedures on a multi-arm caliper and an acoustic CBL tool in more detail.

Sandia's Upgrade Research Program: 1. Multi-arm Caliper - The casing caliper utilizes a single conductor cable plus ground for all power and data transmission. This is accomplished through a multiplexer built into the tool's electronics. The tool uses potentiometers that convert the motion of mechanical arms into electronic signals that are proportional to the casing pipe's inside dimensions. In general, commercially available calipers are specified for operation up to 175°C, because of the temperature limitations of their electronic circuits, DC motors and dynamic seals. Therefore, we propose to use high-temperature-rated linear or rotary displacement transducers (LTDs or RDTs), if high temperature potentiometers cannot be found. As shown in Figure 2, the tool's electronics will consist of a high temperature voltage-to-frequency converter, pulse stretcher, line driver, voltage regulator and a multiplexer. These 275°C circuits were designed by Sandia and will be manufactured by Teledyne and General Electric-Houston. Additional R&D may be necessary to develop the high temperature DC motor that is used to retract the caliper's mechanical arms. In addition, high temperature elastomeric seals or metal bellows may be required to prevent brine from entering the tool's electronic and motor housing compartment.

2. Acoustic CBL Tool - Figure 3 illustrates the cement bond sonde and the related signals. Casing suspended freely, i.e., uncemented, in a well transmits sound energy with relatively little weakening of the vibrations between the transmitter and receiver. However, when hardened cement is behind the casing and when this cement is properly bonded to the pipe and formation, the vibrations sent out from the tool to the casing will impart similar motion in the cement but at a lower velocity. From the strength and the time of the vibrations received, one can determine the degree of cement bonding. A recording of the free pipe is presented on each log as a reference for evaluating the other signals (Figure 4). In the variable density log, strong casing signal and weak formation signal is an indication of free pipe.

TABLE 1

TOOLS COMMONLY USED FOR CASING INSPECTION AND CEMENT BOND EVALUATION

| Tools | Geothermal Limitations | Upgrade Recommendations |
|---|---|---|
| Casing Inspection<br>Spinner Survey | None - Technology is available | None - only field test verification |
| Radioactive Tracer | Operating Temperature <175°C<br>Possible Environmental Impact | Design high temperature circuit |
| Electromagnetic Tool | Operating Temperature <175°C | Redesign circuits using high temperature electronics |
| Caliper | Operating Temperature <175°C | Use high temperature electronics<br>Select high temperature dynamic seals<br>Develop high temperature motor<br>Or use mechanical design to eliminate motor and thus the need of dynamic seals |
| Cement Bond Evaluation<br>Temperature Tool | None - Technology is available<br>Mainly for locating cement top and major void | Extend into a differential temperature logging tool for casing inspection |
| Nuclear CBL Tool | Operating Temperature <175°C<br>Possible Environmental Impact | Develop high temperature op amp and detector |
| Noise Tool | Subject to Operator's Interpretation<br>Operating Temperature not crucial (not being used in production wells) | |
| Acoustic CBL Tool | Electronics limited to <175°C | Develop high temperature SCR<br>Replace circuit with high temperature electronics |

