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THE USE OF A MATHEMATICAL MODEL OF A MARINE
GAS TURBINE TO INVESTIGATE THE EFFECTS OF
ENGINE DEGRADATION

University — Université

CARLETON

Degree for which thesis was presented — Grade pour lequel cette these fut presentee

MENG

Year this degree conferred — Année d'obtention de ce grade

1985

Name of Supervisor — Nom du directeur de these

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THE USE OF A MATHEMATICAL MODEL OF A MARINE GAS TURBINE
TO INVESTIGATE THE EFFECTS OF ENGINE DEGRADATION

BY

Greg John Chapman

A thesis submitted to the Faculty of Graduate
Studies and Research in partial fulfillment
of the requirements for the degree of
Master of Engineering

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April 1985

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ABSTRACT

The marine gas turbine operates in an environment and under load conditions unlike any other gas turbine application. As a consequence of operating in the marine environment the gas turbine is subject to various degradations. Such degradations will usually require the gas turbine to operate at a higher burner exit temperature in order to retain a specific power output. As a consequence of the higher burner exit temperature the lives of the hot section components will be severely reduced. In a naval application the loss of an engine can be disastrous. Even with a multi-engine configuration, the chance of a ship completing its mission with the loss of one engine is substantially reduced.

Efforts to minimize engine outage range from simple visual inspections to more extensive engine stripping, component replacement, and reassembly. Unfortunately, these maintenance actions may fail to locate engine problems and/or the actions themselves may initiate other problems. The detection of engine degradation using techniques that do not require engine shut down are highly desirable. Engine Health Monitoring (EHM) consists of various methods that infer the state of the engine from measured engine parameters. EHM includes vibration, oil, and performance analysis amongst others. The analyst's ability to recognize engine problems using EHM techniques relies on prior knowledge as to the affects various engine degradations will have on certain engine parameters. The ideal method of obtaining this knowledge would be to implement the

degradations on the engine of interest and note the subsequent affect on engine operation. This approach is impractical for various reasons, the most important reason, being economics. Secondly, the exact nature of engine degradation is not known and therefore the manner in which a engine component is to be modified for a particular degradation is not clear. Obviously, the most efficient method of investigating engine faults is by using mathematical models.

A model of a marine FT12 gas turbine was developed using generalized component characteristics. The FT12 is single spool, free power turbine engine used on the DDH280 class ships of the Canadian Forces. This model was used to simulate various engine degradations. Various methods for implementing the degradations were investigated and the results compared to available data for a degraded engine.

ACKNOWLEDGEMENTS

It gives me great pleasure to acknowledge and thank my thesis supervisor, Dr. H.I.H. Saravanamuttoo, for his advice and encouragement during the period of this study.

I would also like thank the Canadian Department of National Defence for the engine data which they provided.

Finally, I am grateful to Carleton University for their financial support during the course of this investigation.

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NOMENCLATURE

A	Area (m^2)
b	Bleed mass flow (%)
Cp	Specific heat (kJ/kg/K)
CPR	Compressor pressure ratio (Po_2/Po_1)
EGT	Exhaust gas temperature (K)
EPR	Engine pressure ratio (Po_4/Po_1)
f/a	Fuel/air ratio
ITT	Inter-turbine temperature (K)
m	Mass flow (kg/s)
ma	Inlet mass flow (kg/s)
$\frac{m}{Po} \frac{To}{Po}$	'Non-dimensional' mass flow
N	Rotational speed (rpm)
$\frac{N_{gg}}{\sqrt{To}}$	'Non-dimensional' gas generator speed
q	Dynamic pressure
R	Gas constant for air (0.287 kJ/kg/K)
Pwr	Shaft power (kW)
Po	Total pressure (kPa)
sfc	Specific fuel consumption (kg/kW/hr)
TIT	Gas generator turbine inlet temperature (K)
To	Total temperature (K)

δ	($P_{amb}/101.3$ kPa) ratio
γ	Ratio of specific heats (C_p/C_v)
Δ	Denotes difference between two quantities
η	Component efficiency
η_m	Mechanical efficiency
θ	($T_{amb}/288.15$ K) ratio

Subscripts

amb	Ambient
b	Burner
c	Compressor
f	Fuel
ggt	Gas generator turbine
inlet	Inlet
L	Liner
ref	Design point reference
pt	Power turbine
1,2,3	Station numbers

INTRODUCTION

The requirements for high power density and fast response have been instrumental in the worldwide acceptance and changeover to gas turbine propulsion for small high-speed combatant vessels. It is well known that with aircraft derivative gas turbines, full power from a cold start is available in less than two minutes and an idling engine can be brought to full power in seconds. The gas turbine has shown a steady increase in mean time between overhauls (MTBO) from only a few hundred hours to several thousand hours. This high reliability has been a significant factor in the continued use of gas turbines in naval ships. Efforts to assure the reliability of these engines at sea has invoked the use of engine health monitoring (EHM) methods, which are similar to the successful techniques that the aircraft industry has used.

Long ago the majority of machines, including some power producing devices, were simple and their continued operation was ensured through the ability of the maintenance engineer to observe most of the working parts. If any component began to wear, corrective action was taken before failure of the machine could occur. As a consequence of increased complexity, as with power plants, the accessibility to a machine's working components is reduced. The inability to observe a machine's components during normal operation has resulted in the practice of periodic machine disassembly. During disassembly, machine components which were deemed to be unacceptably worn were replaced and

the equipment reassembled. Unfortunately, this procedure of preventive maintenance has been known to result in further damage to the machine causing its subsequent failure. The thought was once prevalent that it was easier to replace a worn-out component rather than provide an adequate means of ensuring long life. This premise has proven to be quite costly. Maintenance is expensive; it is not just the cost of the part and its replacement, but the fact that a maintenance staff must be available at all times waiting for maintenance actions. If parts and materials are in short supply then equipment is out of service longer or large inventories must be maintained, although this situation is not usually the norm for power plants. Worn components cause secondary problems such as increased vibration (leading to fatigue), shock loading, misalignment, and accelerated wear. Down time for part replacement due to wear causes a loss of production and the inactivity of associated manpower.

History shows that many advances in engineering materials and design have resulted from an intensive investigation of machine failures. Moreover, these investigations have been responsible for increased reliability, reduced maintenance, and increased production from equipment. Although these are tremendous benefits, from a safety and economic point of view, prevention of failure is far more valuable. Various EHM methods have been developed, including oil analysis, vibration analysis, performance analysis, and Gas Path Analysis. Oil analysis is usually not feasible on board a ship and often the obtaining

of the final results requires a long turn around time. Vibration analysis, especially with sophisticated equipment, requires significant expertise and often the equipment onboard only measures an overall level of vibration. Performance analysis involves the monitoring of trends using existing engine instrumentation and requires minimal data computation. Gas path analysis is a logical extension of performance analysis with the intent of isolating engine deterioration to a particular component. This thesis investigates the application of performance monitoring to the P&W FT12 engine operating in a marine environment which is considerably more harsh than that of the aircraft gas turbine. The FT12 is an engine of about 3700 HP, used for cruise propulsion in gas turbine ships of the CF.

The following chapters deal with circumstances under which a marine gas turbine must operate and the likely engine degradations to be incurred; EHM and the nature of collected data for EHM purposes; a description of a mathematical model used to simulate an FT12 engine, and, finally, the simulation of various engine faults and their effect on engine performance.

GAS TURBINE REQUIREMENTS IN MARINE ENVIRONMENTS

2.1 Marine Propulsion Requirements

Few gas turbines are designed specifically for the marine market, rather they are derived from existing aircraft engines. Aircraft engines are adapted to marine use by exchanging specific components (or perhaps coating them) for materials resistant to sea-salt corrosion and the substitution of a power turbine for the nozzle. This substitution is not difficult in theory, since both components have similar mass flow-pressure ratio characteristics. The principle advantage of using a free power turbine is that the torque characteristics are well suited to driving the ship's propeller. Requirements special to a marine propulsion engine include (i) those demanded by the environment, (ii) those demanded by the load and (iii) those demanded by the service.

2.1.1 Environmental Requirements

Aside from the wet, salty environment, marine gas turbines operate under motions of varying magnitude and frequency. A hull in rough water experiences three periodic rotations -- rolling, pitching and yawing; and three periodic translations -- heaving, swaying and surging. Motions of nearly zero frequency are also significant, these being of several types. For example, there are lists, heels and trims that the hull occasionally experiences. As a consequence of these motions the power demand on the engine is continually varying.

2.1.2 Load Requirements

The rotational speed of any practical power plant can be altered to suit that of any propeller, though the speed changer may be complex and expensive. When the engine torque is greater than the load torque, the arrangement is satisfactory, since fuel input can be reduced to lower the engine torque as necessary. However, engine torque capability must be substantially greater than the load torque, since an excess is essential to accelerate the load up to its maximum condition. The free power turbine arrangement satisfies this situation since its torque versus speed characteristic rises more steeply than that of the ship's propeller. The aircraft-derivative gas turbine does not contain the heavy metal components (unlike the steam turbine) that require careful warm-up and can be brought from a cold start to full output in less than two minutes. This attribute, among others, suggests that the gas turbine is suitable as a boost engine in naval and Coast Guard service. Various arrangements of power plants are available for marine propulsion using combinations of gas turbines, diesels or steam turbines. The twin configuration of separate cruise and boost engines has been widely used, since the added complexity of combining the powers from a small-high speed and a large-low speed engine has little benefit. This engine arrangement is commonly known as COGOG ('COMbined Gas turbine Or Gas turbine'). Such is the case in the DDH 280 class ships, where the FT12 engine is used for cruise and an FT4 engine is used for boost. A typical COGOG

configuration is depicted in the following figure.

C - Controllable pitch propeller
 G - Gear box and clutch
 T - Turbine

TYPICAL COGOG ARRANGEMENT

The power requirement changes rapidly in a manoeuvring situation, especially when an emergency stop is required. The gas turbine with power turbine is outstanding in meeting this requirement, since the inertia of its rotating components is small. The inertia of a modern ship hull is very large compared to that of an engine, and thus its acceleration and deceleration are usually not governed by engine response time.

The propulsion load must have the capability of being reversed, that is, the vessel must have the ability to go astern. The method most widely used with gas turbine propulsion is reversal of propeller thrust through reversal

of its pitch, rather than reversal of rotation. The use of this type of propeller (the controllable-pitch propeller, or CPP) relieves the engine or gear box of the necessity to provide reversal.

2.1.3 Service Requirements

Marine service requirements involve weight/volume characteristics, endurance, load profile, reliability, and maintenance practices. Usually, endurance is specified as the distance the vessel can travel on one fuel loading at a certain fraction of rated propulsive power. A load profile describes the intervals of time that an engine operates at various fractions of its rated power. These vary widely with the type of vessel and its service; thus there is no single unique marine load profile. Naval vessels may spend long intervals in port, or they may remain at sea for extended periods. For naval vessels sea time likely requires prolonged cruising at lower power, with occasional bursts of high power during combat or exercises.

It is notable that the load profile largely determines the effective hours of engine wear per hour of actual clock time and thereby is an indicator of the interval between maintenance events. Vessel reliability is the probability it will accomplish its assigned mission, and is affected by many factors, including weather and sea state. The likelihood of a mission being accomplished with an inoperative propulsion plant (even with a multiengine configuration) is substantially reduced. Even with the most thorough maintenance practices the prevention of this

situation is not guaranteed.

2.2 Environmental Considerations

The marine engine operates in an environment with motions of widely ranging magnitudes and frequencies, as well as being wet and salty. The most obvious source of wetness is the presence of water in the atmosphere. Propulsion machinery is seldom directly exposed to spray, however the working fluid must be drawn from the spray-laden atmosphere close to the sea surface. This spray carries salt that will be deposited in the engine with consequent corrosive effects and obstruction of the air passages in the compressor. The design of intake ducting to minimize salt ingestion and operational procedures to remove salt from the engine are important aspects of marine gas turbine technology. Humid sea air promotes corrosion of several common metals, such that components exposed to this air must be of suitably resistant material, or properly protected by a resistant coating. An important aspect of the water problem is the chance of fuel contamination with deleterious consequences to the engine.

