AEROTHERMAL-MECHANICAL HEALTH MONITORING AND DIAGNOSTICS OF TURBO-COMPRESSOR SETS

by

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INTRODUCTION

High speed turbomachinery plays a critical role in today's petrochemical industry. There are very high penalty costs associated with nonavailability and catastrophic failure of critical unspared trains. Additionally the fuel and maintenance costs over the life cycle of plant turbomachinery is very significant. Both of the above factors point to the need for health monitoring and diagnostic systems. The petrochemical industry has in the past placed a heavy emphasis on mechanical (vibration) analysis for both health monitoring and diagnostics. This paper presents a methodology in which both mechanical and aerothermal parameters are utilized for machinery health monitoring, prognosis and diagnosis.

DIAGNOSTIC SYSTEMS

In order to have a total picture of machine health, both mechanical and aerothermal monitoring should be performed. By utilizing both vibration parameters and aerothermal parameters (flows, temperatures, pressures) one can more accurately pinpoint areas of machine distress. The combination of both these parameters calls for advanced software which can deal with the complex interrelationships between these various parameters. Also needed is a statistical treatment of factors related to prognosis and diagnosis.

Diagnostic systems must monitor the above mentioned mechanical and aerothermal parameters, analyze the problem and then schedule proper preventive maintenance. For a diagnostic system to be effective, a complex set of parameters must be utilized and in order to keep track of all these parameters and their interaction with each other, a computer based system becomes a must.

Health monitoring and diagnostic systems should accomplish the following:

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 shut-down 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. Great emphasis must be placed on man-machine interface to ensure user acceptance.

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, i.e. 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. The system should be flexible enough to incorporate improvements in the state of the art of instrumentation and diagnostics.

G. The system should have expansion capabilities to accept projected increases in installed machinery or increases in number of channels.

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

A diagnostic system is composed of many components and may accomplish numerous functions. The following are some of the components and functions required.

- Instrumentation and Instrumentation Mountings
- Signal Conditioning and Amplifiers for Instrumentation
- Data Transmission System: Cables, Telephone Linkup, or Microwave
- Data Integrity Checking, Data Selection, Data Normalization and Storage
- Baseline Generation and Comparison
- Problem Detection
- Diagnostic Generation
- Prognosis Generation
- Onsite Display
- Systems for Curve Plotting, Documentation and Reporting

Figure 1 shows a schematic diagram of such a system.

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.

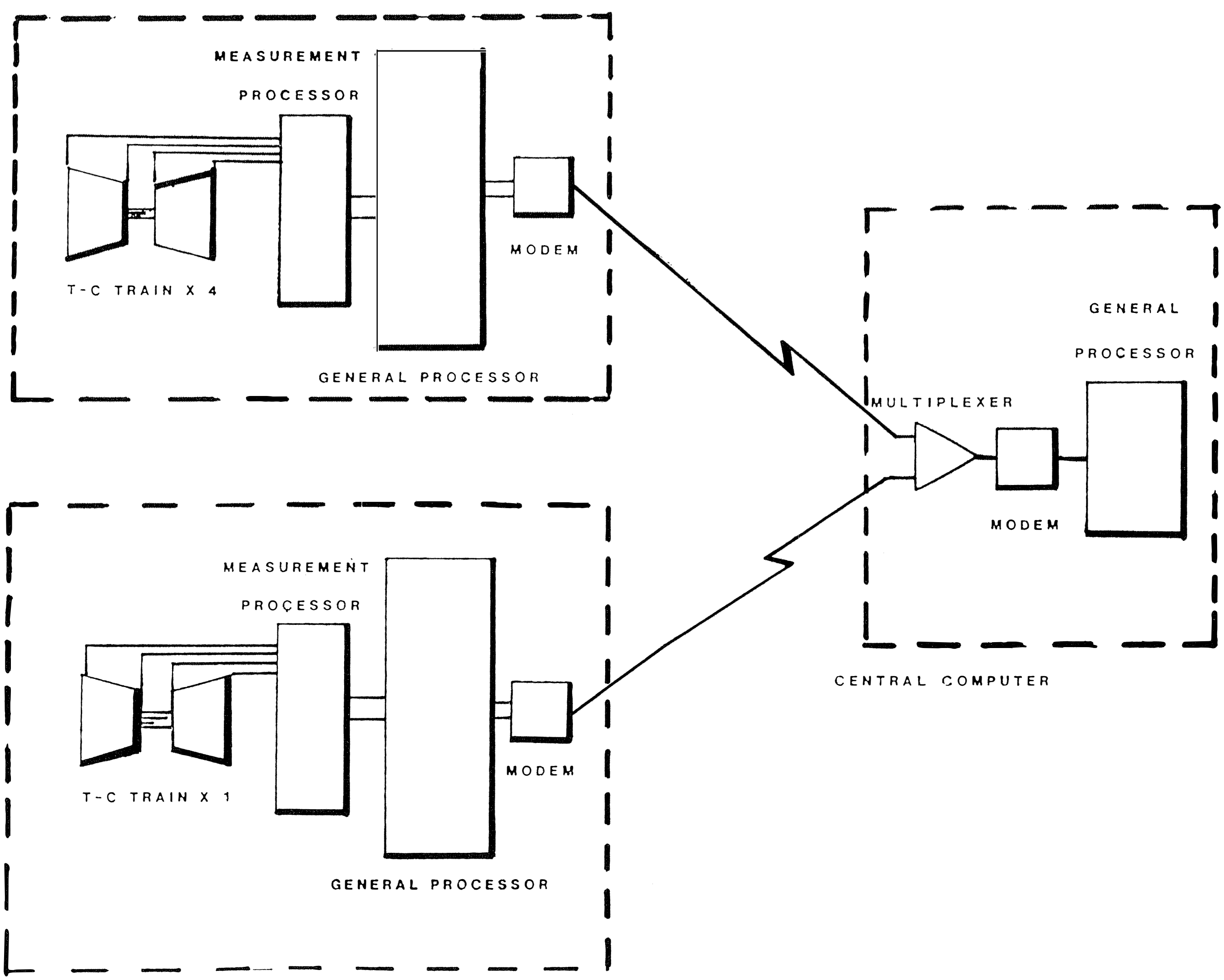


Figure 1. Block Diagram of Computer System.

This should include a study of the failure modes involved, and their effects. It is essential that the instrumentation requirements be tailored to the requirements of the machine being monitored.

Some 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 sensors 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. 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.

Any existing instrumentation should be used if found to be adequate. While there are disadvantages in the use of non-contacting sensors built into the machine for measurement of journal displacements, this instrumentation is often 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. To provide for the aerothermal data base the pressure, temperatures and flows should be recorded. A list of the minimum as well as very desirable optional instrumentation is given.

A. Typical Instrumentation — Minimum Requirements for Each Machine

Note: Locations and type of sensors would depend on the type of machine under consideration. This listing should be used only as a guideline.

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. Two thermocouples or resistance temperature element embedded in forward and aft thrust bearing

B. Instrumentation — Desirable — Optional

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 if 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 2 and 3 show possible instrumentation location for an industrial gas turbine and centrifugal compressor.

CRITERIA FOR THE COLLECTION OF AEROTHERMAL DATA

Turbomachinery operating pressures, temperatures, flows 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 evaluated. The accuracy of pressure and temperature measurements required will depend on the analysis and diagnostics that need to be performed. Table I 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. Thermocouple selection also depends upon the temperature to be measured. Figure 4 shows various types of thermocouples and their temperature ranges. Flow measurement can be done by orifice plates, venturimeters, annubars, nozzles or even around elbow bends. Annubars are used very often as they do not clog easily and do not give rise to high pressure drops. In several cases, the measurement of differential pressures and temperatures yield greater accuracy than absolute measurements. Often times for prognosis, an emphasis is more on relative changes in aerothermal parameters rather than absolute values.

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 5 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. While the displacement sensor is capable of measuring vibration frequencies well above 2Khz, the amplitude of vibrational displacement levels that