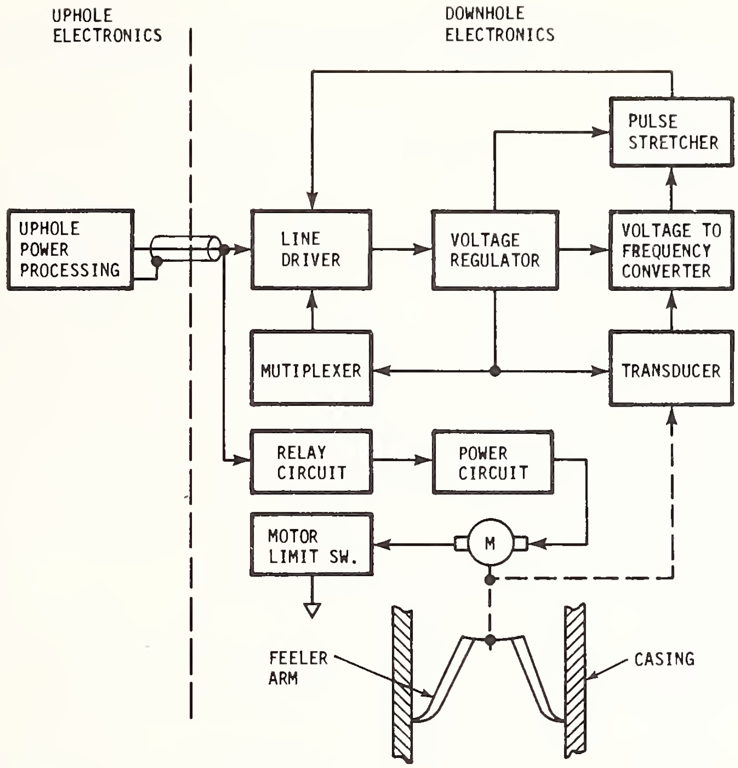


Figure 2. Block Diagram for a Caliper

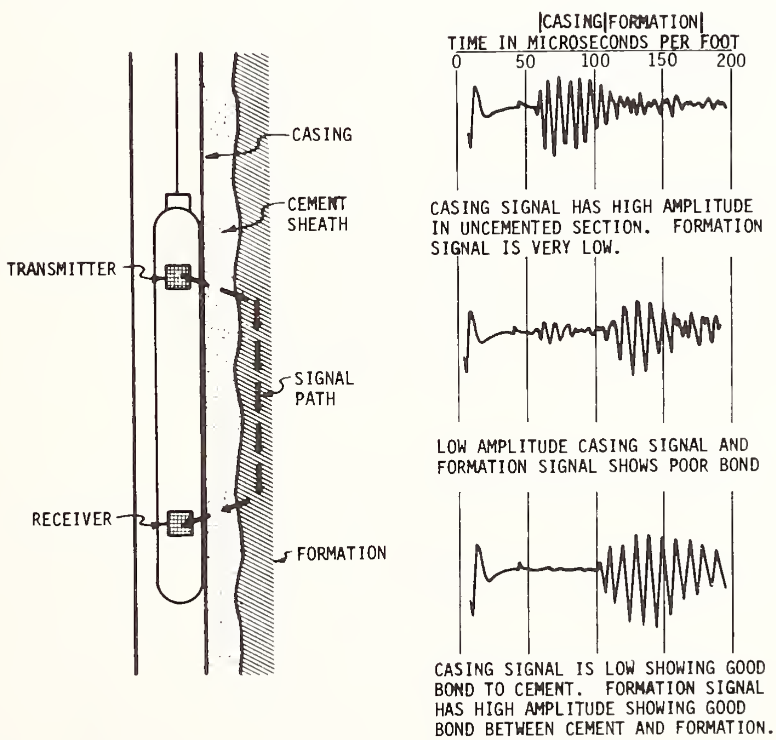


Figure 3. Cement Bond Sonde and Acoustic Signal

A well bonded casing with good cement-to-formation bond will show strong formation signal and weak casing signal.

Some of the commercial CBL tools contain very complex electronic circuits. These systems use many sensors and require complex circuitry in order to provide precise clocking time and signal transmission under a wide variety of conditions in oil and gas wells. However, it is extremely difficult to upgrade a sophisticated circuit for high temperature operation since most commercial electronics are specified for operation only up to 175°C. Therefore, a simplified CBL circuit is essential before any upgrading can take place. In an acoustic CBL tool for operation in geothermal fluid, a single transmitter and a single receiver with adequate spatial separation should provide sufficient information about cement bond conditions in the well. Also, in order to minimize the downhole electronics, signal processing and data reduction will be done uphole.

Figure 5 is an example of a straightforward circuit that is capable of generating a short high voltage, high current pulse for an acoustic transmitter at room temperature. Here, $C_1$ is used to supply energy to the SCR switch and the acoustic transmitter when the line driver cuts off input power. $L_1$ is used to boost the voltage at $C_1$. When the SCR conducts, $L_1$, $L_2$, and $C_2$ prevent the ripples from feeding back to the input. To provide a reference for the returned signal, the input line detects the transmitting time through $C_4$. $R_1$, $R_2$, and $C_5$ limit the imposed voltage on the unijunction transistor which will not be triggered until the capacitor $C_3$ is sufficiently charged. Thus, the $R_3$ $C_3$ time constant will determine the firing frequency of the SCR. Transformer T sharpens and boosts the amplitude of the trigger pulse to the SCR. On Figure 5 we have not specified component values because their values would be different for different acoustic transducers and temperature ranges. We are currently working to develop a high temperature version of this circuit.

Commercial SCRs (Semiconductor Controlled Rectifiers) rated at 175°C or lower place a limitation on a tool's operating temperature. We have initiated a project to upgrade the SCR for operation at 275°C. It is expected that the theory and optimal fabricational approach will be determined by September 1981 and a "GaP" or "GaAs" SCR will be ready in prototype by October 1982. We are also exploring the possibility of combining a thermoelectric cooler with an SCR in order to run the unit in an ambient of 275°C while the temperature of the SCR chip is only 175°C. Commercial vacuum tubes, such as sprytrons and thyratrons, are other possibilities that