2.2.1 Sources of salt

Concentrations of salt varying between 0.01 and 12 ppm (salt to air weight) have been measured in the atmosphere surrounding ships [1]*. The particle size varies from coarse droplets that can be readily observed with the naked eye

*Numbers in parenthesis refer to bibliography

down to particles of a diameter less than 1 micron. The lower the total salt burden, the finer are the majority of the particles, and these particles are the most difficult to collect and measure. In situ measurements indicate that under the conditions likely to be experienced at sea, the salt concentration reaching a gas turbine (after passing through a knitted mesh filter) in a major warship will not exceed 0.05 ppm [1]. Engine intakes make specific and separate provision for mist eliminators, intake screens, emergency inlet by-pass openings, intake silencing splitters, and intake silencing lining. The mist eliminators utilize a series of removable mesh pads which drain the moisture they collect on totrays and then drain overboard via piping. This form of mist elimination requires relatively low air passage velocities (3 to 4.5 m/s) resulting in large surface areas. Often the silencing splitters and other protrusions into the intake passage provide prime locations for salt accumulation which later can break loose and travel through to the hot section of the engine. The practice of cleaning or ballasting (less common) fuel tanks with sea water as opposed to fresh water is another source of salt contamination. Fuel couplings on the deck surface are prone to deck water ingestion during fueling. Furthermore, economic and logistic considerations necessitate that warships burn diesel fuel, which has a greater sulfur content than aviation fuels. The control of contamination of fuel and lubricant supplied to the gas turbine is a combination of well-disciplined and controlled

handling procedures, plus the utilization of effective filtration methods and systems.

Generally, the internal effects of operating in a marine atmosphere are:

1. Corrosion of the compressor section parts resulting from ingestion and deposition of salt.
2. Corrosion of the turbine section parts resulting from highly corrosive molten salt.
3. Sulfidation of the turbine parts caused by burning sulfur bearing fuels in the presence of salt.

2.2.2 Corrosion

The practice of fabricating turbine blades from Ni-base superalloys and their subsequent use in an actual marine atmosphere produces deterioration of the material properties. This deterioration can result from erosion or corrosion. Erosion results from hard particles impinging on the hot section turbine blades and removing material from the blade surface. The particles may enter through the engine inlet or can be loosened scale deposits from within the combustor. Corrosion is described by two processes, hot corrosion and sulfidation. Hot corrosion is an accelerated oxidation of alloys caused by the deposition of Na_2SO_4 . Oxidation results from the injection of salt into the engine combined with sulfur from combustion of the fuel. The corrosion problem includes: (i) erosion, (ii) sulfidation, (iii) intergranular corrosion, and (iv) hot corrosion. Chromium in alloys reduces grain boundary oxidation, whilst high nickel alloys tend to oxidize along grain boundaries.

Age-hardened turbine blades of 10-20% chromium will corrode at more than 1050 K. The addition of cobalt to the alloy increases the corrosion temperature. The use of blade coatings of Al or Al+Cr also provides some resistance to corrosion. Typical turbine blade materials are from the Nimonic series while compressor materials are usually steel or titanium alloys (for marine engine use).

2.2.3 Sulfidation

Sulfidation consists of the depletion of chromium from superalloys by the formation of chromium sulfide causing accelerated oxidation. The most significant point about sulfidation corrosion is that it may also be a problem when commercial fuels of the best purity and quality are used. Sulfur levels as low as 0.05 percent in distillate fuels have been reported to be enough to initiate sulfidation [2].

An optimum composition to satisfy the interaction of stress, temperature, and corrosion has not been developed. The rate of corrosion is directly related to alloy composition, stress level and environment. It is notable that with an increase in chromium content mechanical properties are sacrificed, in particular adequate creep strength and stability at a given temperature.

2.3 Typical Problems Encountered In Gas Turbines

Gas turbines may experience many different types of failure, since the units consist of many components in their overall makeup. Hot section failures are more common than those related to the compressor due to the high

temperatures. Table 1.0 reproduced from Reference [3] depicts some of the more frequent causes of failure for 1-5 MW engines used for electric power generation. The numbers in the table represent the percentage of failures caused by various problems. It is obvious from the table that more than eighty percent of the problems in the gas turbine occur in the hot section. Fuel nozzles and first-stage turbine nozzles appear to be the main problem areas. Similar problems are also likely in a marine situation due to the harshness of the environment.

Another combustion related problem deals with the second stage turbine nozzles. This problem is due to liquids entrained in the fuel or ignition failure at startup. Liquid hydrocarbons carried in the fuel vapour impinge on the turbine, causing hot spots and leading to cracked blades. Ignition failure at startup can lead to pockets of fuel resulting in an explosion and/or fire where the fuel is trapped. The locations for the trapping of fuel are usually in areas of low velocity and where the blades act as flame holders.

Compressor problems are minimized due to the less hostile environment in which it operates compared to a turbine. Erosion of the compressor blade tips and leading edges is not uncommon whilst operating in sandy environments with poor or no filtration. Compressor fouling will reduce the component performance and blade flutter or rotating stall may also result. Abrasive cleaning of the compressor or a water wash will usually restore the blade surface and

thus the performance, however this procedure can also deteriorate the blade profile. Seal problems can give rise to high leakages and bearing thrust problems. The high leakages reduce the efficiency of the unit and can also lead to contamination of lubricants.

2.3.1 Foreign Object Damage (FOD)

Engines in marine environments are unlikely to ingest foreign objects since adequate protection is usually given by the meshes and filters installed in the inlet ducting. However, it is not unknown for bolts, rivets and alike to work loose in the intake air ducts and be drawn into the engine. This situation can be catastrophic for a small axial compressor since it does not have the robust design of the centrifugal compressor. Even if the compressor is not seriously damaged, the rotor is likely to become unbalanced causing vibration. Typically, such damage to the compressor results in a reduction of air flow and efficiency consequently a decrease in available power. Operators compensate by increasing fuel flow, such that the engine operates at a higher turbine inlet temperature and rotational speed. The net effect is that the engine life will be severely shortened. Conceivably an object could pass through the compressor and make its way to the gas generator and power turbines, likely ruining an engine. Generally, the probability of FOD in marine applications (with the exception of battle conditions) is much lower than in other operating environments.

2.3.2 Marine Gas Turbine Maintenance

Maintenance is the process of examination and replacements that are required to assure continued operation of the propulsive machinery (unlike repair). In general, the maintenance process is determined by the type of machinery, however it is inseparable in several regards to the type of service. Quantitatively, little can be stated about reliability, since information on this subject requires extensive data collection during actual service conditions. Such an undertaking has yet to be thoroughly carried out in the marine application of gas turbines. However, the long service of gas turbines in aircraft and industrial service has established a high level of reliability that would be expected to carry over into the marine field providing that the particular features of load (which varies significantly for a marine engine), environment, and service are carefully considered in the engine design and operation.

Scheduled inspection and maintenance procedures for these gas turbines include routine visual inspection, minor inspection, periodic major inspection, and overhaul. A routine visual inspection is simply an external visual check of the gas turbine, including a check of all fluid levels and can often be performed during engine operation. The frequency of these inspections will be defined by the manufacturer and are based on the number of engine operating hours and starts. Minor inspections include a check of the gas turbine's internal filters and screens, and requires shut down of the engine (every 250 hours of operation). Such

an inspection requires approximately 45 minutes and can usually be scheduled during a normal shut down period. A periodic major inspection includes an inspection of the gas generator 'hot section', including fuel nozzles, burner cans, and the first-stage turbine vanes and blades when possible (every 1000 hours of operation). Overhaul is a complete reconditioning consisting of a teardown, cleaning, inspection, reconditioning of parts, reassembly, and testing to ensure like-new performance and serviceability.

The ideal conditions, if attainable, that would permit maximum time between maintenance procedures are;

1. Totally clean combustion air: free of dirt, sea salt, oily vapour and foreign objects.
2. Totally clean fuel: no dissolved or suspended impurities, solid or liquid.
3. Fuel devoid of compositional impurities: no sulfur, vanadium, or other heavy metals, etc.
4. Continuous operation at rated load with minimal thermal cycling due to changes in load or to starts or stops.
5. Uniform distribution of fuel to all combustion chambers, no temperatures above the mean combustion temperature, as might be caused by an excess of fuel to one of the burners.
6. No vibration excitation imposed via shaft coupling, mountings, or ducting connections.
7. A lubricating oil which does not incur any reduction in lubricity and heat capacity.

Under the above conditions there would be some initial random failures. The following might be expected to occur;

1. Component failure due to the use of a substandard material.
2. Blade cracking due to vibration.
3. Ball bearing failure according to their recognized life pattern.

Thermal cycling is an inevitable feature of engine service requirements, and limits life to a less than ideal value. Thermal cycling is likely to be manifested by cracking and distortions in the first-stage nozzles. Sea salt in the combustion air can cause rapid performance degradation by coating compressor passages. This is a transient degradation not requiring repair, nor reducing life (assuming use of corrosion-resistant alloys in the compressor) since salt is removed by periodic washing. More serious and irreversible is the corrosion of turbine vanes and blades due to sulfidation, wherein the sea salt acts in conjunction with sulfur from the fuel. Also, small foreign particles in the combustion air contribute to nozzle and blade degradation via erosion. Dirty fuel can clog fuel nozzles, with consequent local over-heating as the remaining nozzles accept the diverted fuel. Numerous other types of engine degradation are possible; it is the object of maintenance and EHM to detect and correct such situations.

2.3.3 Expected Maintenance Procedures

Planned engine maintenance is largely one of inspection programs. Engine maintenance is intended to be organized around four levels of periodic inspection: running inspections, service inspections, minor inspections, and major inspections. The running inspection is a continuing

observation of engine condition as indicated by the normal monitoring instrumentation. Included are readings of instantaneous values and observations of trends that may point to impending trouble. For example an increase of exhaust gas temperature at a constant-power level may indicate a problem with gas generator turbine erosion. However, gas generator erosion is not the only possible cause of an increase in exhaust gas temperature. This temperature is measured by an array of thermocouples (usually eight or more) spaced around the exhaust duct. If the variation among the individual instruments increases, even though the average temperature remains constant, a combustion chamber problem may be indicated (perhaps the fuel nozzles are clogging) probably resulting in further damage to the turbine blades. A continuing increase in vibration level usually indicates a growing imbalance in the rotors and/or bearing damage.

The service inspection is primarily a check on fuel and lube-oil filters, control linkages, external piping and wiring, and the combustion chambers if readily accessible. Minor inspections are less frequent and usually deal with the hot-section components. The procedure may require extensive opening of casings, or may rely primarily on internal inspection by borescope. A major inspection encompasses a thorough inspection of all engine wearing components. The frequency of these inspections depends somewhat on operating conditions and on experience with rates of deterioration.

The aircraft and aircraft-derived engines differ from industrial engines when a major replacement is required. With the aircraft type engine it is intended that the gas generator is replaced as a unit. Although the free power turbine is intended to remain in the ship for its entire life, some ships have had a power turbine replaced. The power turbine is included in hot-section inspection, however the rate of deterioration is expected to be much lower than that of the gas generator turbine due to the lower gas temperature entering the power turbine. The preceding paragraphs cover the more outstanding aspects of marine gas turbine operation. The following chapter investigates the requirements of EHM and its application to the marine gas turbine.

ENGINE HEALTH MONITORING

3.1 Related Topics

The object of engine health monitoring (EHM) is to take existing engine information and present it in various forms that provide additional insight into the state of the engine. Some definitions of reliability, maintainability and health monitoring are appropriate before proceeding.

Reliability is the probability that a device will perform its specified function for a given period of time under a given environmental condition. Reliability of a gas turbine in a marine installation must embrace not only the basic prime mover but also all of the auxiliaries essential to the prime mover operation. Thus, if the auxiliary systems are proven devices of a simple nature and limited in number, then the basic prime mover becomes the more dominant factor in the system's overall reliability, nonetheless the auxiliaries may not be a simple problem in practice.

Maintainability is the degree to which the defective components can be identified, replaced, and examined. Improvements in maintainability revolve around inspection techniques from borescope ports to fully modular engine design. Health monitoring is a general term encompassing all supervisory inspection techniques which aim at locating defective parts. Monitoring methods include visual, vibration, acoustic, performance, and oil analysis.

The gas turbine, unlike piston engines, does not incur a close distribution of all failure modes. Rather, hot end failures may be of the order of 3,000 hours and cold end

failures may be in the 20,000 hour range or longer for both. Such a failure characteristic does not lend itself to a fixed overhaul life. In the past, components were replaced as required leading to unpredictable part lives and subsequently no clear time as to when a complete overhaul was required. The other alternative available to operators was to overhaul at fixed time intervals which increased with experience. Eventually, the gas turbine achieved such an excellent time between overhauls that the concept of 'on-condition' maintenance became widespread. Consequently, the basis for EHM was established.

3.2 Failure Modes

If machine failure frequency versus operating hours is plotted for nearly any machine the familiar 'bath tub' curve is obtained (figure below).

BATH TUB CURVE

The diagram can be categorized into three sections, ie, early failures, random failures and time dependent failures. Early failures are the result of using substandard parts caused by variation in material quality and/or manufacturing tolerances and misalignments. The random failure mode is normal for new pieces of equipment brought into operation. For this type of failure health monitoring can provide little or nothing to aid in the prediction of a failure. The time dependent failure mode consists of failures that are instantaneous, time dependent and delayed time dependent as shown in the following figure.