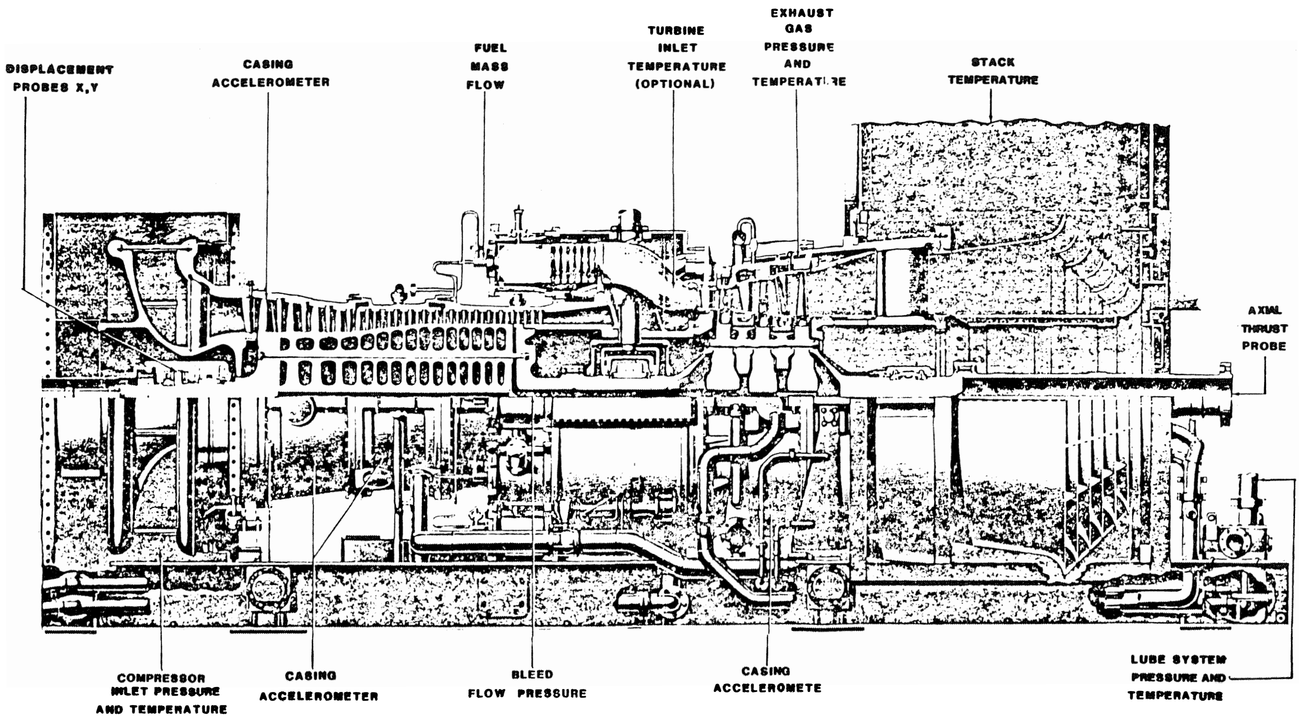


Figure 2. Typical Sensor Locations for Gas Turbine.

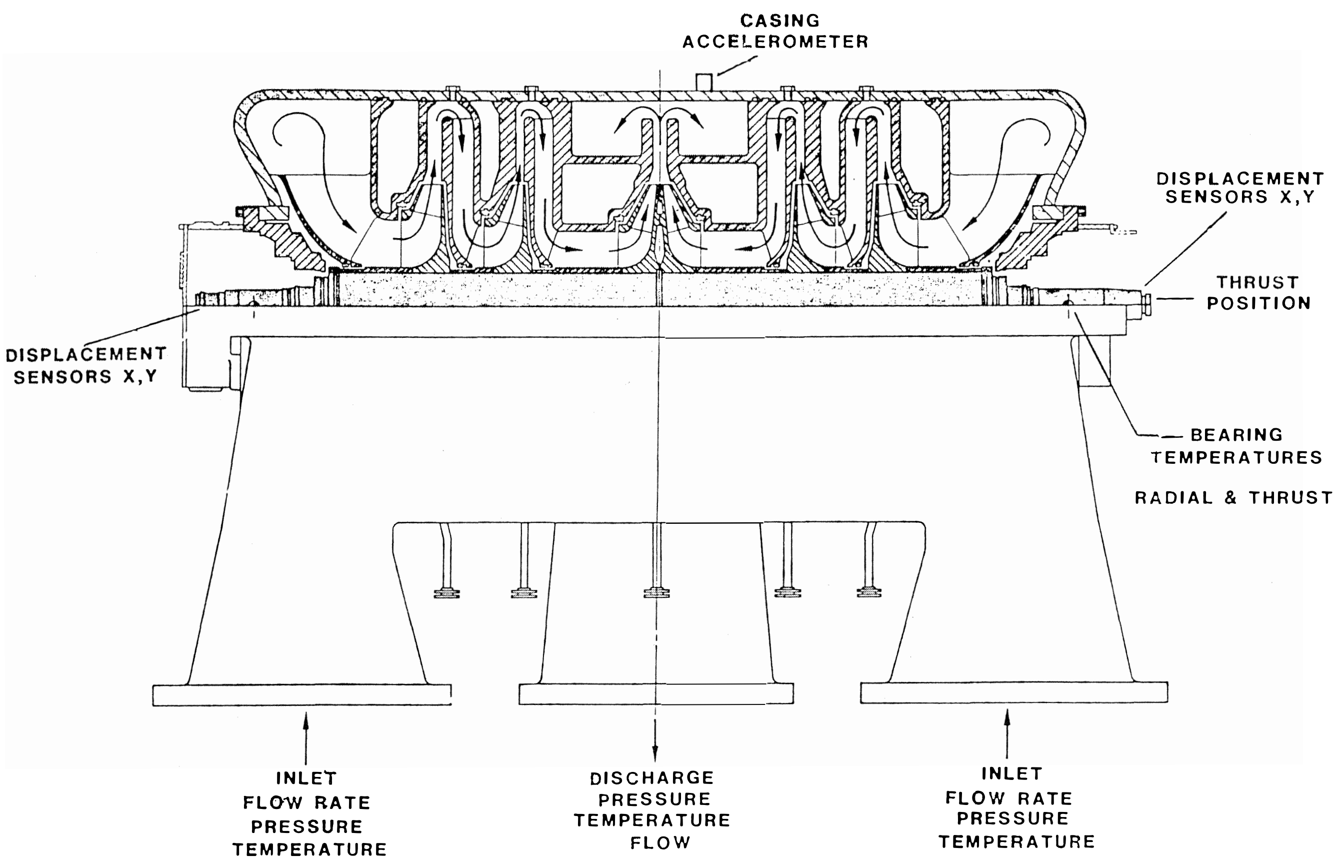


Figure 3. Typical Sensor Location Centrifugal Compressor.

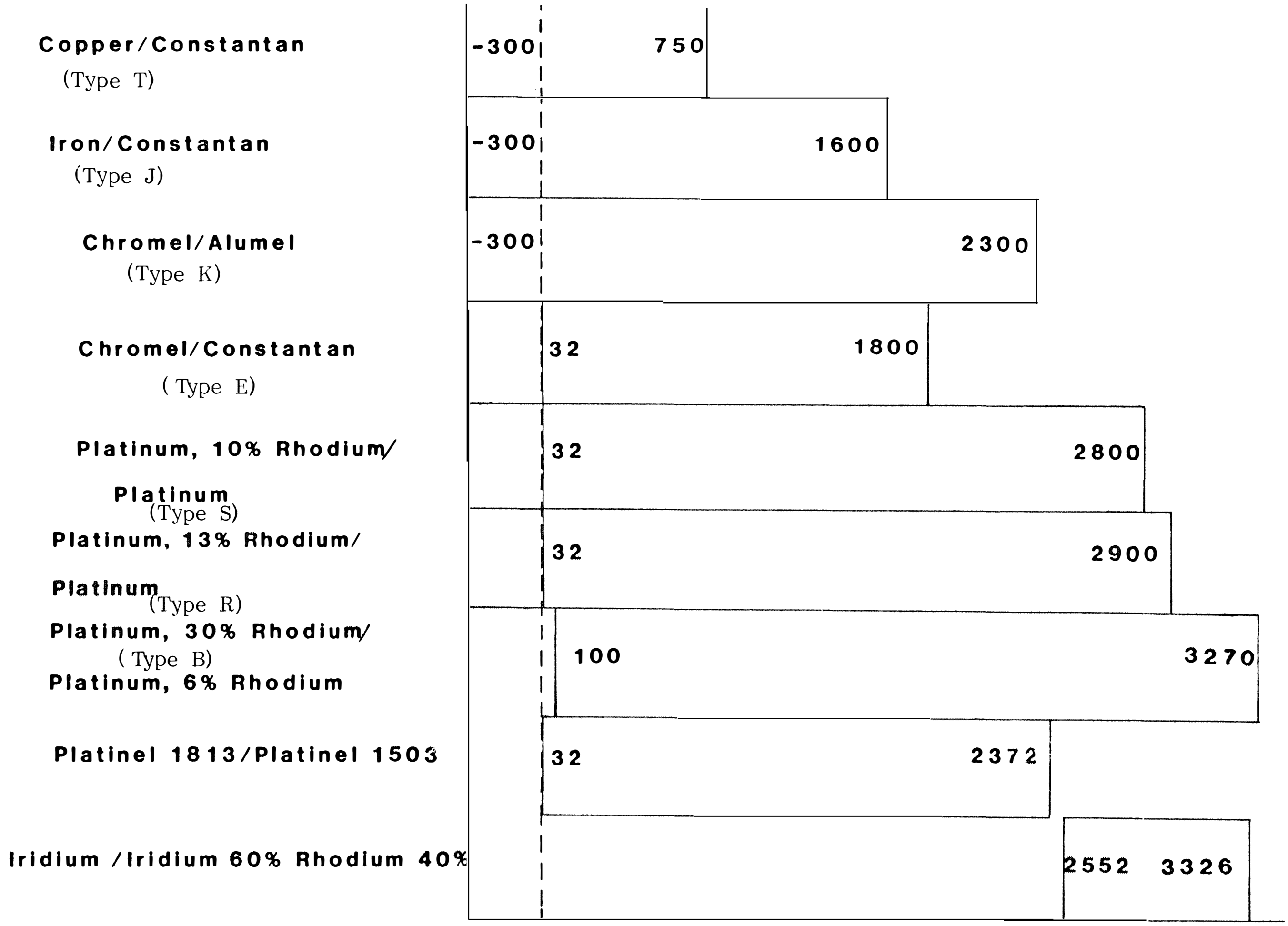


Figure 4. Ranges of Various Thermocouples.

TABLE I. 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.