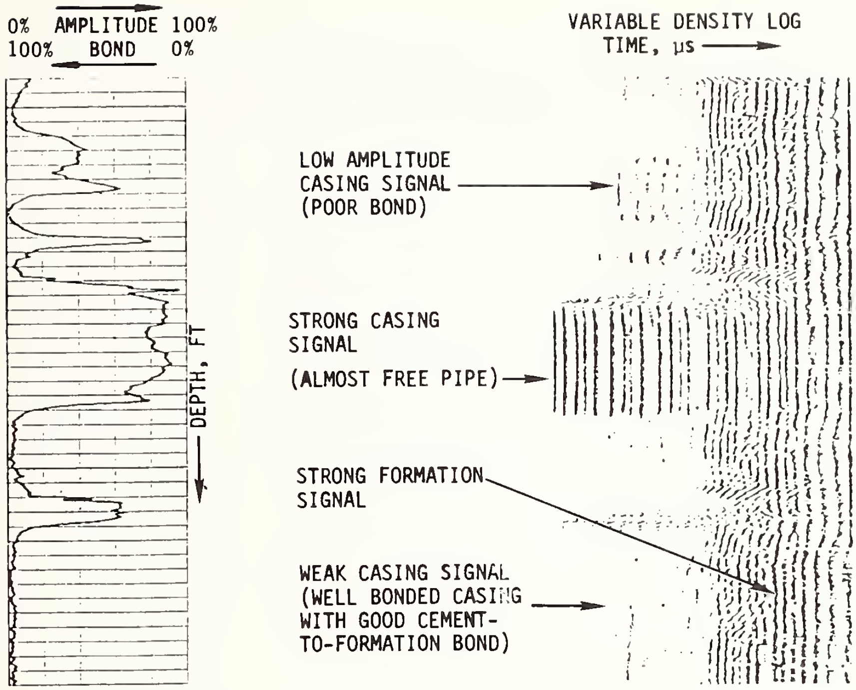


Figure 4. Acoustic Cement Bond Log

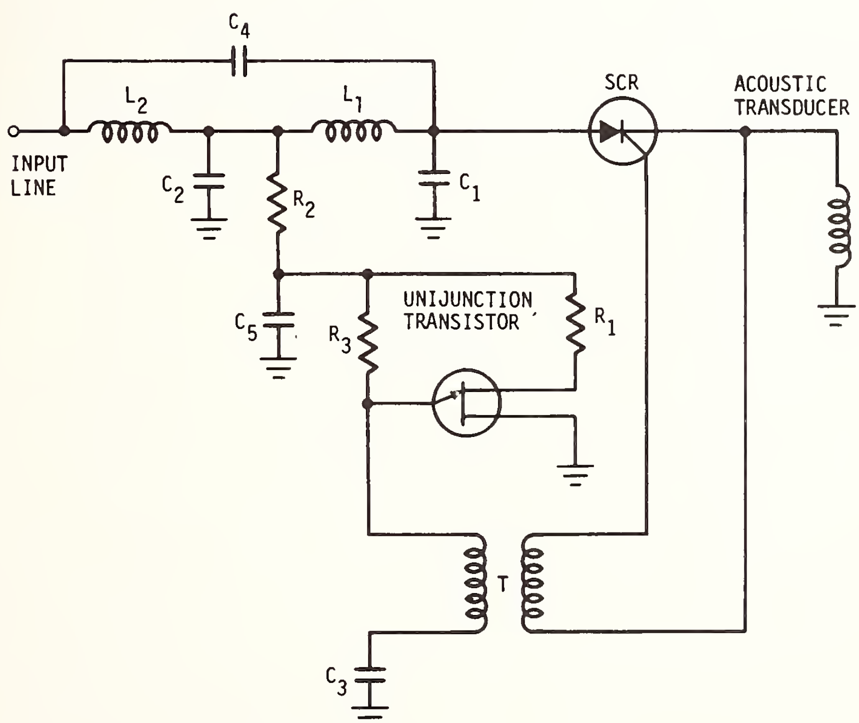


Figure 5. Cement Bond Transmitter Schematic

might be used as a substitute for an SCR. In these options, the shortcoming is in the limited number of shots that can be guaranteed; we are investigating the possibility of modifying a tube's design for longer life time.

Return signals at the receiver must be amplified so that they can be detectable uphole. Specially selected commercial junction field effect transistors (JFETs) can withstand high temperature and will be considered. Another possibility is the electronic tubes. Although, an electronic tube amplifier may operate stably only up to 250°C as reported by Cannon (1979), development of integrated thermionic circuits has resulted in a much higher operating temperature (McCormick and Wilde, 1980). In addition, Harris Semiconductor Corp. is working under contract with Sandia to develop a high-temperature operational amplifier (Ohr, 1980). Other commercial products that may be used to upgrade a tool for geothermal application are summarized in Table 2. For a step-by-step 275°C fabrication technology, one may refer to a report by Bonn and Palmer (1980).