FAILURE MODES

The consequences of instantaneous failures can only be alleviated through redundancy devices, this situation is typical of electronic devices. Fatigue failure is another example of an instantaneous failure, there is no forewarning

and it cannot be predicted. The time dependent failure is the classic reason for fixed overhaul periods. The delayed time dependent mode of failure is a variation of the time dependent failure where both the delay and the rate may be variable.

These last two failures are the ones that health monitoring techniques attempt to predict or provide information about. The time dependency of the failure allows for a series of measurements which should show a degradation trend. Eventually, the condition will have deteriorated to a point where maintenance action will be required.

3.2.1 Primary Health Indicators

In the acquisition of data from any machine for the purpose of determining its health there is always some difficulty in choosing the best parameters to measure. The best parameters are those that indicate the general state of one or more of the engine's components. Ideally, a single parameter is preferred rather than several which have to be interrelated in some manner. Such indicators are difficult to come by in a gas turbine due to the variety of ways that any one parameter can change. It has been said that two of the best indicators of the state of a gas turbine are turbine inlet temperature (TIT) and inlet air flow. Air flow reduction is a sure indication of compressor performance degradation or FOD while a rise in TIT is usually an indicator of hot end trouble (can also be caused by compressor problems). Techniques are available to measure both parameters under laboratory conditions, although TIT is

seldom measured. However, the same techniques are not practical in the field. Clearly, to be cost effective, existing instrumentation must be used; once the operator has developed confidence in the system other instrumentation can be added. This, however, is the operator's prerogative. The installation of additional instrumentation on an engine can result in a significant increase in system complexity with minimal gain.

If the engine is treated as a black box where fuel flow is the input and power the output, a loss in power for a given fuel input is a sure indicator of engine problems. Power, however is not easily measured in a marine installation and fuel flow is seldom measured other than the daily consumption. Performance monitoring aims at using the available instrumentation to infer the general thermodynamic condition of the engine components. Data trending and gas path interm analysis are the common techniques used, and these will be considered in turn.

3.2.2 Data Trending

Data trending consists of plotting the deviation of some parameter from a reference value at a certain power (or speed) condition. Data trending has been successfully applied in the aircraft and pipeline fields. In both these applications the engines operate at constant and moderate power levels for extended periods. This situation is ideal for engine health monitoring since the thermocycling is minimal. Available data is collected, standard day corrections are applied and the deviation from a baseline

condition is plotted versus operating hours accumulated. It is not unusual for an initial deviation other than zero to occur, this may not be of concern provided the engine is known to be in good working order. Such deviations are attributed to slight differences in manufacturing tolerances and engine age. The objective is to detect changes considered to be related to engine degradation as the number of operating hours increases. Efforts to extend the useful range of data to lower power conditions may make use of a reference speed. The method involves either assuming that the deviation at the lower speed will be the same at the higher speed or that the slope of the current engine curve is the same as the reference curve. The slope method involves calculating an expected deviation at the reference condition based on the slope of the curve. The rationale for carrying everything back to a reference condition is so that percentage deviations from the baseline can be plotted. If extensive baselines are available then the current value of a measured parameter can be compared to that of the baseline at the recorded engine speed. Saravanamuttoo and Williams [5] provided such a technique for the monitoring of FT12 and FT4 engines.

3.2.3 Gas Path Analysis

The objective of gas path analysis is to determine more information about the observed engine degradation. That is, through the use of energy balances throughout the engine and isentropic relations for the components, an attempt is made to isolate the deteriorated component. The method

relies on predicting the effect of small changes in component behaviour and the analysis is usually presented in the form of fault matrices. The technique is also very good for detecting instrumentation problems, it can also be quite sensitive to slight errors in key parameters such as fuel flow and thus must be applied with caution. The technique is well documented in References [6], [7], and [8], and will not be further discussed. Generally, the above methods are limited to high power conditions.

3.2.4 Establishment of a Baseline

The previously described engine health monitoring techniques require the initial establishment of a baseline or set of baselines. On a fleet wide basis the establishment of baselines for each engine of the same type may become unwieldy. Typically, one general set of baselines for all engines is used instead. Even for new engines off the production line some variation in performance will exist due to manufacturing tolerances, slight misalignments, etc. However, the general form of the performance characteristics are expected to be nearly the same. The user may be able to obtain manufacturer's data, however the variation of the measured parameters provided may only cover a relatively small portion of the entire range of interest. A second difficulty is that if power turbine speed and power (or thrust) characteristics are provided, they are usually based on some propeller load or nozzle exit area. For gas turbine operators wishing to initiate an EHM program for engines in service, manufacturer's baselines may not be adequate. Since

such engines may have undergone several overhauls, clearances may have increased, blade profiles may have changed (erosion/coking), etc. Such engines are unlikely to produce the same performance characteristics as when they were first fabricated. This also implies that for an operator of several engines that have seen extensive service, the use of a generalized set of baselines for all engines may not be possible. In light of the above discussion the user may have to carry out performance measurements to establish a 'new' set of baselines. For the operator of a small number of engines the use of individual baselines may be advantageous. If the user develops baselines for the engines in service, the first hand experience with the data scatter likely to be encountered is obtained. This aspect is very important to the success of the user's detection of an engine and/or instrumentation degradation. Conceivably, those degradations deemed significant (eg. causing a rise in exhaust gas temperature of 25°C) may be of the same order of magnitude as the data scatter. The success of any engine health monitoring program will ultimately depend on the analyst's ability to determine if the engine's operation is normal (ie. no change in the overall operating level and data scatter). This goal is best realized if the user has as much experience with the engine(s) as possible. If the user is to establish baselines, the data must be collected over the operating range of interest as well as at steady state power. Several recordings of the critical parameters at each of the

selected conditions are necessary to determine the normal data scatter. Finally, it is imperative that the user organize the data sheet such that the critical parameters are recorded first and within a short time frame, and secondary parameters (such as lubricant temperatures and pressures) follow.

3.3 Instrumentation

It is obvious that the success of engine health monitoring will be heavily dependent on the accuracy and reliability of the measurements recorded. The deterioration of instrumentation with time will introduce further deviations in the observed engine performance. A method of instrument validation is necessary to separate deviations due to changes in plant performance from those caused by instrumentation problems. Ellis [9] of United Airlines for example stated 'the flight log monitoring system (FLMS) detects more faulty engine instrumentation than faulty engines each month'.

The concept of validating a measurement suggest that a comparison with some known and reliable information must be made in order that logical conclusions can be drawn from the data themselves. For the best results, a combination of logical analysis and baseline data comparison are required for daily measurement validation. The possible interpretations resulting from the analysis are;

1. Both the plant and instrumentation are in acceptable condition.
2. An item of instrumentation has degraded or failed.

3. The engine state of health has changed.
4. Both the engine state of health has changed and an instrumentation degradation or failure has been incurred.

Generally, instrumentation failures are detected easily since in the trending of corrected parameters only one will usually be affected. Engine degradation results in the rematching of the respective component operating points, each mode of deterioration should produce a predictable change in the operating point.

3.3.1 Data Acquisition and Recording

The practice of automated data collection and recording is becoming more common. In the past, critical machine parameters were recorded manually, usually by an employee with a limited technical background. Often the individual was provided a data sheet which was laid out without thought to the primary parameters of interest. Moreover, management often did not have the technical background to realize which parameters were critical. Such situations have resulted in the collection of data which was lacking or recorded over a period of time longer than the event of interest. The listed data was frequently only a gross approximation to the true state of affairs and may, on investigation, be related to something entirely different from that intended by management. Furthermore, missing values were numerous and it was not unknown for the data to be partially or wholly fictitious. Further aggravating the situation, usually a long time elapsed between the time of

data collection and subsequent analysis. Subsequent analysis of the data produced nothing of observable use and consequently the practice of analyzing data was often discarded.

It is imperative that engineers and their management understand the uncertainty from instrument errors, the influence of the errors on calculated performance, and acceptable methods for estimating the errors. It is also imperative that the users of the data do not influence errors through carelessness and/or lack of understanding of the techniques being applied. Two types of data errors are prevalent;

1. Systematic (a reading that is consistently higher or lower than the actual).
2. Precision (variations between repeated measurements of a parameter under constant operational condition).

The correct interpretation of engine state from transducer data cannot be severed from the practical, since the location and environment of the transducer may introduce problems. For condition monitoring of gas turbines the measurement of pressures, temperatures, rotational speeds and vibration levels are required at various power plant speeds and vibration levels is required at various power plant locations. All these transducer signals suffer to some extent from signal scatter due to the random fluctuations which reduce the immediate usefulness of such signals for condition monitoring. Therefore extensive conditioning of the incoming signal may be required to remove the worst of

this scatter prior to the use of the data for EHM purposes. In practice it is not necessary to insist too rigorously on the absence of errors in the measured values of the independent variables(s), provided the errors are small compared to the ranges covered and with the dependent variable(s) errors. The user of EHM must be aware of the effects of combining measurements and error propagation. The task of determining instrument error and the associated confidence levels is not easy, a simplified method is provided in reference [10].

It is quite true that the study of plant records (when much time has elapsed), is much inferior to carrying out properly planned, and realistic, data gathering on the full scale plant. Despite this reservation, there is frequently no immediate alternative to such an analysis; the data already is on hand and could contain useful information. Even if the analysis does not always directly result in improvements to the machine's performance, it can at least give indications which variables are likely to be worth considering in a subsequent analysis.

To date performance monitoring has not had a large success rate in marine applications. This in part is probably due to the extra work load on the engineering staff combined with the fact that in-service degradation is not large. Also, partly due to those who do not have confidence in EHM and their simplest alternative is to do nothing. Each ship at sea is akin to a small user. Ultimately, the success of decisions made by the marine

engineering officer will be heavily dependent on the quantity and, particularly, the quality of engine data available. The instrumentation implemented on most naval installations is primarily concerned with safe operation rather than performance monitoring.

Figures 1 - 6 clearly indicate that data collected automatically can indeed have a low amount of scatter. Each figure covers approximately the cumulation of two hundred engine operating hours. The engine of interest had been in service several thousand operating hours prior to the collection of this data. In each case the first and second graph of engine pressure ratio (EPR), power turbine exit temperature (To_5), and compressor pressure ratio (CPR) encompass approximately one thousand hours of operation. The values along the axes have been removed since the information is proprietary. A problem with manual data collection is that the data sheet provided usually requires the recording of a large number of parameters. Of these measurements only a few are critical for performance evaluation, consequently the time required to collect all the data may extend over a period during which the operating condition may change. Ultimately, any further analysis requires the use of credible data otherwise conclusions drawn are likely to be erroneous.

3.3.2 The Nature of Data Scatter

The signal scatter on pressure and temperature measurements made in the gas stream are such that they may exceed the changes which condition monitoring seeks to

detect, for diagnostic and predictive purposes.

Several observations dealing with the nature of the scatter are:

1. Transducer signal scatter during so-called steady state plant conditions is normal.
2. Transducer signal scatter severity for parameters in the gas path increase with distance through the engine, since the working fluid is subject to increased distortion from inlet to outlet.
3. The experience of others [12] indicates that the larger the engine in terms of size and power, the better the gas stream parameter signal quality.
4. Prime movers subjected to power changes experience reversible hysteresis effects on the gas stream measured parameters, which have a finite time dependency. This is particularly noticeable with hot gas temperatures when moving from a high to a lower power condition, the temperature will take some finite time to settle to a lower value. It is important that this has occurred before recording the data as being steady state. This phenomenon can be a major source of error in the analysis of the observed measurements.
5. Measured pressures and temperatures in the gas path are location sensitive in absolute value terms and also engine speed dependent. Usually this is not a problem as such in actual condition monitoring, however, it does imply that direct comparison between prime movers is difficult and complicates transducer replacement.

6. Condition monitoring predictions can only be performed on data which has been collected whilst the gas turbine can be considered as operating in a steady state condition.

Scatter can obscure trends and consequently must be minimized.

1. Scatter has no meaning in the dynamic sense, else the gas turbine would behave erratically which is contrary to practical observation of prime engine parameters.
2. In terms of condition monitoring if scatter had an attached implication, the deterioration of the engine would be indicated, however the same data can be obtained from an engine said to be in 'good' condition.