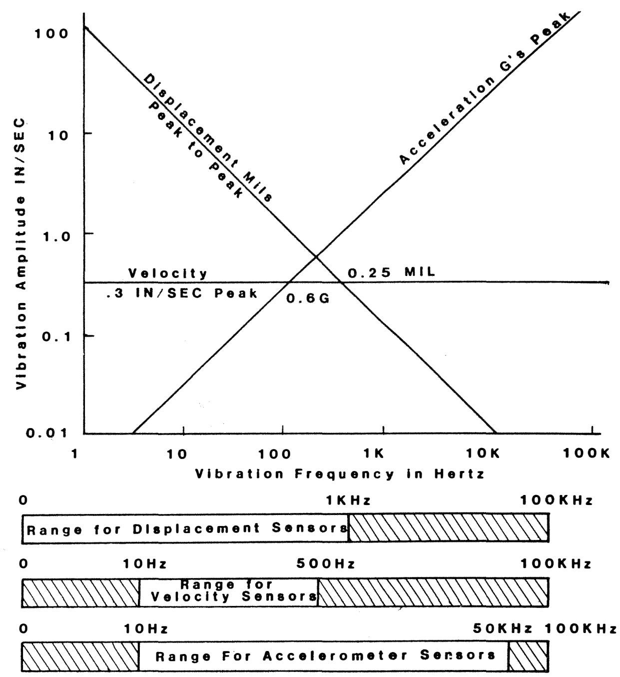


Figure 5. Vibration Transducer Selection Guide.

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, 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.

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 as shown in Figure 6. 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

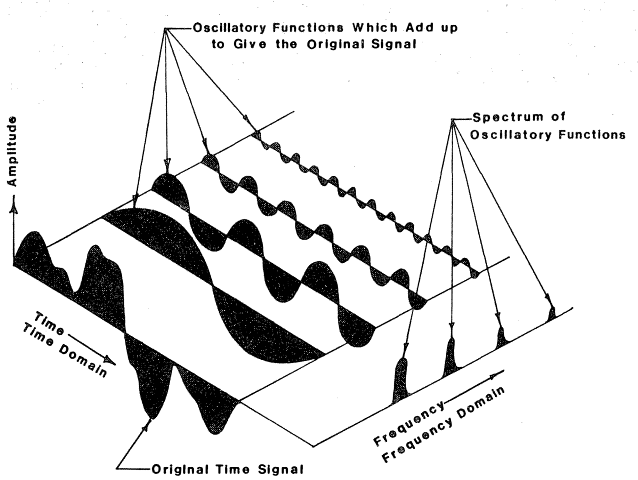


Figure 6. Decomposition of a Time Signal into a Sum of Oscillatory from Which a Spectrum Can be Obtained.

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 microprocessors and FFT dedicated circuitry.

The FFT can be implemented in a computer using 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 usage 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:

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

BASELINE FOR MACHINERY

Mechanical Baseline

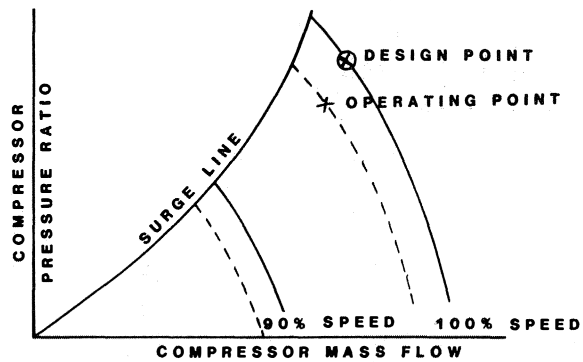
The vibration baseline for a machine can be defined as the normal or average operating condition of a machine. It can be represented on a vibration spectrum plot showing vibration frequency on the X-axis and vibration amplitude (peak-to-peak displacement, peak velocity, or peak acceleration) on the Y-axis. 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, the 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.

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 characteristics. 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. Typical compressor characteristics are presented in Figure 7 with data input, as well as diagnostic and monitoring output. Some of the monitoring and diagnostic outputs are decrease in compressor flow, decrease in pressure ratio and increases in operating fuel costs 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 degredation 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 Table II.



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 Throughput
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 7. Aerothermal Condition Monitoring for Compressors.

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 8 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 9 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

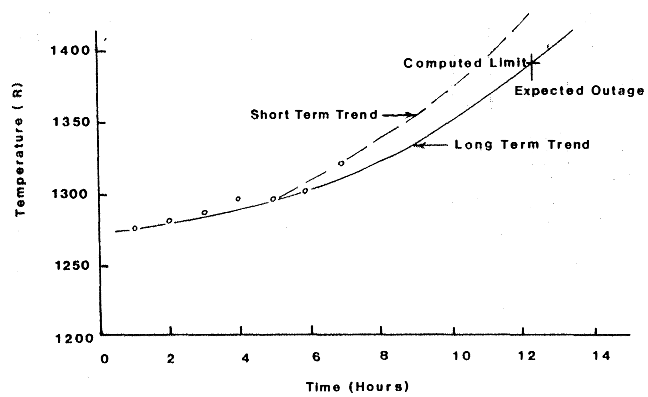


Figure 8. Temperature Versus Expected Outage Time.

TABLE II. GAS TURBINE AEROTHERMAL PERFORMANCE EQUATIONS FOR CORRECTION TO STANDARD DAY CONDITIONS.

FACTORS FOR CORRECTION TO STANDARD DAY TEMPERATURE AND PRESSURE CONDITIONS

Assumed Standard Day Pressure	14.7 psia
Assumed Standard Day Temperature	60° F(520°R)
Conditions of Test	
Inlet Temperature	T_1 °R
Inlet Pressure	P_1 psia

Corrected Temperature	= (Observed Temperature) $(520/T_1)$
Corrected Pressure	= (Observed Pressure)$(14.7/P_i)$
Corrected Speed	= (Observed Speed) $\sqrt{520/T_1}$
Corrected Air Flow	= (Observed Flow) $(14.7/P_i)$ $\sqrt{T_i/520}$
Corrected Horsepower	= (Observed Power) $(14.7/P_i)$ $\sqrt{520/T_i}$

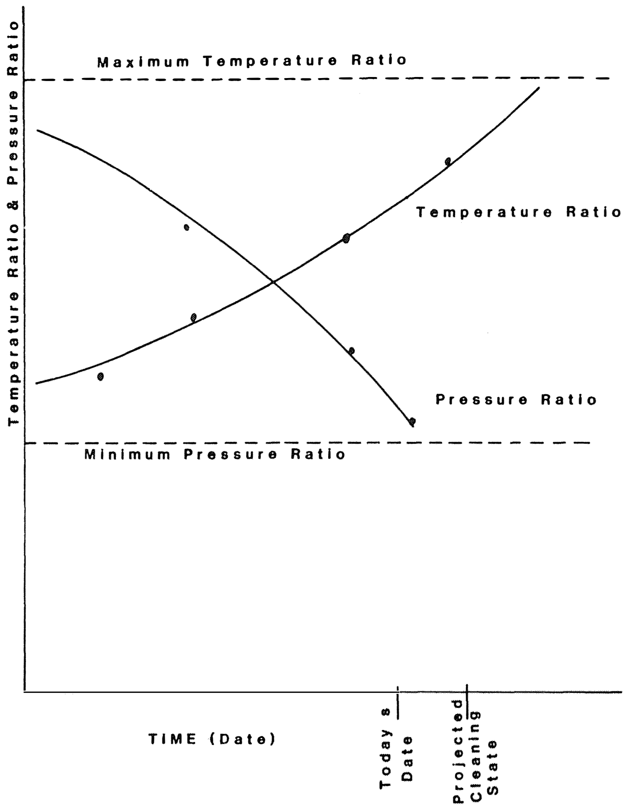


Figure 9. Data Trending to Predict Maintenance Schedules.

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.

COMPRESSOR AEROTHERMAL CHARACTERISTICS AND COMPRESSOR SURGE

Figure 10 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.

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. Surge is discussed in some more detail ahead.

DIAGNOSTICS

Problem evaluation in the turbomachinery is complex, but with the aid of performance and mechanical signals, solutions

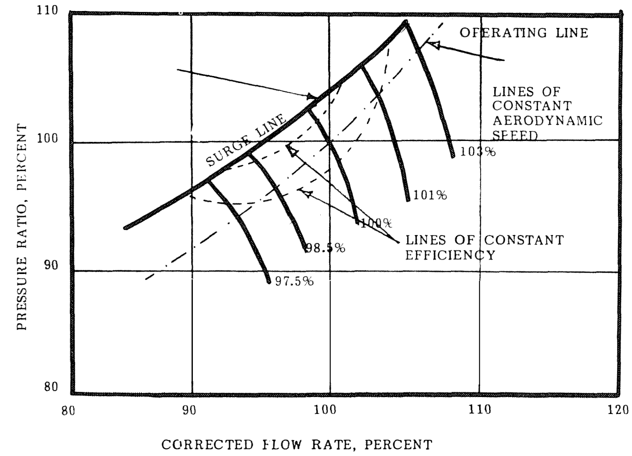


Figure 10. Typical Compressor Map.

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. Diagnostic systems must address these problems.

COMPRESSOR PROBLEMS AND DIAGNOSTICS

Centrifugal compressors, as used in pumping service, are units which in most cases operate above their first shaft critical. These types of shafts are known as flexible shafts and are prone to instability problems. It is thus very important that in these cases the unit must be designed to minimize the possibility of excited instabilities. Self excited instabilities such as oil whirl, hysteretic whirl, aerodynamic cross coupling and coulomb friction can result in the rotor being completely destroyed.