Summary: In this report, we describe a project to upgrade a multi-arm caliper and an acoustic cement bond logging tool for operation at a minimum temperatue of 275°C and pressure of 7000 psi for up to 1000 hours. The commercially available materials and devices, and the electronic components developed by Sandia National Laboratories for high temperature operation will be fabricated and field tested in partnership with industry. The final stage of this project is to commercialize the design and transfer the technology to industry.

TABLE 2

MAXIMUM OPERATING TEMPERATURE FOR SOME COMMERCIAL PRODUCTS

| Company | Item | Maximum Temperature |
|---|---|---|
| | <u>Resistors</u> | |
| Caddock Electronics | Thick Film Chips | 500°C |
| Cermalloy | Thick Film Inks | 500°C |
| | <u>Capacitors</u> | |
| Philips (MEPCO) | Solid Aluminum Electrolytic | 300°C |
| Custom Electronics | High Voltage | 300°C |
| Sprague | Thin Film $SiO_2$ | 300°C |
| Cermalloy | Thick Film Inks | 500°C |
| | <u>Transformers</u> | |
| General Magnetics | Transformers | 500°C |
| | <u>Conductors</u> | |
| Permalustre | Anodized Aluminum Wire | 500°C |
| Hy-Temp Transducers | Ceramic Coated Copper Wire | 500°C |
| Cermalloy | Conductor Inks | 300°C |
| | <u>Solder</u> | |
| DuPont | High Temperature Paste | 300°C |
| | <u>Epoxy</u> | |
| Ablestick | Conductor or Dielectric | 300°C |
| | <u>Transistors</u> | |
| Motorola | JFET | 300°C |
| | <u>P. C. Board</u> | |
| DuPont | Polyimide | 300°C |
| | <u>Packages</u> | |
| Tekform | Metal Packages | 350°C |
| 3M | Ceramic Packages | 350°C |

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# APPENDIX

MECHANICAL FAILURES PREVENTION GROUP

DETECTION, DIAGNOSIS AND PROGNOSIS:
CONTRIBUTION TO THE ENERGY CHALLENGE

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October 7-9, 1980

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NBS-114A (REV. 2-80)

| U.S. DEPT. OF COMM. **BIBLIOGRAPHIC DATA SHEET** (See instructions) | 1. PUBLICATION OR REPORT NO. NBS SP 622 | 2. Performing Organ. Report No. | 3. Publication Date October 1981 |
|---|---|---|---|

**4. TITLE AND SUBTITLE**

Detection, Diagnosis and Prognosis: Contribution to the Energy Challenge
Proceedings of the 32nd Meeting of the Mechanical Failures Prevention Group, Held at the Inn at Santa Monica, Santa Monica, CA, October 7-9, 1980

**5. AUTHOR(S)**

T. Robert Shives and William A. Willard, Editors

**6. PERFORMING ORGANIZATION** (If joint or other than NBS, see instructions)

**NATIONAL BUREAU OF STANDARDS**
**DEPARTMENT OF COMMERCE**
**WASHINGTON, D.C. 20234**

**7. Contract/Grant No.**

**8. Type of Report & Period Covered**

Final

**9. SPONSORING ORGANIZATION NAME AND COMPLETE ADDRESS** (Street, City, State, ZIP)

NBS; Office of Naval Research, Dept. of the Navy, Arlington, VA 22217; Naval Air Systems Command, Washington, DC 20360; National Aeronautics & Space Admin., Goddard Space Flight Center, Greenbelt, MD 20771

**10. SUPPLEMENTARY NOTES**

Library of Congress Catalog Card Number: 81-600125

☐ Document describes a computer program; SF-185, FIPS Software Summary, is attached.

**11. 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 21 submitted entries (19 papers and 2 abstracts) from the 32nd meeting of the Mechanical Failures Prevention Group which was held at the Inn at Santa Monica, Santa Monica, CA, October 7-9, 1980. The subject of the symposium was the contribution of failure detection, diagnosis and prognosis to the energy challenge. Areas of special emphasis included energy management, techniques for failure detection in energy related systems, improved prognostic techniques for energy related systems, and opportunities for detection, diagnosis and prognosis in the energy field.

**12. KEY WORDS** (Six to twelve entries; alphabetical order; capitalize only proper names; and separate key words by semicolons)

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## NONPERIODICALS

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NOTE: The principal publication outlet for the foregoing data is the Journal of Physical and Chemical Reference Data (JPCRD) published quarterly for NBS by the American Chemical Society (ACS) and the American Institute of Physics (AIP). Subscriptions, reprints, and supplements available from ACS, 1155 Sixteenth St., NW, Washington, DC 20056.

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