The first important step in reducing the scatter of 'good' data is the correction to ISO conditions (15°C and 101.3 kPa). All internal engine data are corrected to standard ambient intake pressure and temperature. It is assumed that the applied correcting factors do not introduce any inaccuracies other than those due to scaling and resolution, although the correction factors are non-linear. Typical ranges for ambient temperature and pressure are -6 to 30°C and 98.6 to 103.6 kPa respectively [11]. Typical plant usage will determine the range of engine operation which the majority of the data will be collected. This is of interest since it defines the type of data likely to be collected and subsequently available for condition monitoring purposes. Frequent starts, stops, accelerations and decelerations from one power condition to another

produce changes in engine parameters which deviate from engine baseline conditions. These conditions along with the previously mentioned factors produce 'rogue' points, from a condition monitoring point of view, however such points are readily observable and easily eliminated from further analysis.

ENGINE COMPONENT MODEL DESCRIPTIONS AND MATCHING

4.1 Component Models

The use of mathematical models is the most efficient (and usually the only feasible) method of systematically investigating the various levels of component degradation due to fouling, erosion, mechanical damage, foreign object damage, etc. However, often the actual component data may not be available. The possible reasons are (i) proprietorship, (ii) the data may be lost in antiquity (this is usually the case for compressors), and (iii) the component may have never been fully tested (this is usually the case for turbines). Thus, simulated component characteristics will have to be substituted for the actual component data. Indubitably, simulated or generalized component models are no substitute for the true component characteristics, however they are the next logical progression from analysis using the small change data approach, as described in Reference [8]. Such models enable the full operating range of the engine to be considered. Clearly, the most important advantage of these models for fault diagnosis is the controlled implementation of various engine degradations through modification of the component characteristics. Once the equilibrium running line has been established for a series of speeds, it is a relatively simple matter to obtain the variation in engine parameters with speed or some other variable. The following sections describe the characteristics for the engine components used to model an FT12 engine.

4.1.1 Inlet Losses

Inlet performance is secondary to the rotating components as far as component matching is concerned. However, the inlet will be dealt with first since it affects the flow entering the compressor and thus influences the engine performance. The variation of inlet loss with rotational speed $N_{gg}/\sqrt{\theta}$ was obtained from data of an inservice FT12 gas turbine (Figure 7). After matching of the modeled engine components proved satisfactory, a curve fit was applied such that inlet losses were described as a function of absolute mass flow at inlet instead of gas generator speed. The inlet pressure loss must be input as a function of mass flow. This is necessary for the modelling of compressor fouling which at a constant gas generator speed causes a reduction in mass flow and hence inlet pressure loss. It can be shown that inlet loss is proportional to $(m_1\sqrt{T_{01}}/P_{01})^2$ [13]. However, absolute mass flow was used as the independent variable since an equation describing the loss data over the expected range was more easily fitted. It is felt that no serious error is incurred by not using non-dimensional mass flow to describe the intake loss since the affect is adequately modeled using the absolute mass flow. The final equation is provided in Appendix A.

4.1.2 Axial Compressor Characteristics

Axial flow compressors consist of a series of stages, each stage having its own characteristic. The stage characteristics are similar to that of the entire

compressor, however, the pressure ratios are substantially lower (of the order 1.1:1 to 1.4:1). The mass flow through the compressor is controlled by choking in various stages, these stages may be the early or the later depending on the conditions. At low rotational speeds the rear stages choke due to a high velocity resulting from low pressure and temperature rise producing a low density at the rear of the compressor. At high rotational speeds the latter stages can pass all the flow provided by the early stages, however, eventually the first stages choke. A typical axial compressor characteristic is provided in Figure 8.

Saravanamuttoo and MacIsaac [14] studied available multi-stage compressor maps and developed a generalized map (reproduced in Figure 9). The reference condition occurs at 100% non-dimensional rotational speed and isentropic efficiency (design-to-reference). The use of such a map is a good first approximation in engine modeling, however, it is no substitute for that of the actual components. The most difficult aspect of the use of such a map is the application of scaling factors to distort the mass flow and efficiency contours such that the resulting component matching produces the desired equilibrium running line. Care must be taken so that the map retains reasonable characteristics about the running line. Otherwise when faults are introduced the new running line (although lying close to the original) will not be representative of the sought degradation. The user is unlikely to obtain a satisfactory agreement between the model and engine at all rotational speeds; a better matching

at the higher rotational speeds is preferred since actual engine degradation is expected to be more evident at these conditions. Equations fitted to the lines of efficiency and non-dimensional mass flow against pressure ratio and rotational speed for the generalized compressor map are provided in Appendix A.

4.1.3 Axial Turbine Characteristics

Whether calculated or measured the axial turbine performance is normally presented by plotting $m_3\sqrt{T_{03}/P_{03}}$ and η_t against pressure ratio P_{03}/P_{04} for various values of $N_{gg}/\sqrt{T_{03}}$ as shown in Figure 10. The efficiency plot shows that η_t is nearly constant over a wide range of conditions due to the accelerating nature of the flow. Similar to a nozzle there is a maximum value of $m_3\sqrt{T_{03}/P_{03}}$ that is obtained at some pressure ratio. Choking may occur in the nozzle throats or in the rotor annulus. When choking occurs in the throat (the more common situation) the speed lines merge into a single horizontal line. A generalized turbine map is depicted in Figure 11 and has been derived from data given in Reference [15]. Equations fitted to lines of efficiency and non-dimensional mass flow versus pressure ratio and rotational speed are provided in Appendix A.

4.1.4 Exhaust Losses

The original FT12 data available did not measure the parameters required so that exhaust pressure loss could be determined. However, in both References [5] and [11] the exhaust loss was assumed to be equal to the inlet loss. This

assumption allowed values of EPR to be calculated.

4.2 Combustion Chamber Modeling

The role that the combustor plays in component matching is secondary to the rotating components. Assuming a constant burner efficiency and constant non-dimensional burner pressure loss is adequate for most modeling. However, if the effect of a degraded engine on fuel flow is to be investigated a more detailed combustor model is required. The following sections introduce the basic equations for modeling combustor efficiency and pressure loss.

4.2.1 Pressure Loss

Pressure losses in combustors are comprised of two basic independent sources. Firstly, there is the cold pressure loss resulting from viscous friction and turbulence. Secondly, there is the hot fundamental loss due to combustion. The total pressure loss is given as follows.

$$\Delta P_{23} = \Delta P_{23\text{cold}} + \Delta P_{23\text{hot}} \quad (1)$$

It can be shown that [16];

$$\frac{\Delta P_{23\text{cold}}}{P_{02}} = \frac{\Delta P_{23}}{q_{\text{ref}}} \frac{R}{2} \left[\frac{m_2 \sqrt{T_{02}}}{A_{\text{ref}} P_{02}} \right]^2 \quad (2)$$

Where $(\Delta P_{23}/q_{\text{ref}})$ is known as the cold pressure loss factor and denotes the flow resistance introduced into the airstream between the compressor exit and the turbine inlet. The quantity q_{ref} is the dynamic pressure at the design point. Unlike the total pressure loss, which is operating condition dependent, the pressure loss factor is a fixed

property of the combustor. this quantity represents two separate sources of pressure loss: (1) the pressure drop in the diffuser and (2): the pressure drop in the liner. R is the gas constant for air and $m_2 \sqrt{T_{02}} / P_{02}$ is the non-dimensional combustor inlet mass flow. For uniform mixtures flowing at low Mach numbers in a constant area duct, the hot pressure loss derived from momentum considerations can be shown to be related to the total temperatures by [16],

$$\frac{\Delta P_{23hot}}{q_{ref}} = \frac{T_{03}}{T_{02}} - 1 \quad (3)$$

Analysis of the factors governing the hot pressure loss in practical chambers indicate that relationships of the following form are applicable.

$$\frac{\Delta P_{23hot}}{q_{ref}} = K1 \frac{T_{03}}{T_{02}} - K2 \quad (4)$$

where $K1$ and $K2$ are determined experimentally. Typically, the hot pressure loss is 0.5 to 1.0 percent of the compressor delivery pressure.

4.2.2 Efficiency

The combustion efficiency of modern gas turbines is nearly constant at 0.99 over their operational range. For most combustor modeling simply assuming a fixed value is adequate. Nevertheless, a method for estimating the variation in burner efficiency has been included. From Lefebvre [16] the following relationship was obtained

$$\eta_b = f \frac{P_{02} A_{ref}}{m_2 \sqrt{T_{02}}} \left[\frac{\Delta P_L}{P_{02}} \right]^{0.5} \quad (5)$$

where ΔP_L is the liner pressure loss. Derivations of the pertinent constants used for modeling are presented in Appendix B.

4.3 Propeller Characteristics

Although the propeller is not an engine component its operational characteristics significantly influence the power turbine operation. Thus a discussion of propeller operation and its interaction with the engine and ship is appropriate.

Generally, the following hold true for a fixed pitch propeller (these statements constitute the 'propeller law');

1. Propeller speed is proportional to ship speed.
2. Propeller torque is proportional to (propeller speed)².
3. Propeller power is proportional to (propeller speed)³.

TYPICAL RESISTANCE CURVE FOR A DISPLACEMENT HULL

Source: Reference [17]

The previous conditions apply for a hull operating at a low speed-length ratio such as the displacement hull of the DDH-280 class ships. A typical resistance curve for a displacement hull is presented in the previous figure.

As seen from the figure the ship's resistance rises rapidly with speed and since a fixed pitch propeller has similar characteristics it is well suited for this kind of hull. However for low speed maneuvering and ease of implementing reversal without a gearbox most gas turbine powered ships use a controllable pitch propeller (CPP). In fact, the CPP was virtually unknown before the marine gas turbine was introduced, and basically was brought in to satisfy the ship reversal requirement. The use of a CPP eases some of the aspects of the matching problems due to varying hull resistance since pitch can be adjusted in service. However, this second degree of freedom (the other being fuel control) also allows operation of the propulsion plant at undesirable points. For every condition an optimum propeller pitch and fuel flow exists. The designer usually removes the second degree of freedom by linking pitch control and throttle control in a definite program. The matching of an engine and propeller is an application of the principle of conservation of energy.

If hull resistance increases, as caused by roughening of the surfaces by corrosion and/or barnacles, or additional ship weight (resulting in the ship resting lower in the water) the proportionality between resistance and speed changes, specifically, the resistance increases. Whatever

propeller power versus rev/min curve (as shown in the following figure) is chosen in design, it is unlikely to remain unchanged in service.

EFFECT OF SHIP RESISTANCE ON REQUIRED POWER

Source: Reference [17]

Thus, for a given propeller speed (or ship speed) an increase in power will be required. Therefore, the propeller curve tends to shift in service (A to A', above figure) and loss of power must be accepted or the engine must be operated above its rating (B to B') to match the original power (increase in TIT). It is notable that a variation of weather and sea state can also have a similar affect as above.

The following figure shows the simplified relationship between control lever angle, propeller pitch, and shaft speed.

PROPELLER PITCH AND SPEED VERSUS CONTROL LEVER ANGLE

Source: Reference [18]

For low vessel speeds, the propeller pitch can be reduced to a low value while rev/min remain high. Although the propeller becomes very inefficient, its required input power remains sufficiently high to load the engine adequately for self sustained operation. The usual practice is to use the CPP as a fixed pitch propeller at a higher rev/min whilst at a lower rev/min the pitch is adjusted to maintain constant shaft speed. For the DDH-280 Canadian Destroyer Class the CPP usually operates at a fixed pitch from 12 knots and above while at lower speeds the engine rev/min is held constant. The required control system is quite complex and usually contains the the following features.

1. A fuel control which sets gas generator power.
2. Integral speed limiting or topping governors which are set above the maximum free turbine operating speed.
3. Automatic torque limiting to reduce engine power if over-torque is sensed on the shaft torquemeter.
4. An 'anti-windmilling' control on the pitch control which coarsens pitch in the event of propeller shaft overspeed.

These last three items are likely to be encountered in rapid acceleration or deceleration situations. Since such conditions are transient, they are not of direct interest to EHM. To model such a control system would be difficult even with detailed information which is proprietary. A more detailed description of the controllable pitch propeller, its operation and engine matching can be found elsewhere, reference [17]. Although, the FT12 engine modeled drove a CPP no attempt was made to model this situation primarily due to lack of information regarding the control system. Nonetheless, the background provided will aid understanding of inservice engine data to be presented later. The next sections describe the methods used to obtain a running line for the engine model.

4.4 Governing Equations for Engine Component Matching

The governing parameters for component matching are; conservation of energy, compatibility of flow, and equal mechanical speed of components on the same shaft. Consider first the compressor section of the engine. The total temperature rise from the inlet of the compressor (station 1) to the diffuser exit (station 2) is obtained from

the isentropic relationship

$$\Delta T_{012} = \frac{T_{01}}{\eta_c} \left[\left(\frac{P_{02}}{P_{01}} \right)^k - 1 \right] \quad (6)$$

where $P_{01} = P_{amb} - P_{inlet} \quad (7)$

and $k = (\gamma - 1) / \gamma \quad (8)$

The total temperature drop from the inlet of the gas generator turbine (station 3) to the turbine exit (station 4) is obtained from considering a work balance between the compressor and the turbine, given by

$$\dot{m}_1 C_{p12} \Delta T_{012} = \eta_{m1} \dot{m}_3 C_{p34} \Delta T_{034} \quad (9)$$

where η_{m1} is the mechanical efficiency of compressor-gas generator turbine shaft and accounts for the losses due to windage and friction.