In the case of reinjection compressors, it is not uncommon for liquid slugging to take place. Liquid slugging can result in considerable damage to the thrust bearings by creating an axial force which results in an axial movement. In some cases liquid slugging can result in high forces in the impeller which can result in impellers which the shroud can be separated from the blades.

Oil whirl problems in the bearings exist usually above the first critical and can create major destruction to the rotor and the bearings. The addition of pressure dams or the changing to tilting pad bearings reduces this tendency. It is not uncommon that oil whirl can also be initiated by increasing back pressure. In one case an increase in the back pressure of about 50 psig in a total pressure rise of 700 psig caused instabilities which led to a total destruction of the rotor. Hysteretic whirl is found in rotors with large shrink fits and operating above their first critical. This was a common cause of failure in early rotor design. Aerodynamic cross coupling is a common problem in axial flow compressors due to the unsymmetry of the radial clearance between the blades and the shroud and this also does exist in centrifugal compressors. In centrifugal compressors, the problem is not as severe as that encountered in axial flow compressors. However, unshrouded centrifugal compressors have a much larger problem due to cross-coupling than do shrouded impellers. Coulomb friction occurs when a rub takes place between the journal and the bearing. These instabilities all occur usually in rotors operating above their first critical. The problem of surge is always existant in all compressors.

Surge is the reversal of flow in compressors. This flow reversal is also cyclic in nature and can lead to complete destruction of the rotor. There are many reasons which can place a compressor, operating away from the surge line, into surge. The compressor can be surged by a change in mol. wt. of the gas; increasing the mol. wt. causes the surge line to move to the right as shown in Figure 11. Other factors which can cause a compressor to surge sooner than predicted, is fouling of the blades by the build-up of dirt, change in speed, changes in inlet or exit conditions.

Centrifugal compressors also suffer from a phenomena called a rotating stall. This occurs usually in the inducer section of the impeller. The flow enters the compressor at a wrong angle thus causing the blockage of the flow between two adjacent blades as shown in Figure 12. The flow finding resistance goes to the adjacent blade passage, thus unclogging the first blade passage, but soon creating the same problem in the adjacent blade passage. Thus the rotating stall moves in the direction opposite to rotation and at a velocity equal to about forty or sixty percent of the rotating speed.

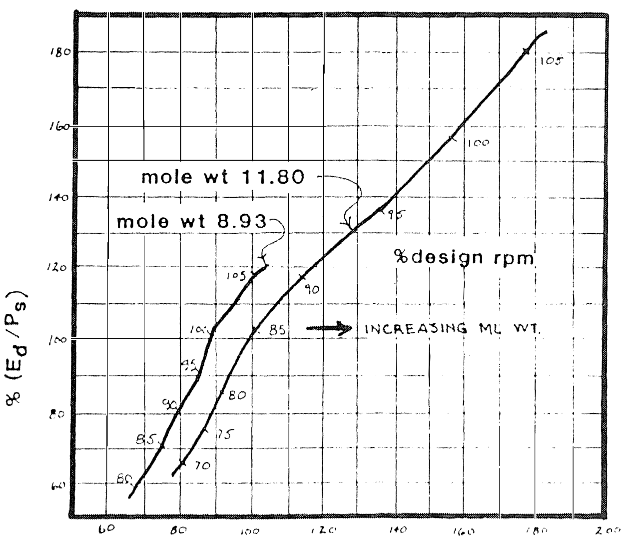


Figure 11. Effect of MW on Surge Line.

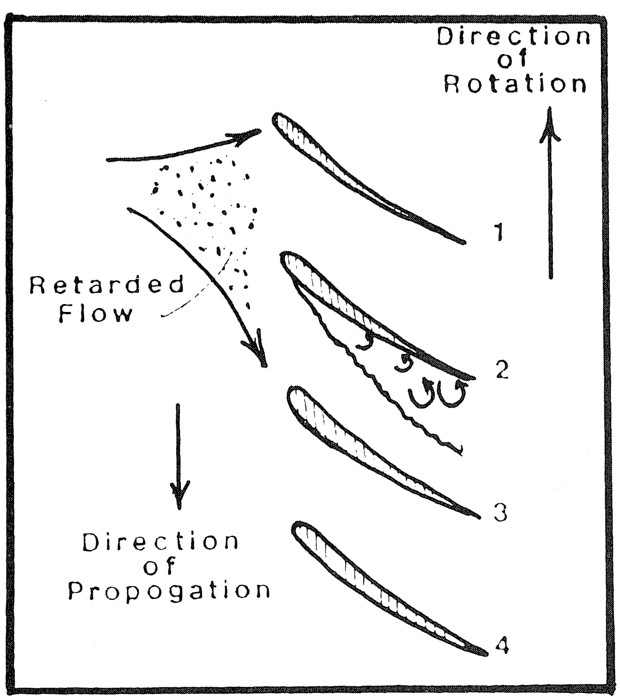


Figure 12. Propagating Stall in a Cascade.

Centrifugal compressors often have problems of high stresses on the shrouds and disks at their outer radius. Field repairs of this problem have led to the removal of the offending part by scalloping of the discs as shown in Figure 13.

Compressor diagnosis 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 III 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, 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 diference across the bearing, and an in-

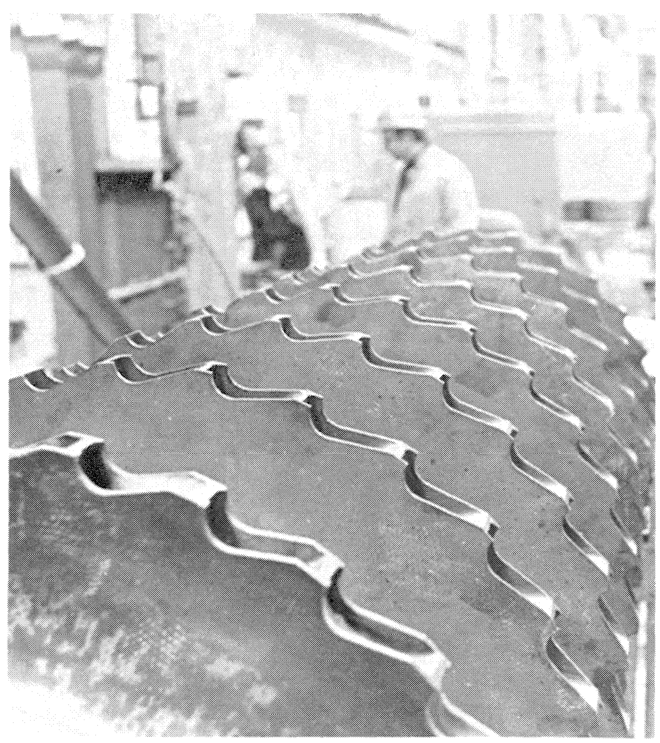


Figure 13. Scalloped Centrifugal Compressor to Relieve Stresses at Outer Disc and Shroud Diameter.

TABLE III. COMPRESSOR DIAGNOSTICS.

	η_c	P_2/P_1	T_2/T_1	Compressor Fluid Mass Flow	Vibration	Δ T Bearing	Bearing Pressure	Bleed Chamber Pressure
Clogged Filter								
Surge		Variable			Highly Fluctuating			Highly Fluctuating
Fouling								
Damaged Blade								Highly Fluctuating
Bearing Failure								

crease in vibration. If oil whirl or other bearing instabilities are present, there will be a vibration at subsynchronous frequency.

PROBLEMS & DIAGNOSTICS OF GAS TURBINES

There are several special considerations that have to be taken into account for gas turbines. Some of these are presented ahead.

Gas turbines are very sensitive to the type of fuel, the turbine inlet temperature and the number of starts. This is shown in Table IV where it is obvious that a change from natural gas to diesel fuels would reduce the life by about 30%.

FUELS

Fuel candidates encompass the entire spectrum from gases to solids. Gaseous fuels traditionally include natural gas, process gas, low Btu coal gas and vaporized oil gas. Process gas is a broad term used to refer to gas formed by some industrial process. Process gases include refinery gas, producer gas, coke oven gas, and blast furnace gas as well as others. Natural gas is usually the basis on which performance for a gas turbine is compared, since it is a clean fuel giving rise to a long life. Vaporized fuel oil gas in most cases behaves very closely to natural gas in that it provides a high performance with minimum reduction of component life. About 40% of the turbine power installed operates on liquid fuels. Liquid fuels can vary from light volatile naptha through kerosene to the heavy viscous residuals. The classes of liquid fuels and their requirements are shown in Table V. The light distillates are equal to natural gas as a fuel, and between light distillates and natural gas as fuels, 90% of the installed units can be accounted for. Care must be taken in handling liquid fuels to avoid contamination, and the very light distillates like naphtha require special concern in the design of fuel systems because of high volatility. Generally, a fuel tank of the floating head type which has no area for vaporization is employed. The heavy true distillates like #2 distillate oil can be considered the standard fuel. The true distillate fuel is a good turbine fuel; however, due to the fact that trace elements of vanadium, sodium, potassium, lead, and calcium are found in the fuel, the fuel has to be treated. The corrosive effect of sodium and vanadium is very detrimental to the life of the turbine. Vanadium originates as a metallic compound in crude oil and is concentrated by the distillation process into heavy oil fractions. Sodium compounds are most often present in the form of salt water which results from salty wells, transport over sea water, or mist ingestion in an ocean environment. Fuel treatments are costly and do not remove all traces of these metals. As long as the fuel oil properties fall within specific limits no special treatment is required. Blends are residuals which have been mixed with lighter distillates to improve properties. The specific gravity and viscosity can be reduced by blending. About 1% of the total installed machines can operate on blends.