The compatibility of speed requires that the compressor and gas generator turbine turn at the same mechanical speed since they are connected to a common shaft. Consequently, their non-dimensional speeds are related by

$$\frac{N_{gg_t}}{\sqrt{T_{03}}} = \frac{N_c}{\sqrt{T_{01}}} \sqrt{\frac{T_{01}}{T_{03}}} \quad (10)$$

It is notable that if a speed-independent turbine flow map is employed, the relationship between speeds is only useful in dealing with the corresponding speed-dependent efficiency map. The total temperature drop across the gas generator turbine is related to the corresponding pressure

drop through the following isentropic relation

$$\Delta T_{034} = \eta_{gg} T_{03} \left[1 - \left(\frac{P_{04}}{P_{03}} \right)^k \right] \quad (11)$$

Considering the engine layout in Figure 12, it is evident that several physical laws must be obeyed. To begin with, flow compatibility (conservation of mass) requires that during steady-state operation, the mass flow through all components must be equal, except for bleeds, leakages, and fuel supplement. Hence, for the gas generator, expressing the mass flow in terms of non-dimensional groups gives

$$\frac{\dot{m}_3 \sqrt{T_{03}}}{P_{03}} = \frac{\dot{m}_1 \sqrt{T_{01}}}{P_{01}} \frac{P_{01}}{P_{02}} \frac{P_{02}}{P_{03}} \sqrt{\frac{T_{03}}{T_{01}}} \frac{\dot{m}_3}{\dot{m}_1} \quad (12)$$

where $P_{03} = P_{02} - \Delta P_{23}$ (13)

Referring to Figure 12, the product of pressure ratios across all the components must equate to unity. Thus,

$$\frac{P_{01}}{P_{amb}} \frac{P_{02}}{P_{01}} \frac{P_{03}}{P_{02}} \frac{P_{04}}{P_{03}} \frac{P_{05}}{P_{04}} P_{amb} = 1 \quad (14)$$

The mass flow at any station can be determined if the bleeds, leakages, and fuel added are known. For example, the flow into the power turbine is given as

$$\dot{m}_4 = \dot{m}_1 - \dot{m}_{leakage} + \dot{m}_f \quad (15)$$

The mass flow from the gas generator to the power turbine must be constant, except for leakages and return of cooling air from the compressor. Thus,

$$\dot{m}_4 \sqrt{\frac{T_{o4}}{P_{o4}}} = \dot{m}_3 \sqrt{\frac{T_{o3}}{P_{o3}}} \frac{P_{o3}}{P_{o4}} \sqrt{\frac{T_{o4}}{T_{o3}}} \frac{\dot{m}_4}{\dot{m}_3} \quad (16)$$

The total temperature drop across the power turbine is given by the isentropic relation

$$\Delta T_{o45} = \eta_{pt} T_{o4} \left[1 - \left(\frac{P_{o5}}{P_{o4}} \right)^k \right] \quad (17)$$

The power output is obtained from

$$P_{wr} = \eta_{m2} \dot{m}_5 C_{p34} \Delta T_{o45} \quad (18)$$

where η_{m2} is the mechanical efficiency of the free power turbine shafting and the sfc is given as the following

$$sfc = \dot{m}_f / P_{wr} \quad (19)$$

4.5 Outline of Solution to Off-Design Operation

The mass flow through the compressor and the gas generator turbine must be conserved less any bleeds plus the additional fuel flow. The gas generator turbine pressure ratio (P_{o3}/P_{o4}) will not be known and it must be determined by equating the turbine work to the compressor work and using equation (11). The gas generator is matched to the power turbine by conservation of mass flow through both components, less any leakages. The pressure ratio available to the power turbine is determined by the compressor and gas generator turbine pressure ratios and the ducting losses. The fundamental solution of the off-design calculations is performed by utilising the equations in Section 4.4 while

following the computational outline depicted in the information flow diagram in Appendix C. The computation is initiated by choosing two points on the compressor speed line. These guesses are input by the user for each speed line, the purpose being to minimize the time for convergence. The method of convergence relies upon iterating on CPR. An initial iteration loop is performed to match the compressor to the gas generator turbine; there is a possible match for every CPR chosen. A second iteration loop is required to match the power turbine, which results in a single CPR value which satisfies both the gas generator matching and power turbine matching.

In more specific terms, the procedure for obtaining an equilibrium running point is as follows.

1. Having selected a point on the compressor map, the quantities $N_c/\sqrt{To_1}$, CPR, P_{inlet} , $m_1\sqrt{To_1}/Po_1$, and η_c are determined, then $\Delta To_{12}/To_1$ is obtained from equation (6).
2. A value of Po_3/Po_4 is guessed, the value of $m_3\sqrt{To_3}/Po_3$ is obtained from the gas generator turbine map, enabling the temperature ratio To_3/To_1 to be obtained from flow compatibility, equation (12).
3. This previous value of To_3/To_1 is used to calculate $N_{gg}/\sqrt{To_3}$ from equation (10).
4. Using $N_{gg}/\sqrt{To_3}$ and Po_3/Po_4 the turbine efficiency can be obtained from the turbine map. (however, since the gas generator was assumed to remain choked throughout the operating range a fixed turbine efficiency was used.)

5. The non-dimensional temperature drop $\Delta T_{034}/T_{03}$ is calculated from equation (11) using work balance (equation (9)) another value of T_{03}/T_{01} is calculated.
6. The second value of T_{03}/T_{01} will not, in general, equal the first value obtained from flow compatibility indicating that the guessed value of the pressure ratio P_{03}/P_{04} is not valid for an equilibrium running point.
7. A new value of P_{03}/P_{04} is estimated and the previous calculations are repeated until the two values of T_{03}/T_{01} agree.
8. The value of $m_4\sqrt{T_{04}/P_{04}}$ at exit from the gas generator is calculated from equation (16).
9. The corresponding pressure ratio across the power turbine is established from equation (14).
10. The power turbine is assumed to operate with the flow characteristics of the 100% speed line. Thus from the power turbine map and P_{04}/P_{05} , $m_4\sqrt{T_{04}/P_{04}}$ is obtained.
11. Generally, the second value of $m_4\sqrt{T_{04}/P_{04}}$ will not agree with that obtained from flow compatibility indicating that the guessed value of the pressure ratio CPR is not valid for an equilibrium operating point.
12. A new value of CPR is estimated and the previous calculations are repeated until the two values of $m_4\sqrt{T_{04}/P_{04}}$ agree.
13. The power turbine efficiency is then obtained from its characteristic map and the power output and sfc are calculated for the operating point.

The convergence is very fast and well behaved even if the final solution lies outside the interval bounded by the initial guesses on the compressor map. The required inputs to the program are shown in Appendix C. Included in this data are scaling factors for each compressor speed line and turbine characteristic, the purpose being to modify the characteristic curves such that the resulting running line and performance curves agree with those desired. The choice of these factors is mostly a result of trial and error. However, the running line will most likely pass through the locus of highest efficiency or close to it.

The off-design program was written and implemented on an IBM personal computer. This computer was found to be more than adequate for this work, especially when considering the significant advantages of self-service operation and around-the-clock availability offered by this machine. The program was written and debugged in a modular form, this being reflected by the number of subroutines contained. A program listing is provided in Appendix C. Granted this machine does not have the speed of large mainframes, however, the solution of a running line required less than five minutes using double precision variables. The FORTRAN package used has some non-standard calls for opening and closing files, nevertheless, these differences are not seen as being a major obstacle for a user to implement this program on another system. The results of this study, as computed by the finalised program are provided in the following chapter.

4.6 FT12 Engine Description

The FT12 marine engine is an aircraft derivative based on the JT12 turbojet and the JFTD12A free turbine helicopter engine. The engine consists of four major components, namely a compressor, combustor, gas generator turbine and free power turbine. Air enters the engine through the compressor inlet and is compressed by a nine-stage axial-flow compressor; no variable geometry is used in the compressor. It is expected that the engine makes use of bleed air and/or blow off, however amounts involved and the schedules used could not be found in the open literature. The air is collected from the compressor and diffused into eight cannular combustion chambers. The air is discharged to each combustion chamber via holes in the liner dome and walls and subsequently mixed with fuel sprayed from a duplex nozzle to sustain combustion. A two stage gas generator turbine drives the compressor and a two stage power turbine provides the output power through a reducing gear box to drive the load. The engine layout is typical of a free turbine configuration. More detailed engine specifications for the JFTD12A are presented in Table 2.0. Additional information regarding the FT12 marine engine and related operating experience is provided in References [19,20, and 21].

4.7 Rating Philosophy

The rating of an engine is a term that usually designates the power that it produces in a particular application. A specific free power turbine rev/min at some gas generator turbine speed is always implied or stated, since the development of the intended power depends on the rev/min setting.

The normal or base load rating is established by the importance of extended time between major maintenance periods for the application. The maximum continuous rating is selected for those applications where high output is required and reduced time between major maintenance periods or engine overhauls can be tolerated, such as in marine applications. The maximum intermittent rating or overload rating is offered in conjunction with the maximum continuous rating when more output than the maximum continuous is required for short periods particularly in marine installations. This power level can be used for limited periods of time combined with the maximum continuous power level. Such power conditions are likely to be encountered in pursuit situations for boost engines or during rapid acceleration of the vessel by the cruise engines.

The use of three ratings is arbitrary, many manufacturers publish two. Other complicating factors in ratings are the effects of external conditions. Manufacturer's performance curves are specified for a definite ambient pressure and temperature and zero inlet and exhaust losses. Since duct losses are always present, and

ambient temperature is often greater and/or lower than the stated value over some of the vessel's operating experience, the designer must carefully correct a given rating for the actual conditions. This is carried out using the standard correcting factors of θ ($T_{amb}/288 \text{ K}$) and δ ($P_{amb}/101.3 \text{ kPa}$).

4.8 Selection of the Design Point and Matching Requirements

From data collected from a marine FT12 engine, curves of (Po_2/Po_1) versus $(N_{gg}/\sqrt{\theta})$, (To_5/θ) versus $(N_{gg}/\sqrt{\theta})$ and (Po_4/Po_5) versus $(N_{gg}/\sqrt{\theta})$ (Figures 1, 2, 3, 4, 5, and 6 respectively) were obtained over a substantial range of operation. The absolute quantity of any of the previous parameters at a specific $(N_{gg}/\sqrt{\theta})$ will vary from engine to engine due to manufacturing tolerances, load profiles, overhauls, etc. However, since the object of this work was to study systematically changes in engine parameters as a result of degradation, it is believed that no loss of generality is encountered by considering a specific engine. To start the procedure, a design condition at 105 percent speed was chosen. (Po_2/Po_1) , sfc, shaft power, air flow, and fuel flow quantities were readily available in the literature for the JFTD12A (Table 2.0). Initially, values of 0.98, 0.98 and 0.80 were assumed for shaft mechanical efficiencies, burner efficiency, and compressor efficiency respectively. The chosen values are reasonable for this type of engine using a nine stage axial compressor. Different pressure ratios across the gas generator and power turbine were tried until a satisfactory turbine inlet temperature

and turbine efficiencies were obtained. The detailed calculations are presented in Appendix B and the pertinent results are summarized in the following table.

$T_{o_1} = 288 \text{ K}$	$P_{o_1} = 101.3 \text{ kPa}$	$\eta_b = 0.95$
$T_{o_2} = 549 \text{ K}$	$P_{o_2} = 683.8 \text{ kPa}$	$\eta_c = 0.80$
$T_{o_3} = 1175 \text{ K}$	$P_{o_3} = 656.4 \text{ kPa}$	$\eta_{gst} = 0.86$
$T_{o_4} = 930 \text{ K}$	$P_{o_4} = 215.0 \text{ kPa}$	$\eta_{pt} = 0.85$
$T_{o_5} = 789 \text{ K}$		$\eta_m = 0.98$

$$\dot{m}_a = 23.18 \text{ kg/s}$$

$$P_{wr} = 3581 \text{ kW}$$

INITIAL DESIGN POINT CONDITIONS

The above results are not the final design point, they only served as an initial guess to the running line. As mentioned earlier, when using generalized component characteristics a satisfactory simulation of the actual engine may not be possible over the entire running line. This was the case in this study; as the following paragraphs indicate a new design point had to be selected and the range of the simulated running line reduced.

The above parameters served as the initial inputs to the computer simulation. Scaling factors were applied to the component characteristics and adjusted until a satisfactory component matching was obtained on the 105 percent speed line. Continuing the procedure of adjusting the scaling factors on the other speed lines and using Figures 1, 3, and

5, a running line over the entire operating range was attempted. As a consequence of this effort the following modifications were incorporated.

1. Since the gas generator turbine was expected to remain choked throughout the operating range of interest the reference conditions were adjusted to reflect this situation. Since the non-dimensional mass flow lines collapse under these conditions, operation was assumed to be independent of speed.
2. Due to the lack of information available regarding the power turbine conditions relative to the gas generator, computations were carried out assuming a speed independent turbine characteristic, however the power turbine may operate unchoked.
3. A satisfactory running line such that that curves similar to Figures 1, 3, and 5, and including the initial design condition could not be attained.

Subsequently, the initial design point conditions were later dropped from the range of the running line with prospect of extending to lower speeds from the 100 percent speed line match.

Adjustment of the scaling factors continued until a satisfactory agreement with Figures 1, 3, and 5, was achieved. The same figures as above show the resulting running obtained from the computer simulation. Both the manufacturer's and computed specific fuel consumptions versus shaft power for a propeller load are presented in Figure 13. The agreement of the computed engine performance

with an actual engine is quite good in-light of the assumptions taken and the scaling factors used. A sample and description of data input file for the component matching program and resulting output is contained in Appendix C. The final match at the 100 percent compressor speed line is presented as the design point and the results are summarized in the following table.

$T_{o_1} = 288 \text{ K}$	$P_{o_1} = 101.3 \text{ kPa}$	$\eta_b = 0.94$
$T_{o_2} = 521 \text{ K}$	$P_{o_2} = 614.2 \text{ kPa}$	$\eta_c = 0.85$
$T_{o_3} = 1005 \text{ K}$	$P_{o_3} = 587.0 \text{ kPa}$	$\eta_{gst} = 0.87$
$T_{o_4} = 794 \text{ K}$	$P_{o_4} = 195.0 \text{ kPa}$	$\eta_{pt} = 0.87$
$T_{o_5} = 685 \text{ K}$		$\eta_m = 0.98$
$\dot{m}_a = 22.95 \text{ kg/s}$		
$N_{gg} = 15950 \text{ rpm}$		
$P_{wr} = 2694 \text{ kW}$		

FINAL DESIGN POINT CONDITIONS

The most notable changes between the initial and final design points is the lower TIT and CDT, and increased component efficiencies. The significant decrease in TIT (170 K) is considered reasonable since the usual practice is to derate a modified aircraft gas turbine for marine use. Moreover, the TIT of 1120 K estimated from Figures 1, 3, and 5, agrees quite well with the final design point.

4.9 Simulated Faults

The following fundamental degradations were modeled;

1. Gas generator turbine damage simulated by a 2 percent reduction in the isentropic turbine efficiency with no change in non-dimensional mass flow.
2. Power turbine damage; simulated by a 2 percent isentropic efficiency reduction with no change in non-dimensional mass flow.
3. Compressor fouling resulting in a 7 percent reduction in inlet mass flow and a 2 percent reduction in isentropic efficiency.
4. Gas generator turbine coking causing a 5 percent decrease in non-dimensional mass flow.
5. Gas generator turbine erosion causing a 5 percent increase in non-dimensional mass flow.
6. Power turbine coking resulting in a 5 percent reduction in non-dimensional mass flow.
7. Power turbine erosion resulting in a 5 percent increase in non-dimensional mass flow.

The values used to model the compressor fouling and turbine damage are the same as those used by Saravanamuttoo and MacIsaac [14]. The most correct method of implementing the required parameter changes to simulate the faults is not clear. Three simple approaches can be taken (i) a percentage change (likened to a multiplying factor), (ii) the use of a constant (appropriately added or subtracted), and (iii) a linear variation of the fault over the operating range. All three of these methods were investigated for the case of

compressor fouling. It is understood that the power turbine is much less likely to incur erosion or coking as compared to the gas generator turbine. Regardless, the degradations were investigated with the purpose of evaluating the relative effects on the engine as caused by the same faults when applied to either turbine. Finally, a combined fault of compressor fouling and gas generator turbine erosion was investigated. The results of the study are presented in the following chapter.

RESULTS OF THE ENGINE FAULT SIMULATIONS

5.1 Review of Component Matching Between the Gas Generator and the Power Turbine

Prior to a detailed description of the individual engine faults a review of component matching is appropriate. In practice the variation of turbine efficiency with non-dimensional speed at a fixed pressure ratio is minor. Moreover, the variation has little affect on non-dimensional exit mass flow since the resulting change in $\sqrt{(T_{03}/T_{04})}$ is small. Thus for mean values of turbine efficiency the gas generator exit flow becomes only a function of $(m_3\sqrt{T_{03}/P_{03}})$ and (P_{03}/P_{04}) . Using equation (16) and assuming m_3 equal to m_4 a single curve representing the turbine characteristic is readily obtained (the dashed curve in the following figure).

OPERATION OF TURBINES IN SERIES

Source: reference [13]

As shown by the previous figure the requirement of flow compatibility between the two turbines places a major restriction on the regime of gas generator operation. So long as the power turbine is choked the gas generator turbine will operate at a fixed pressure ratio. With the power turbine unchoked the gas generator turbine will operate at a specific pressure ratio for each power turbine pressure ratio. Due to the restriction of flow compatibility, the gas generator turbine operates over a narrow range. Thus, the gas generator turbine efficiency is nearly constant. From the relationship between the pressure ratios of the two turbines and the fact that the percentage burner pressure loss is nearly constant, the compressor pressure ratio can readily be determined from,

$$(P_{o2}/P_{o1}) = (P_{o2}/P_{o3}) (P_{o3}/P_{o4}) (P_{o4}/P_{o5}) (P_{o5}/P_{amb}) \quad (20)$$

An energy balance within the gas generator and the requirement of continuity of flow between the gas generator and power turbines provides the following relationships,

$$T_{o3} = C_{p12} \Delta T_{o12} (T_{o3}/T_{o1}) T_{o1}/\eta_m/C_{p34}/\Delta T_{o34} \quad (21)$$

$$\sqrt{T_{o3}} = (m_3/\sqrt{T_{o3}}/P_{o3})(P_{o3}/P_{o2})\sqrt{T_{o1}}/(m_1\sqrt{T_{o1}}/P_{o1}) (P_{o2}/P_{o1}) \quad (22)$$

Thus, for any value of CPR, the pressure ratios across the turbines, $m_3/\sqrt{T_{o3}}/P_{o3}$, $\Delta T_{o12}/T_{o1}$, and $\Delta T_{o34}/T_{o3}$ are all determinable. The FT12 gas generator turbine is expected to operate choked, however the power turbine is not expected to be choked throughout the entire engine operating range. The latter assumption is based on the low cycle pressure ratio.

These assumptions were implemented in the previously described engine model. The use of the above engine characteristics will be of value in validating the results of the fault simulations.

5.2 Results

5.2.1 Results of the Compressor Fouling Simulation

Figures 14 (a)-(k) inclusive are the computed performance curves for a fouled compressor. In each case the solid line is the reference or baseline condition as computed initially. Unless otherwise stated, references to the engine parameters such as air flow imply the standard non-dimensional form. Figure 14(a), intake air mass flow versus rotational speed, depicts the three methods of fault modeling investigated. Method 1, (short dashed line) shows the increasing deviation from the baseline brought about by the use of fixed percentage decreases in air flow and compressor efficiency. Both methods 2 and 3 use the results of method 1 to model compressor fouling in a different manner. The highest deviation obtained in the first method was taken as the constant deviation for the second. As expected, the resulting performance curve (long-short dashed line) remains parallel to the baseline curve. Finally, the last method, (long dashed line) used a linear variation in degradation proceeding from a zero value at the lowest rotational speed to the maximum value encountered in method 1 at the highest rotational speed. A zero deviation at the low speed end is probably unrealistic, similarly a constant deviation over the entire operating range is unlikely.

However, without practical evidence to support any of the methods their depiction of the true situation must be considered of limited validity.

The effect of compressor fouling on the engine component matching and subsequent engine performance is not readily apparent. Both the gas generator turbine and power turbine are assumed not to be affected by fouling and consequently their matching is unchanged. The change in compressor efficiency due to fouling will be ignored for the moment, since changes in component efficiencies have a lesser effect than flow or pressure ratio on component matching. The scaling of the compressor map to produce a 7 percent reduction in mass flow can be thought as merely shifting the non-dimensional speed lines. Thus, if a non-fouled compressor line is considered, a reduction in mass flow will require the compressor to operate at a higher rotational speed to maintain the same operating point on the running line. The effect of reduced compressor efficiency for a certain pressure ratio is to raise the required compressor work. Since the gas generator turbine is choked the only way to balance the compressor work is to have an increase in TIT. The increase in TIT will cause a slight non-dimensional flow imbalance with the gas generator turbine which can be balanced with an increase in CPR. The effect of compressor fouling on the running line is shown in 14(1). Consequently a fouled compressor causes the running line to move towards surge and hot section temperatures to increase.

Of additional interest, the performance prior to and after a compressor wash for an ST6 gas turbine at Carleton University is reported in reference [22]. The results indicated that compressor efficiency dropped a constant amount for all gas generator speeds, whereas the deviation in intake mass flow decreased with speed for the fouled compressor. Figures 15 (a)-(d) present the engine performance before and after the compressor wash in terms of compressor mass flow, efficiency, and pressure ratio versus gas generator speed. Figure 15(d) supports the previous statement that the running line moves towards surge. Although the situation described above was not modeled, the effect on CPR is quite similar to the linear variation in mass flow used method 3 for simulating compressor fouling. A description of the results of the other faults is contained in the following sections.

5.2.2 Gas Generator Turbine Faults

Three gas generator turbine faults were simulated; mechanical damage, blade erosion, and blade coking. For either erosion or coking its development will usually require an extensive number of operating hours. Blade coking will cause a reduction in the blade to bladethroat area through the build up of deposits. Turbine mechanical damage was introduced by scaling the isentropic turbine efficiency by 0.98. The effect of coking was introduced by scaling the non-dimensional turbine mass flow by a 5 percent decrease and erosion by a 5 percent increase. The resulting engine performance curves are presented in Figure 16 (a)-(j).

The reduction of spool mechanical efficiency will require additional work from the gas generator turbine. Since the gas generator turbine pressure ratio is essentially constant the TIT must increase to balance the compressor work. The rise in TIT will cause the non-dimensional gas generator turbine inlet flow to increase above its previous matching value. To retain flow compatibility the CPR will have to increase slightly. Therefore, mechanical damage has the effect of moving the running line towards surge and increasing the hot section temperatures.

Gas generator erosion has the effect of increasing the turbine area. The matching of the gas generator turbine and power turbine will result in a lower gas generator turbine pressure ratio for every power turbine pressure ratio. Thus, compressor pressure ratio for each power turbine pressure ratio will be less. Consequently, the running line will move away from surge. Since the compressor efficiency will also be decreasing the compressor work will be nearly constant. However, the lower gas generator turbine pressure ratio will necessitate a TIT increase to balance the compressor work and flow compatibility.

Gas generator turbine coking has the inverse effect of erosion on the engine behaviour. The reduced gas generator turbine area will result in the turbine operating at a higher pressure ratio. The CPR will also rise and the compressor efficiency will change, albeit slightly. The increase in gas generator turbine pressure ratio will allow

the turbine to supply substantially more work for the same TIT. Thus the TIT must decrease such that the balance of work will be satisfied as well as the lower non-dimensional inlet mass flow of the turbine.

In practice the engine control system would limit the fuel flow in the instances where an increase in TIT has occurred. The engine would then operate at a lower TIT and rotational speed. For the coked turbine condition the operator would call for more fuel, increasing the engine's rotational speed and power until the governor overrides to prevent any further fuel addition.

5.2.3 Power Turbine Faults

Three power turbine faults were simulated; mechanical damage, blade erosion and blade coking. Both blade erosion and blade coking may be encountered over an extended number of operating hours with several gas generator module replacements. It is expected that blade coking will cause a reduction in blade to blade throat area through the build up of deposits. Turbine mechanical damage was simulated by scaling the isentropic efficiency by 0.98. Such damage usually relates to the rotor. The effect of erosion was introduced by scaling up the non-dimensional mass flow by 5 percent and coking by a 5 percent reduction. Figure 17 (a)-(j) present the results of the faults on engine performance. Mechanical damage is given by the short dash line, coking by the long dash line, and erosion by the long-short dash line.