A final group of fuels are high ash crudes and residuals. These account for 5% of the installed units. Residual fuel is the high ash by-product for distillation. Low cost makes them attractive; however, special equipment must always be added to the fuel systems before they can be utilized. Crude is attractive as a fuel since in pumping applications it is burned straight from the pipeline. Table VI gives the specification of various fuels as far as the gas turbine is concerned. Table VII indicates the properties of various liquid fuels which may be used in gas turbines.

To fully understand the table a definition of some of the properties are given. The flash point is the temperature at which vapors may begin combustion. The flash point is an indication of the maximum temperature at which a fuel can be

TABLE IV. OPERATION AND MAINTENANCE LIFE OF AN INDUSTRIAL TURBINE.

		Type Inspection — Hrs. of Operation			Expected Life (Replacement) — Hrs. of Operation		
Type & Number Load Fuel & Starts		Service	Minor	Major	Comb. Liners	1st. Stage Nozzles	1st. Stage Buckets
BASE	*	+	+	+	+	+	+
Nat. Gas	1/1000	4500	9000	28000	30000	60000	100000
Nat. Gas	1/10	2500	4000	13000	7500	42000	72000
Distillate Oil	1/1000	3500	7000	22000	22000	45000	72000
Distillate Oil	1/10	1500	3000	10000	6000	35000	48000
Residual	1/1000	2000	4000	5000	3500	20000	28000
Residual	1/10	650	1650	2300			
SYSTEM PEAKING x							
Nat. Gas	1/10	3000	5000	13000	7500	34000	
Nat. Gas.	1/5	1000	3000	10000	3800	28000	
Distillate	1/10	800	2000	8000			
Distillate	1/5	400	1000	7000			
TURBINE PEAKING x							
Nat. Gas	1/5	800	4000	12000	2000	12000	
Nat. Gas	1/1	200	1000	3000	400	9000	
Distillate	1/5	300	2000	6000			
Distillate	1/1	100	800	2000			

* 1/5 = One Start per five operating hours

x No Residual usage due to low load factor and high capital cost

BASE = Normal maximum continuous load

SYSTEM PEAKING = Extra load resulting from operating temperature 50° to 100° F above base temperature for short durations

SERVICE = Inspection combustion parts, required downtime approximately 24 hours

MINOR = Inspection of combustion plus turbine parts, required downtime approximately 80 hours

MAJOR = Complete inspection and overhaul, required downtime approximately 160 hours

NOTE: Maintenance times are arbitrary and depend on manpower availability and training, spare parts and equipment availability, and planning. Boroscope techniques can help reduce downtime.

handled safely. The pour point is an indication of the lowest temperature at which a fuel oil can be stored and still be capable of flowing under gravitational forces. Fuels with higher pour points are permissible where the piping has been heated. Water and sediment in the fuel lead to fouling of the fuel system and obstruction in fuel filters. The carbon residue is a measure of the carbon compounds left in a fuel after the volatile components have been vaporized. Two different carbon residue tests are used, one for light distillates and one for heavier fuel. For the light fuels, 90% of the fuel is vaporized and the carbon residue is found in the remaining 10%. For heavier fuels, since the carbon residue is large, 100% of the sample can be used. This gives a rough approximation of the tendency to form carbon deposits in the combustion system. Ash is the material remaining after combustion has taken place. Ash can be present in two forms:

1. as solid particles corresponding to that material designated as sediment, and
2. oil and water soluble traces of metallic compounds.

The metallic compounds present in the ash are related to the corrosion properties of the fuel. Viscosity is a measure of the resistance to flow and is important in the design of fuel pumping systems. Specific gravity is the weight of the fuel in relation to water. This property is important in the design of centrifugal fuel washing systems. Sulfur content is important in connection with emission concerns and in connection with the alkaline metals present in the ash. Sulfur, reacting with alkaline metals, forms compounds that corrode by a process labeled sulfidation. The luminosity measures the amount of chemical energy in the fuel that is released as thermal radiation. Finally, the weight of a fuel, light or heavy, refers to volatility. The most volatile fuels will vaporize easiest and come out early in the distillation process. That which remains after distillation is referred to as residual. The ash content of residual fuels is high.

Corrosion is generally described as hot corrosion and sulfidation processes. Hot corrosion is generally defined as accelerated oxidation of alloys caused by the deposition of Na_2SO_4, resulting from the ingestion of salts in the engine, and sulfur from the combustion of fuel. Sulfidation corrosion is considered a form of hot corrosion in which the alkali ash (combustion product) combines with sulfur to form a reactive residue which contains alkaline sulfates. Corrosion causes deterioration of blade materials and reduces the life of the components.

TABLE V. COMPARISON OF LIQUID FUELS FOR GAS TURBINES.

GENERAL FUEL TYPE	TRUE DISTILLATE & NAPHTHAS	BLENDED HEAVY DISTILLATES & LOW ASH CRUDES	RESIDUALS & HIGH ASH CRUDE
Fuel Pre-Heat	NO	YES	YES
Fuel Atomization	Mech/LP air	HP/LP air	HP air
Desalting	NO	Some	YES
Fuel Inhibitation	Usually None	Limited	Always
Turbine Washing	NO	Yes Except Distillate	YES
Startup Fuel	With Naphtha	Some Fuels	Always
Base Fuel Cost	Highest	Intermediate	Lowest
Description	High Quality Distillate Essentially Ash Free	Low Ash, Limited Contaminant Levels	Low Volatility High Ash
Types of Fuels Included	True Distillates (Naphtha, Kerosene, No. 2 Diesel, No. 2 Fuel Oil, JP-4,JP-5)	High Quality Crudes, Slightly Contaminated Distillates Navy Distillate	Residuals and Low Grade Crude (No. 5 Fuel, No. 6 Fuel, Bunker C)
ASTM Designation	1-GT, 2-GT, 3-GT	3-GT	4-GT
Turbine Inlet Temperature	Highest	Intermediate	Lowest

TABLE VI.

GASEOUS FUEL SPECIFICATIONS

Heating Value	300-500 Btu/ft^3
Solid Contaminants	30ppm
Flamability Limits	2.2:1
Composition S,Na,K,Li	5ppm
(Sulfur + Sodium + Potassium + Lithium)	(When formed into alkaline metal sulfate)
H_2O (by Weight)	.25%

LIQUID FUEL SPECIFICATIONS

Water and Sediment	1.0% (V%) Max.
Viscosity	20 centistokes at fuel nozzle
Pour Point	About 20°F below min. ambient
Carbon Residue	1.0% (wt) base on 100% of sample
Hydrogen	11.% (wt) minimum
Sulfur	1% (wt) maximum

TYPICAL ASH ANALYSIS AND SPECIFICATIONS

Metal	Lead	Calcium	Sodium & Potassium	Vanadium
Spec. MAX (ppm)	1	10	1	0.5 untreated 500 treated
Naptha	0-1	0-1	0-1	0-.1
Kerosene	0-1	0-1	0-1	0-.1
Light distill.	0-1	0-1	0-1	0-.1
Heavy distill.(true)	0-1	0-1	0-1	0-.1
Heavy distill.(blend)	0-1	0-5	0-20	.1/80
Residual	0-1	0-20	0-100	5/400
Crude	0-1	0-20	0-122	.1/80

TABLE VII. FUEL PROPERTIES.

Property	Kerosene	Diesel Fuel #2	Burner Fuel Oil #2	JP-4	High Ash Crude Heavy Residual	Typical Libyan Crude	Navy Distillate	Heavy Distillate	Low Ash Crude
Flash Point °F	130/160	118-220	150/200	RT	175/265		186°F	198	50/200
Pour Point °F	−50	−55 to + 10	−10/30		15/95	68	10°F		15/110
Misc. CS @ 100°F	1.4/2.2	2.48/2.67	2.0/4.0	.79	100/1800	7.3	6.11	6.20	2/100
SSU		34.4					45.9		
Sulfur %	.01/.1	.169/.243	.1/.8	.047	.5/4	.15	1.01	1.075	1/2.7
API gr.		38.1	35.0	53.2			30.5		
Sp. Gr. @ 100°F	.78/.83	.85	.82-.88	.7543@60°F	.92/1.05	.84	.874	.8786	.80/.92
Water & Ded			0			.1%wt			
Heating Valve $\frac{BTU}{lb}$	19330/19700	18330	19000/19600	18700/18820	18300/18900	18250		18239	19000/19400
Hydrogen %	12.8/14.5	12.83	12.1/13.2	14.75	10/12.5			12.40	12/13.2
Carbon Residue									
10% Bottoms	.01/.1	.104	.03/.3			2/10			.3/3
Ash PPM	1/5	.001	0/20		100/1000	36ppm			20/200
Wa + K PPM	01.5		0/1		1/350	2.2/4.5			0/50
V	0/.1		0/.1		5/400	0/1			0/15
Pb	0/.5		0/1		0/25				
Ca	0/1	0/2	0/2		0/50				

The hot corrosion mechanism is not fully understood and there is considerable controversy on the subject. However, hot corrosion can be discussed in general. Hot corrosion includes two mechanisms.