Logically, power turbine rotor damage has no effect on the operation of the upstream components. Consequently,

decrease in available power and an increase in EGT is observed. It is notable that the variation in the engine parameters is minor and a substantial increase in mechanical damage would be necessary to distinguish the deterioration from the normal variation in recorded data. The second fault, coking of the power turbine blades has the effect of decreasing the power turbine area. The new value of choking mass flow for the power turbine requires the gas generator turbine to operate at a lower pressure ratio. The lower gas generator turbine pressure ratio will require a higher combustion temperature in order to satisfy the compressor work. However, the increase in TIT will necessitate a slight CPR increase to satisfy the non-dimensional inlet flow to the gas generator turbine. The combination of a higher TIT and higher power turbine pressure ratio results in a higher shaft power. Power turbine coking causes the equilibrium running line to move towards surge.

Erosion of the power turbine has the effect of increasing its area. Thus, flow matching of the gas generator turbine and the power turbine will require a higher pressure ratio across the gas generator turbine. The higher gas generator turbine pressure ratio will allow a reduction in TIT such that the same work is produced by the turbine. The compatibility of flow requirement within the gas generator will cause a reduction in CPR. Therefore, the running line will move away from surge.

In practice, at the higher powers the engine control system will cut back the fuel so that the engine will

operate at a lower rotational speed and temperature. However, the temperatures throughout the hot-section will remain above the normal condition and consequently reduce the engine life. For a coked power turbine the operator will just increase the fuel flow causing the engine to operate at a higher rotational speed. Eventually the engine governor will prevent any further increase in rotational speed and fuel flow. The fault investigation indicates that changes in the free power turbine area have a more substantial affect than an equal percentage change of the gas generator turbine area. Typically, the effective area of the free power turbine is significantly larger than the gas generator turbine. Therefore, to observe the affect of equal area changes a smaller percentage change is required for the free power turbine.

5.2.4 Combined Compressor Fouling and Gas Generator

Turbine Erosion

Clearly, there are many possible combinations of the previous faults simulated. It is unlikely that the faults would develop simultaneously, rather, faults are likely to develop at different rates and be time delayed relative to one another. The situation may become more complex through regular maintenance where some engine degradations may not be detected and others that have initiated are corrected. The combination of compressor fouling and gas generator turbine erosion was implemented in the same manner as the faults individually. Generally, the individual faults do not superimpose, that is, the result obtained by averaging the

effects produced by the individual faults do not equal the effects produced by the combined fault. Also if the deviations from the reference condition are summed and superimposed on the reference condition the combined fault results are still not obtained. However, if the performance curves as produced by the individual faults are compared to those produced by the combined fault the trends still hold. That is, rotations in the curves are propagated, and increases in a parameter caused by one fault tend to cancel with decreases in the same parameter as produced by a different fault. An illustration of a rotation is most easily seen in the trend of EGT (Figure 18 (i)). The compressor fouling produces a rotation plus deviation, and the gas generator erosion causes only a deviation; in the combined fault (Figure 18 (i)) the rotation is still evident. Again for the same parameter, at the highest gas generator speed, the decrease in EGT caused by compressor fouling combined with the larger increase in EGT resulting from the erosion produces a net increase in EGT. Similarly, if both faults produce a change in the same direction from the baseline the combined fault will be in the same direction and larger than any one of the individual faults. The results of the simulation are provided in Figure 18 (a)-(j). In summary, combined faults can only be analyzed properly if they are initiated as a combined fault.

5.3 EHM Parameter Selection Based on the Computer Simulation Results

Figure 19 presents a fault matrix for the high speed range of the engine simulated. The deterioration effects on the engine parameters can be summarized as follows:

1. Only faults involving compressor fouling will experience a decrease in air flow and inlet pressure loss.
2. Temperatures in the hot section of the engine change in the same direction as the fuel flow.
3. With the exception of gas generator turbine coking or erosion, changes in the power turbine rotational speed, and shaft power follow that of fuel flow.
4. With the exception of gas generator turbine coking or erosion, changes in (Po_4/Po_5) follow that of (Po_2/Po_1) .
5. Changes in To_2 follow those of (Po_2/Po_1) since the compressor efficiency is nearly constant at high powers.

Due to the difficulty of maintaining instrumentation and measurement difficulty, TIT is usually not measure in practice. However, both ITT and EGT follow the changes in TIT. Therefore, measuring either one would suffice for EHM with the exception that for the case of power turbine damage, EGT must be measured if a power measurement is unavailable. Usually inlet air flow is not measured, however, any measurement of inlet pressure loss would probably suffice, since any reduction in inlet pressure is a sure indication of compressor fouling. Indubitably, the measurement of fuel flow is an excellent indicator as to the engine state. The fault matrix would seem to indicate that

the measurement of fuel flow would be a sure indicator of engine problems, however if fuel flow is not measured a temperature measurement in the hot section will suffice. The reason for not measuring this parameter in the field is not readily apparent, since there is a wide range of instrumentation available for this purpose, all with excellent reliability. Certainly, fuel flow is always measured on aircraft, which is a much more difficult problem than on a ship. It is suggested that an accurate fuel flow measurement was not provided on the DDH class ships since at the time of construction, fuel was inexpensive and only daily fuel consumptions were of interest. Like TIT, turbine inlet pressure is not measured in practice. Since the combustor pressure loss is nearly constant the gas generator turbine pressure ratio can be deduced from the inlet and exhaust pressure losses, and compressor and power turbine pressure ratios.

In summary, the prime parameters of interest are; gas generator speed, ambient temperature and pressure, inter-turbine temperature, inlet pressure loss, compressor delivery pressure, power turbine inlet and exit pressures (EPR calculation), with the possible addition of fuel flow and exhaust gas temperature (power turbine efficiency and shaft power calculation).

5.4 Isolation of Faults

From the analysis of the full fault matrix some engine degradations are seen to produce similar results, namely, power turbine coking and gas generator damage. If the fault

5.7 An Example of an In-Service Instrumentation Failure

Figures 23-32 inclusive present engine pressure ratio (EPR) versus gas generator speed ($N_{gg}/\sqrt{\theta}$) covering approximately six hundred operating hours. Figure 24 shows a sudden rise in EPR from the previous operating conditions and is complete by the time covered by Figure 25. It is striking that during the same period there is no significant change in compressor pressure ratio or exhaust gas temperature and if CPR is not changing EPR certainly should not be changing. Further investigation of the data indicated that the pressure measurement at the power turbine inlet was defective. This conclusion is also supported by the fault modeling presented earlier. No combination of likely faults could be determined such that only a change in EPR could be produced. Indubitably, the use of EPR as an independent variable in trending would have lead to erroneous conclusions in this case. If EPR had been used for trending it would have appeared that a significant decrease in all the other engine parameters had occurred. The case cited exemplifies the requirement for instrument validation such as comparing the results for an engine parameter based on the predictions from rotational speed and EPR. The use of EPR as an independent variable emanates from the use of trending on the turbojet engine where EPR is closely related to total thrust. Clearly, for shaft power engines the appropriate equivalent variable is power. In practice power is not measured directly, instead torque and rotational speed are measured. Usually, the techniques to measure

response time, and occasionally data recording errors. However, the data scatter may still be too severe for EHM purposes, the problem being that the changes of interest may be of the same magnitude as the scatter. For example, if the bandwidth of the scatter on the EGT (after correction to ISO conditions) is 20 to 25 °C, this same change on the overall performance would be considered significant. As pointed out earlier such a rise in temperature would reduce the hot-section component life substantially.

Each of the Figures 1-6 shown represents approximately two hundred hours of engine operation. It is apparent that the engine does not operate at or near one power condition. The original data indicated that the amount of engine operating time was evenly distributed throughout the gas generator speed range, with the exception of the very low power levels where the amount of engine operating time at these conditions was significantly less than at higher power operation. This is primarily a result of the fact that the engine is intended for cruising and serves for maneuvering (both fore and aft), and low speed operation. From the point of view of trending or gas path analysis the preferred data is obtained at high power conditions. The problem therein lies with the available data, if the technique is restricted to a range of 1000 rev/min, nearly eighty percent of the data is unused. Secondly, the minimum number of data points required to detect a change in engine condition is unknown. If twenty good data points (out of a possible forty) are required out of the restricted speed range, this would be

equivalent to one hundred operating hours or more. Probably, the engine will have experienced a significant amount of damage during this time. Therefore, the use of all (or at least most) of the available data is imperative for EHM purposes. The following section presents the results of simulated trending for engine degradations.

5.6 Data Trending Simulation

The first time user of data trending has little guidance as to the expected changes resulting from engine deterioration. Such information is difficult to provide since it depends on state-of-health of an engine, its environment, and the load characteristics. The ideal situation would be that the engine operator would have access to data for engines (similar to those being used) depicting the effects of various degradations. Unfortunately, data currently available concerns a limited number of engines and usually the engine state indicated is typical of the norm. A trending simulation was carried out with the purpose of trying to provide some insight into the expected fault results. Two cases were considered, compressor fouling and gas generator erosion. These are common engine problems for engines in marine applications.

The engine model indicates compressor fouling is a particularly difficult case since the deteriorated engine performance curves have an offset and rotation relative to the baseline condition. Due to this phenomenon straight trending of the performance deviation from the baseline may not indicate any engine deterioration. This is a result of

the deviation of the deteriorated performance curve from the baseline increasing with speed. Although the average deviation would increase so would the scatter about this average. Consequently, quite a large change in the average deviation may be required to detect an unhealthy engine. The trending simulation was performed only on inter turbine temperature, since as shown previously, this is a prime indicator of engine health. Secondly, trending other parameters would produce an unwieldy number of figures whose benefit may be minimal as far as general conclusions. From Figure 3 the normal data scatter was estimated to be 10°C peak-to-peak about the mean. A computer program was written to produce random gas generator speeds between 11000 and 16000 rev/min. These speeds were input to third order polynomials fitted to the reference and degraded engine conditions as produced by the engine component matching. Finally, a random data scatter was superimposed on the deviation between the baseline and degraded performance curves. Figures 20 and 21 show the fouling as simulated with two different data scatters. The figures show the change of T_{04}/θ from the baseline condition at gas generator speeds, N_{gg}/θ , between 11000 rev/min and 16000 rev/min. The independent variable is an arbitrary time function that could be taken as operating hours accumulated. Each figure follows the pattern as depicted in the following figure.

PARAMETER TRENDING SIMULATION

Region A is the initial baseline condition showing the normal data scatter. Region A covers a time period of fifty units. Region C is the fully degraded engine with no change in the normal data scatter and covers a time period of fifty units. Region B covers the fault implementation which was applied linearly over one hundred time units. There is no evidence to suggest that engine deterioration follows the pattern as previously described (particularly in region C), (a control system which limits temperature to a maximum value may produce a similar affect for a temperature parameter). No matter how unrealistic, region C aids in concentrating on region B. Figure 21 shows an increase in the range of scatter and the average deviation from the baseline condition as predicted. Note, that if the region up to 120 time units is only considered the deterioration is difficult to detect. Yet the amount of degradation has been

increased to 70 percent of the final value. Figure 21 is a repeat of the previous trending with double the data scatter. Again, the final fouled condition is fairly obvious however, the early deterioration is more difficult to detect than the previous trending.

Figure 22 presents the results of gas generator erosion and is classic of other faults. That is, there is negligible change in the nominal data scatter accompanied by an increase (or decrease) in the average deviation. As with the compressor fouling the fault was introduced linearly to a maximum value and offset by fifty time units. For gas generator coking or other faults that produce similar effects trending does not appear to pose any problems. For compressor fouling improvements in trending may be obtained by considering smaller ranges however the problem with using reduced quantities of data is incurred.

As a general guideline, a parameter change that is at least as large as the peak-to-peak nominal data scatter is the minimum required to detect a change in engine condition. Thus, if the nominal data scatter on temperature is 10°C a minimal change of 10°C in the average deviation is required to detect the variation. For smaller degradations than those previously simulated a longer period (and more data) will likely be required to determine that the engine's health has changed.