1. *Accelerated Oxidation*
   during initial stages — blade surface clean
   $Na_2SO_4 + Ni$ (metal) NiO (porous)
2. *Catastrophic Oxidation*
   occurs with Mo, W, and V present and reduces NiO Layer — increases oxidation rate

Reactions — Ni-base alloys

Protective oxide films

$2Ni + O_2 \rightarrow 2NiO$

$4\ Cr + 3O_2 \rightarrow 2Cr_2O_3$

Sulfate

$2\ Na + S + 2O_2 \rightarrow Na_2SO_4$

Na — from NaCl (salt)

S — from fuel

Other Oxides

$2Mo + 3O_2 \rightarrow 2MoO_3$

$2W + 3O_2 \rightarrow 2WO_3$

$4V + 5O_2 \rightarrow 2V_2O_5$

The Ni-base alloy surface is exposed to an oxidizing gas, oxide nuclei form, and a continuous oxide film forms (NiO, Cr_2O_3, etc.). This oxide film is a protective layer. The metal ions diffuse to the surface of the oxide layer and combine with the molten Na_2SO_4 which destroys the protective layer. Ni_2S and Cr_2S_3 result in sulfidation.

$NaCl \rightarrow$ (sea salt) $Na + Cl$

$2Na + S$ (fuel) $+ 2O_2 \rightarrow Na_2SO_4$

Cl — grain boundaries — cause intergranular corrosion.

The extent of the corrosion depends on the amount of nickel and chromium in the alloy. The oxide films become porous and nonprotective which increases the oxidation rate (accelerated oxidation).

Catastrophic oxidation requires the presence of Na_2SO_4 and Mo, W, and/or V. Crude oils are high in V; ash would be 65% V_2O_5 or higher. The rate at which corrosion proceeds is related to temperature. At temperature over 1500°F, attack by sulfidation takes place rapidly. At lower temperatures with vanadium rich fuels, oxidation catalyzed by vanadium pentoxide can exceed sulfidation. The effect of temperature on IN 718 corrosion by sodium and vanadium is shown in Figure 14. The corrosive threshold is generally accepted to be in the range of

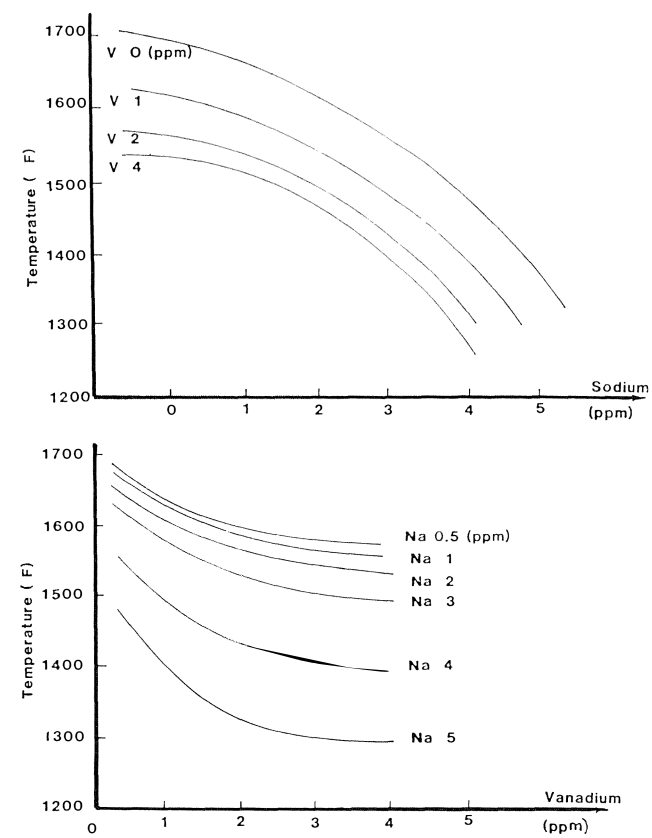


Figure 14. Temperature for a Maximum Corrosion Rate of 2.5 mm/20,000 hr. Burning Fuels with Variable Na and V-Concentrations (Alloy IN 738 IC).

1100-1200°F, and this cannot be considered a feasible firing temperature due to low efficiency and power output. Figure 15 shows the effect of sodium plus potassium and vanadium on life. Allowable limits for 100, 50, 20, and 10% of normal life with uncontaminated fuel at standard firing temperatures are shown. Sodium, potassium and calcium compounds are most often present in fuel in the form of sea water. Methods developed to remove the salt and reduce the sodium, potassium and calcium rely on the water solubility of these compounds. Removal of these compounds through water solubility is known as fuel washing. Fuel washing systems fall into (3) categories — centrifugal, electrostatic and hybrid.

Vanadium originates as a metallic compound in crude oil and is concentrated by the distillation process into the heavy oil fractions. Oxidation of the blades occurs when liquid vanadium is deposited onto the blade to act as a catalyst. Vanadium compounds are oil soluble and are thus unaffected by fuel washing. Without additives, vanadium forms low melting temperature compounds which deposit on the blade in a molten slag state causing rapid corrosion. However, by the addition of a suitable compound, magnesium for example, the melting point of the vanadates is increased sufficiently to prevent them from being in the liquid state under service conditions. Thus, slag deposition on the blade is avoided. Calcium had been initially selected as the inhibiting agent as tests indicated it was more effective at 1750°F. Subsequent tests showed magnesium gave better protection at 1650°F and below. However, at temperatures of 1750°F and over, magnesium no longer inhibits but accelerates corrosion. Magnesium also gives more friable deposits than calcium inhibitors. A magnesium/vanadium ratio of 3:1 reduces corrosion by a factor of 6 between temperatures of 1550 and 1400 degrees F. Figure 16 is a photograph of a turbine blade which has been operating in a high vanadium fuel. Note the heavy buildup of ash on this blade. This turbine needs water washing every 150 hours.

Temperature Effects

The melting point of various metals varies considerably and their strength at various temperatures are different. At low temperatures all materials deform elastically and then plastically and are time independent. However at higher temperatures deformation is noted under constant load conditions. This high temperature time dependent behavior is called creep-

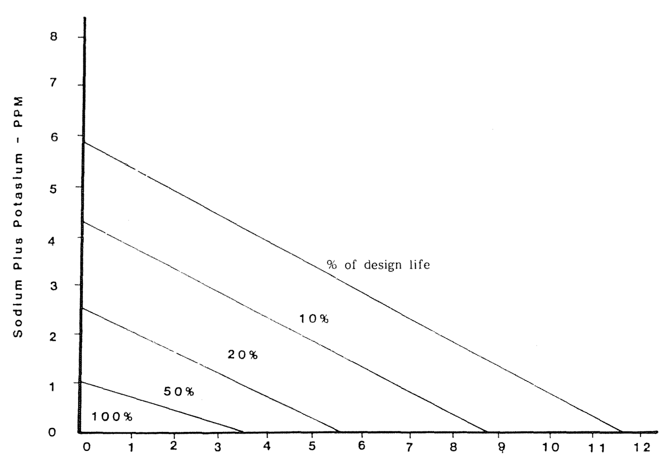


Figure 15. Corrosion Life Effects for Trace Metals in the Fuel.

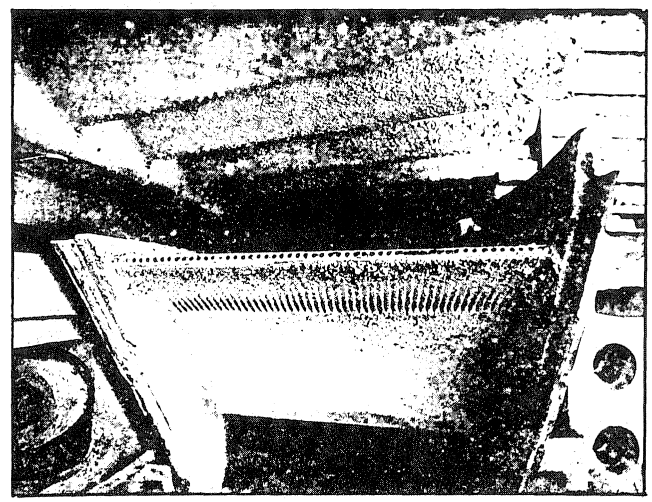


Figure 16. Turbine Blade Covered With Ash Operating With High Venadium Content Fuel.

rupture. Figure 17 shows the creep curve with the various stages of creep. The initial or elastic strain is the first region; we then proceed into a plastic strain region at a decreasing rate, and then, a normally constant plastic strain rate followed by an increasing strain rate to fracture.

The nature of this creep depends on material, stress, temperature and environment. Limited creep (less than 1%) is desired for turbine blade application. Cast superalloys fail with only a minimum elongation. These alloys fail in brittle fracture even at the elevated operating temperatures.

The stress-rupture data are often presented by using a Larson-Miller curve which presents the performance of an alloy in a complete and compact graphical style. While widely used to describe an alloy's stress-rupture characteristics over a wide temperature, life and stress range, it is also very useful in

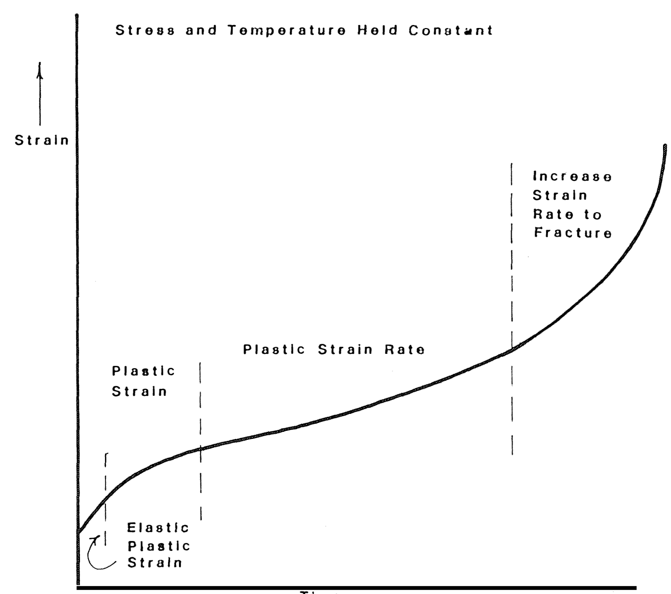


Figure 17. Time Dependent Strain Curve Under Constant Load.

comparing the elevated temperature capabilities of many alloys. The Larson-Miller parameter is:

$P_{LM} = T(20 + \log t) \times 10^{-3}$

P_{LM} = Larson-Miller parameter

T = temperature (°R)

t = rupture time (hour)

The Larson-Miller parameters are plotted in Figure 18 for the specified turbine blade alloys. A comparison of A-286 and Udimet 700 alloy curves reveals the difference in capabilities. The operational life (hours) of the alloys can be compared for similar stress and temperature conditions.

Ductility is commonly measured by elongation and reduction in area. In many cases all three stages of creep shown in Figure 17 are not always present. At high temperatures or stresses very little primary creep is seen. While in the case of cast superalloys, failure occurs with just a small extension. The amount of extension is ductility. In a time creep curve there are two elongations of interest, one is the elongation due to the plastic strain rate and the second is the total elongation which is the elongation at fracture. Ductility is erratic in its behavior and is not always repeatable even under laboratory conditions. Ductility of a metal is affected by the grain size, specimen shape and techniques used for manufacturing. The fracture which results from elongation can be of two types, brittle or ductile, depending on the alloy. Brittle fracture is intergranular with little or no elongation, ductile fracture is trangranular and typical of normal ductile tensile fracture. Turbine blade alloys tend to indicate low ductility at operating temperatures. As a result, surface notches are initiated by errosion or corrosion and then cracks are propagated rapidly.

Thermal fatigue of turbine blades is secondary failure mechanism. Temperature differentials developed during starting and stopping of the turbine produce thermal stress. The cycling of these thermal stresses is thermal fatigue. Thermal fatigue is low-cycle and is similar to a creep-rupture failure. The analysis of thermal fatigue is essentially a problem in heat transfer and properties such as modulus of elasticity, coefficient of thermal expansion, and thermal conductivity.

The most important metallurgical factors seem to be ductility and toughness. High ductility materials tend to be more resistant to thermal fatigue. They seem more resistant to crack initiation, and more propagation resistant.

Of all the gas turbine components, the first-stage blade must withstand the most severe combination of temperature, stress, and environment. Advances made in gas turbine bucket alloys from 1950 to the present are shown in Figure 19. It indicates that the temperature capability of these alloys has improved at a rate of approximately 15°F per year. While this increase does not appear very large at first glance, it corresponds to an increase in attainable output of between 1.5 and 2.0 percent and an improvement in efficiency of from 0.3 to 0.6 percent.

Inconel 738 is being presently used in the first stages and the U-500 or Nimonic blades are often used in the last stages of turbines. Table VIII shows the properties of the various alloys as used in turbines.

The superalloys currently used by gas turbine engine manufacturers for fabrication of blades and vanes for turbine hot sections are predominantly either nickel-or cobalt-based with compositions emphasizing high temperature mechanical properties. Their chemical formulations, therefore, preclude sufficient amount of those elements, primarily chromium and aluminum, which would make these alloys inherently resistant to the chemical reactions encountered during engine service. The use of coatings is becoming more widespread as higher temperatures are being achieved and better coatings are being produced.

For an example of comparative corrosion, coated and uncoated IN-738 blades were run side-by-side in the same machine, under severe corrosive conditions. The two blades were removed for interim evaluation from a MS-5002 Gas Turbine after 11,300 service hours (289 starts). This unit burns sour natural gas containing about 3½ percent sulfur, and is located in a region where the soil surrounding the site contains up to 3 percent sodium.

The uncoated blades showed about 0.006 in. corrosion attack over about 50 percent of the airfoil concave face, with about 0.010 in. penetration at the base of the airfoil. Examination of the coated blade revealed no visual evidence of attack except for one small roughened spot on the leading edge, about 2 in. up from the platform and a second spot in the middle of the convex side, about 1 in. down from the tip.

Metallographic examination of other areas revealed similar degrees of corrosion on the two blades. At no point on the coated blades had the coating penetrated to the base metal, although in the two areas discussed above, about 0.002 in. of the original 0.003 in. coating had been oxidized. Experience

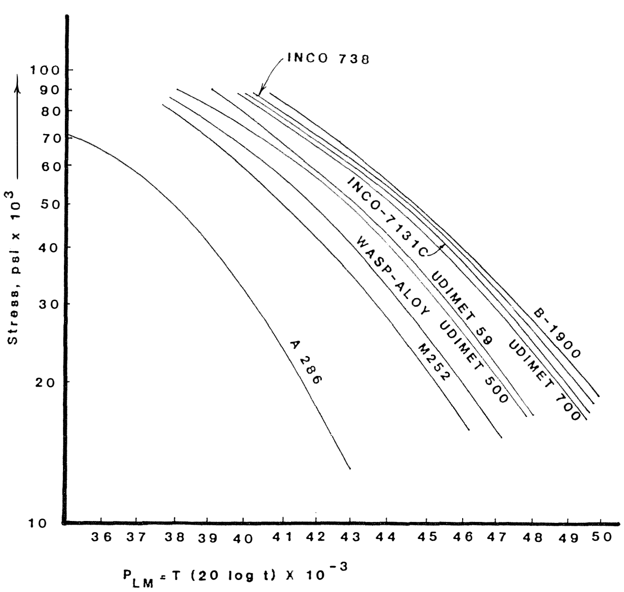


Figure 18. Larson-Miller Parameters for Turbine Blade Alloys.

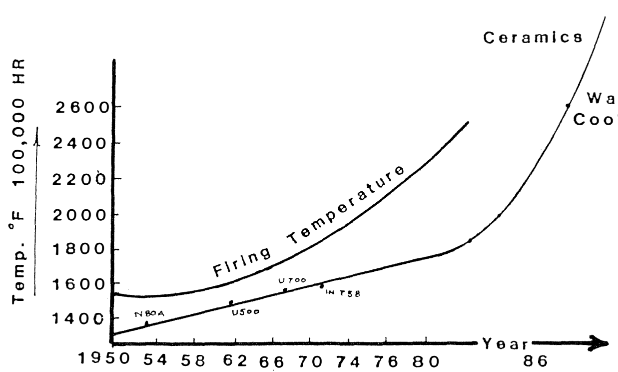


Figure 19. Trends Showing Improvement in Firing Temperature and Bucket Materials.

TABLE VIII. HIGH TEMPERATURE ALLOYS (Nominal Composition — %.

	Cr	Ni	Co	Fe	W	Mo	Ti	Al	Cb	V	C	B	Ta
BUCKETS													
S816	20	20	BAL	4	4	4	—	—	4	—	0.40	—	—
NIM 80A	20	BAL	—	4	—	—	2.30	1	—	—	0.05	—	—
M-252	19	BAL	10	2	—	10	2.50	0.75	—	—	0.10	—	—
U-500	18.5	BAL	18.5	—	—	4	3	3	—	—	0.07	0.006	—
RENE 77	15	BAL	17	—	—	5.3	3.35	4.25	—	—	0.07	0.02	—
IN 738	16	BAL	8.3	0.2	2.6	1.75	3.4	3.4	0.9	—	0.11	0.01	1.75
GTD 111	14	BAL	9.5	—	4.0	1.5	3.0	5.0	—	—	0.11	0.01	3.0
PARTITIONS-1st													
X-40	25	10	BAL	1	8	—	—	—	—	—	0.50	0.01	—
X-45	25	10	BAL	1	8	—	—	—	—	—	0.25	0.01	—
FSX	29	10	BAL	1	7	—	—	—	—	—	0.25	0.01	—
N-155	21	20	20	BAL	2.5	3	—	—	—	—	0.20	—	—
TURBINE WHEELS													
Cr-Mo-V	1	0.5	—	BAL	—	1.25	—	—	—	.25/5	0.30	—	—
A-286	15	25	—	BAL	—	1.2	2	0.3	—	0.25	0.08	0.006	—
M-152	12	2.5	—	BAL	—	1.7	—	—	—	0.3	0.12	—	—
IN 706	16	41	—	BAL	—	—	1.7	0.4	3.0	—	0.06	0.006	—

with uncoated IN-738 in this very hostile environment indicates about 25,000 hours blade life can be attained. The coated blade life, based on this interim evaluation, could add an additional 20,000 hours of life.

STARTS

The number of starts creates a major effect on the life of the blades. Temperature differentials developed during starting and stopping of the turbine, produce thermal stress and the cycling of these stresses causes thermal fatigue. Thus starts and stops reduce the life of a gas turbine. Large turbines also require some sort of mechanism to slow roll the shaft after the unit has been brought to rest. Figures 20 and 21 show the effect of the number of starts on the conbustor and the turbine section. The concept of equivalent hours as shown in these figures is very useful in planning maintenance intervals since it gives a more realistic view of the condition of the machinery than do the fired hours.

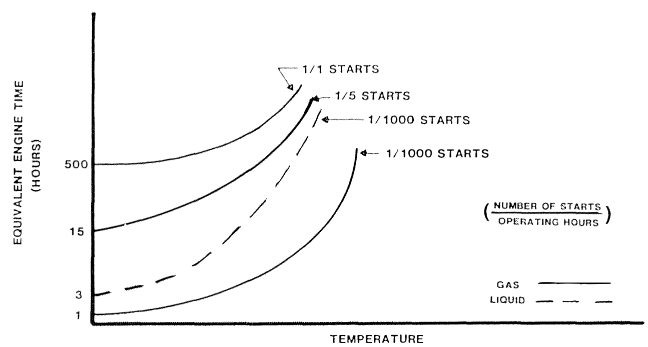


Figure 20. Equivalent Engine Time in the Combustor Section.

COMBUSTOR DIAGNOSTICS

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 measured, due to very high temperatures and limited probe life. Table IX 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 conjunction with increased combustion unevenness. 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.

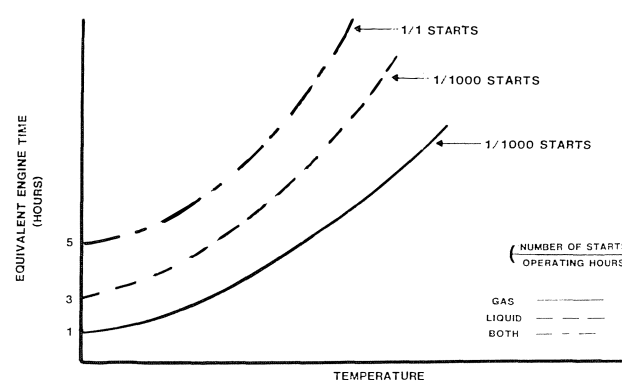


Figure 21. Equivalent Engine Time in the Turbine Section.

TABLE IX. COMBUSTOR DIAGNOSIS.

	FUEL PRESSURE	UNEVENNESS OF COMBUSTION (SOUND)	EXHAUST TEMPERATURE SPREAD	EXHAUST TEMPERATURE
CLOGGING	↑	↑	↑	↑
COMBUSTOR FOULING	↑ OR ↓	↑	↑	↓
CROSSOVER TUBE FAILURE	↑ OR ↓	—	↑ +	—
DETACHED OR CRACKED LINER	↑ OR ↓	↑	↑	—

3. Combustor Inspection or Overhaul — This should be based on equivalent engine hours which is based on number of starts, fuel and temperature.

TURBINE ANALYSIS

To analyze a turbine, it is necessary to measure pressures and temperatures across the turbines, shaft vibration and the temperature and pressure of the lubrication system. Table X 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 imbalances.
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 here 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 turbine vibration.
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.

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 corrected simply by washing or cleaning 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 22 shows the significant dollar cost penalties that occur when operating a turbine at a very small percentage efficiency degredation.
3. Table XI shows a load distribution program for a 87.5 MW power station comprised 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.

TABLE X. TURBINE DIAGNOSIS.

	η_t	P_3/P_4	T_3/T_4	VIBRATION	ΔT BEARING	COOLING AIR PRESSURE	WHEEL SPACE TEMPERATURE	BEARING PRESSURE
FOULING	↓	—	↓	↑	—	—	↑	—
DAMAGED BLADE	↓	—	↓	↑	—	—	—	—
BOWED NOZZLE	↓	↓	↓	↑	—	—	↑	—
BEARING FAILURE	—	—	—	↑	↑	—	—	↓
COOLING AIR FAILURE	—	—	—	—	↑	↓	↑	—

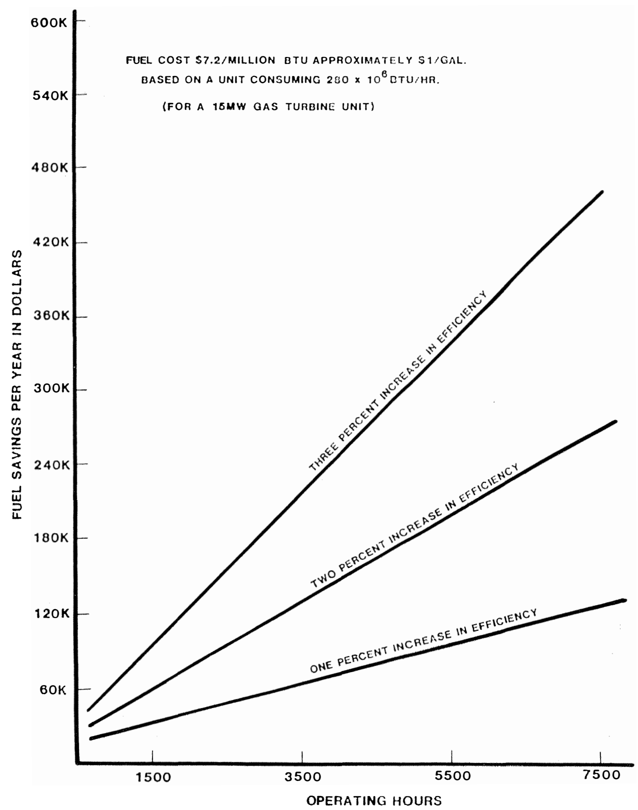


Figure 22. Savings vs. Efficiency.

TABLE XI. LOAD SHARING PROGRAM.

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 DEMAND 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 Demand = MW (Max. Demand = 87.5)

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 Table XII. While this is a general 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. Diagnostics should be based on sophisticated software that combine a large set of information and can provide a diagnosis.

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 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:

- 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.
- 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.
- Retrieval by Turbine Exhaust Temperature — The exhaust temperature can be an important parameter in

TABLE XII. VIBRATION DIAGNOSIS.

USUAL PREDOMINANT FREQUENCY*	CAUSE OF VIBRATION
0-40% Running Frequency	Loose assembly of bearing liner, bearing casing, or casing and support
	Loose rotor shrink fits
	Friction induced whirl
	Thrust bearing damage
40-50% Running Frequency	Bearing support excitation
	Loose Assembly of bearing liner, bearing case, or casing and support
	Oil whirl
	Resonant whirl
	Clearance induced vibration
Running Frequency	Initial Unbalance
	Rotor bow
	Lost rotor parts
	Casing distortion
	Foundation distortion misalignment
	Piping forces
	Journal and bearing eccentricity
	Rotor bearing system critical
	Coupling critical
	Structural resonance
	Thrust bearing damage
Odd Frequency	Loose casing and support
	Pressure pulsations
	Vibration transmission
Very High Frequency	Gear inaccuracy
	Valve vibration
	Dry whirl
	Blade passage

*Occurs in most cases predominantly at this frequency, harmonics may or may not agree.

failure investigations. An analysis of this parameter can verify the existence of a problem with either the combustor or turbine.

- 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.
- Retrieval by Output Power-in — 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.
- 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.

OPERATING EXPERIENCE OF A MONITORING & DIAGNOSTIC SYSTEM

Figure 23 shows a picture of a health monitoring and diagnostic system which has been installed in a major petrochemical complex. This system monitors over 1200 points on five large compressor trains. These trains are driven by both condensing and back pressure steam turbines as well as by electric motor drives. Total analysis is carried out on all systems by various heat balance techniques. Vibration readings are taken with all other "static" readings every second. Some 300 pressure and vibration points are analyzed through the use of FFT spectrums every 2 hours or when upset conditions are noted. The system deals not only with the turbines and compressors, but also with auxiliary systems. The system was installed in late January 1982 and has been responsible for saving a major catastrophe when it noted a drop in balance flow, thus alarming the operator. A previous problem similar to this had lead to a major failure costing hundreds of thousands of dollars in repair bills and over a month of downtime. This

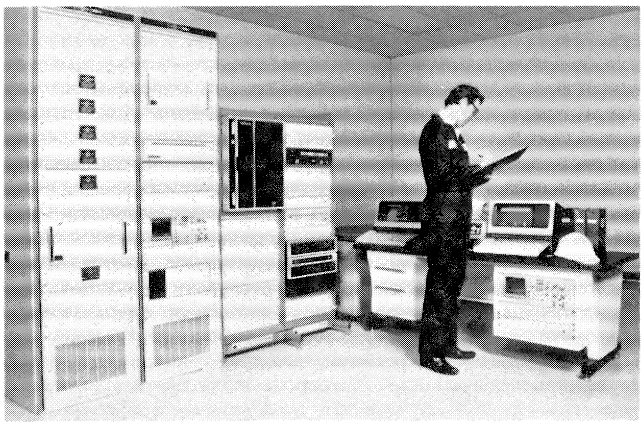


Figure 23. Health Monitoring and Diagnostic System for a Petrochemical Complex.

system is designed to update charts and maps every second so that an operator during startup knows exactly where he is and can thus avoid many pitfalls.

CONCLUSIONS

A. The need for higher turbomachinery train availability and cost effective maintenance make health monitoring and diagnostic systems a necessity. For critical trains, round the clock surveillance calls for computer based systems. The diagnostic software would be a complex set of code that integrates vibration analysis, aerothermal analysis, data bases, logical analyses, interdependent cause-manifestation relationships and statistical techniques. Systems must be carefully designed, keeping the man-machine interface in mind so as to ensure user acceptability.

B. The high cost of machinery replacements and downtime makes machinery operational reliability very important; however, with the currently prevailing and projected further increases in fuel costs, aerothermal monitoring has become very important. Aerothermal monitoring can provide not only increased operational efficiency for turbomachinery but, when combined with mechanical monitoring, it 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.

E. Aerothermal monitoring can allow correction of problems that cause poor efficiency. This can result in very significant savings in fuel costs.