5.7 An Example of an In-Service Instrumentation Failure

Figures 23-32 inclusive present engine pressure ratio (EPR) versus gas generator speed ($N_{gg}/\sqrt{\theta}$) covering approximately six hundred operating hours. Figure 24 shows a sudden rise in EPR from the previous operating conditions and is complete by the time covered by Figure 25. It is striking that during the same period there is no significant change in compressor pressure ratio or exhaust gas temperature and if CPR is not changing EPR certainly should not be changing. Further investigation of the data indicated that the pressure measurement at the power turbine inlet was defective. This conclusion is also supported by the fault modeling presented earlier. No combination of likely faults could be determined such that only a change in EPR could be produced. Indubitably, the use of EPR as an independent variable in trending would have lead to erroneous conclusions in this case. If EPR had been used for trending it would have appeared that a significant decrease in all the other engine parameters had occurred. The case cited exemplifies the requirement for instrument validation such as comparing the results for an engine parameter based on the predictions from rotational speed and EPR. The use of EPR as an independent variable emanates from the use of trending on the turbojet engine where EPR is closely related to total thrust. Clearly, for shaft power engines the appropriate equivalent variable is power. In practice power is not measured directly, instead torque and rotational speed are measured. Usually, the techniques to measure

torque are best suited to laboratory conditions. For a free power turbine engine there are many possible power turbine speeds for any one gas generator speed. Therefore, although power turbine speed can be measured accurately, gas generator speed is a superior independent variable since for the majority of the other measured parameters it has a unique value. Furthermore, the non-dimensional form of speed only requires the measurement of two parameters (shaft speed and ambient temperature), both can be measured quite accurately. From an operator point of view, speed is a natural parameter to monitor as opposed to pressure ratio, mass flow or temperature. Certainly, for this case, where no other engine deterioration is apparent the use of performance curves to present the collected data is superior to trending.

Finally, Figure 33 is a plot of power turbine speed against gas generator speed obtained from the same source as the previous data. This curve shows a higher amount of data scatter relative to the other performance curves and is likely caused by pitch trims and reversals. The curve appears to be fairly flat for about half the range and then increases linearly. This relationship is typical of the controllable pitch propeller as described earlier. Consequently, power turbine speed as an engine health parameter has little use for the cruise engine. However, for a boost engine, power turbine speed may be useful since the propeller's pitch is nearly fixed. Certainly for a boost engine the propeller's pitch will be fixed at high power

settings where engine degradation is expected to be most evident.

FUTURE DEVELOPMENTS

The use of generalized component characteristics that have been subsequently modified such that the desired engine performance results on matching is a significant step towards modeling degraded engine behaviour. Logically, the preferred route is to employ manufacturer's component characteristics for the analysis of engine degradation, however such information is usually proprietary. Advances in this area may only result if the quantity of in service engine data acquired under engine degraded conditions increases.

To develop a complete engine model, additional information is required with regard to the controllable pitch propeller control system. The details of the control system are more important for the cruise engine than for the boost engine, since at higher vessel speeds the controllable pitch propeller operates at a fixed pitch. The method of using generalized component characteristics can be applied to twin spool engines although to do so requires substantially more information along the gas path compared to the single-spool free turbine engine. Certainly, the investigation of the effects of various faults on a twin spool engine would be of interest.

A statistical analysis of typical engine data is probably warranted, the purpose of the analysis being to establish a minimum deviation in a parameter which is considered to be significant. Such a change may not be any different from what can be determined by merely eye-

2

MOST FREQUENT FAILURE CAUSES FOR 1-5 MW GAS TURBINES (%)

(Electric Power Generation)

Failed Component	Type of Service		
	Base Load	Intermittent	Peaking
Fuel Nozzle	30	5	5
Cracked Liners	5	11	15
Transition Piece	5	5	10
1st Stage Turbine Nozzle	15	20	20
2nd & 3rd Stage Turbine Nozzle	5	10	9
Turbine Blade	20	20	20
Turbine Disc	3	5	6
Compressor Disc	2	2	2
Compressor Diffuser	3	4	4
Radial Bearings	5	4	4
Thrust Bearings	2	5	
Seals	4	3	3
Couplings		2	2
Foreign Object Damage		2	
Lubrication System Failure	1	2	

Source: Reference [3]

engine pressure ratio, and intake and exhaust duct losses are necessary for gas path analysis. A simulation of trending was carried out which showed that only changes in engine performance equal to or greater than the normal data scatter can be recognized readily.

Actual in service data from a marine FT12[®] gas turbine indicated that the data is well behaved over a long number of operating hours for a healthy engine. The collected engine data may include rogue points, nevertheless such points are easily detected and thus can be eliminated from any further analysis. The data analysis will require the selection of an independent variable. This variable should be one that can be measured accurately and reliably such as shaft speed. Gas generator speed is a superior independent variable for engine health monitoring as opposed to Engine Pressure Ratio. The best method of presenting the engine data is not clear. Each method has its drawbacks, however the use of performance curves is a satisfactory technique for health monitoring purposes. If trending of the deviation of a parameter from a baseline is being performed some form of validation is highly recommended. Finally, it is imperative that people involved in the collection and analysis of engine data fully appreciate the importance of their tasks and the techniques involved in order to ensure the success of the EHM program.

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MOST FREQUENT FAILURE CAUSES FOR 1-5 MW GAS TURBINES (%)

(Electric Power Generation)

Failed Component	Type of Service		
	Base Load	Intermittent	Peaking
Fuel Nozzle	30	5	5
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2nd & 3rd Stage Turbine Nozzle	5	10	9
Turbine Blade	20	20	20
Turbine Disc	3	5	6
Compressor Disc	2	2	2
Compressor Diffuser	3	4	4
Radial Bearings	5	4	4
Thrust Bearings	2	5	
Seals	4	3	3
Couplings		2	2
Foreign Object Damage		2	
Lubrication System Failure	1	2	

Source: Reference [3]

JFTD12A-5A ENGINE SPECIFICATIONS

Model and type ..JFTD12A-5A. Turboshaft, 2-shaft, 9-stage axial compressor², cannular combustor, 2-stage gas generator turbine, 2-stage free power turbine.

IntakeHot bleed air anti-icing system.

CompressorFront section contains 1 row of inlet vanes and 4 rows of stator blades. Rear section has 5 rows of stator blades. Disc-type rotor with 9 rows of blades connected to drive shaft from 1st and 2nd-stage turbine wheels. Automatic air bleed ports for starting. Pressure ratio 6.75:1, and air mass flow 23 kg/s at 16,750 rpm.

Combustor8 burner cans with duplex fuel burners.

TurbineAir-cooled nozzle guide vanes. 2 turbine wheels (gas generator) mounted on drive shaft to compressor.

Turbine3rd and 4th-stage free turbine wheels in (power) housing attached to gas generator housing, with wheels mounted on rearward drive shaft extending through exhaust duct.

Control system ..Hydro-mechanical. Automatic speed control for starting, acceleration, steady, deceleration. Free turbine limiter.

Fuel consumption0.4197 kg/kw/hr

Rating (take-off) ...3,579 kw at 16,750 rpm static

Rating (max. cont.) .3,303 kw at 16,450 rpm static

Source: Reference [23]

FT12 MARINE ENGINE SPECIFICATIONS

	Normal	Max. Cont.	Max. Int.
Free Turbine Power (kw)	1864	2386	2811
Free Turbine rpm	9000	9000	9000
Estimated sfc (kg/kw/hr)*	0.476	0.436	0.421
Estimated EGT (K)	704	733	764
Estimated Exhaust Mass Flow (kg/s)	20.5	21.7	22.5

* based on a LHV of 42942 kJ/kg at an 26.7 C inlet temperature

Source: P & W Brochure

Fig. 1

EPR versus N_{gg} for a FT12 Engine (Sept 1980)

Fig. 2

EPR versus N_{gg} for a FT12 Engine (Nov 1981)

Fig. 3

EGT versus N_{gg} for a FT12 Engine (Sept 1980)

Fig. 4

EGT versus Ngg for a FT12 Engine (Nov 1981)

Fig. 5

CPR versus N_{gg} for a FT12 Engine (Sept 1980)

Fig. 6

CPR versus Ngg for a FT12 Engine (Nov 1981)

Fig. 7

Fig. 8

Source: Reference [15]

THREE METHODS OF MODELING COMPRESSOR FOULING AND THE RESULTING PERFORMANCE CURVES EPR versus Gas Generator Speed

Fig. 14(f)

Ngg (rpm x 10⁻³)

Fig. 10

GENERALIZED TURBINE MAP

Fig. 11

Source: Reference [15]

Fig. 13

FT112 sfc vs Power ($T_{amb}=26.7\text{ C}$)

POWER (kW)

THREE METHODS OF MODELING COMPRESSOR FOULING AND THE RESULTING PERFORMANCE CURVES
Inlet Air Flow versus Gas Generator Speed

Fig. 14(a)

8.27 THREE METHODS OF MODELING COMPRESSOR FOULING AND THE RESULTING PERFORMANCE CURVES

Inlet Pressure loss versus Gas Generator Speed

Fig. 14(b)

THREE METHODS OF MODELING COMPRESSOR FOULING AND THE RESULTING PERFORMANCE CURVES Fuel Flow versus Gas Generator Speed

$\text{Ngg} (\text{rpm} \times 10^{-3})$

Fig. 14(c)

Fig. 14(d)

THREE METHODS OF MODELING COMPRESSOR FOULING AND THE RESULTING PERFORMANCE CURVES
CPR versus Gas Generator Speed

THREE METHODS OF MODELING COMPRESSOR FOULING AND THE RESULTING PERFORMANCE CURVES

CTPR versus Gas Generator Speed

NGG (rpm x 10⁻³)

Fig. 14(e)

Fig. 14(f)

THREE METHODS OF MODELING COMPRESSOR FOULING AND THE RESULTING PERFORMANCE CURVES
EPR versus Gas Generator Speed

Fig. 14(g)

THREE METHODS OF MODELING COMPRESSOR FOULING AND THE RESULTING PERFORMANCE CURVES
CDT versus Gas Generator Speed

Fig. 14(h)

THREE METHODS OF MODELING COMPRESSOR FOULING AND THE RESULTING PERFORMANCE CURVES

TIT versus Gas Generator Speed

Fig. 14(1)

THREE METHODS OF MODELING COMPRESSOR FOULING AND THE RESULTING PERFORMANCE CURVES
ITT versus Gas Generator Speed

ITT (K)

THREE METHODS OF MODELING COMPRESSOR FOULING AND THE RESULTING PERFORMANCE CURVES
EGT versus Gas Generator Speed

THREE METHODS OF MODELING COMPRESSOR FOULING AND THE RESULTING PERFORMANCE CURVES

Power versus Gas Generator Speed

Ngg (rpm x 10⁻³)

Fig. 14(k)

Fig. 14(1)

THREE METHODS OF MODELING COMPRESSOR FOULING AND THE RESULTING PERFORMANCE CURVES Effect of Fouling on the Running Line

CORRECTED AIR FLOWRATE VS CORRECTED GAS GENERATOR SPEED

--- Before Wash

— After Wash

$$\frac{M\sqrt{\theta}}{\delta}$$

lbm/sec

$$N_G/\sqrt{\theta} \text{ (RPM } \times 10^{-3} \text{)}$$

Source: Reference [22]

Fig. 15(a)

Fig. 15(b)

COMPRESSOR EFFICIENCY VS CORRECTED GAS GENERATOR SPEED

-- * -- * Before Wash

—○— After Wash

$N_g/\sqrt{\theta}$ (RPM $\times 10^{-3}$)

Source: Ref. 100

COMPRESSOR PRESSURE RATIO VS CORRECTED GAS GENERATOR SPEED

- - - Before Wash
 — After Wash

$$\frac{P_2}{P_1}$$

Fig. 15(c)

$$N_g / \sqrt{\theta} \times 10^{-3}$$

Source: Reference [22]

Fig. 15(d)

Source: Reference [22]

24. PERFORMANCE CURVES FOR GAS GENERATOR TURBINE EROSION, COKING AND MECHANICAL DAMAGE
 Inlet Air Flow versus Gas Generator Speed

$N_{gg} \text{ (rpm} \times 10^{-3}\text{)}$

Fig. 16(a)

Fig. 16(b)

PERFORMANCE CURVES FOR GAS GENERATOR TURBINE EROSION, COKING AND MECHANICAL DAMAGE
 Fuel Flow versus Gas Generator Speed

Fig. 16(c)

PERFORMANCE CURVES FOR GAS GENERATOR TURBINE EROSION, COKING AND MECHANICAL DAMAGE

CPR versus Gas Generator Speed

$N_{gg} \text{ (rpm} \times 10^{-3}\text{)}$

Fig. 16(d)

PERFORMANCE CURVES FOR GAS GENERATOR TURBINE EROSION, COKING AND MECHANICAL DAMAGE

N_{gg} ($\text{rpm} \times 10^{-3}$)

Fig. 16(h)

PERFORMANCE CURVES FOR GAS GENERATOR TURBINE EROSION, COKING AND MECHANICAL DAMAGE
ITT versus Gas Generator Speed

$\text{Ngg} (\text{rpm} \times 10^{-3})$

540. PERFORMANCE CURVES FOR GAS GENERATOR TURBINE EROSION, COKING AND MECHANICAL DAMAGE

CDT versus Gas Generator Speed

Fig. 16(f)

NGG (rpm x 10⁻³)

PERFORMANCE CURVES FOR GAS GENERATOR TURBINE EROSION, COKING AND MECHANICAL DAMAGE

TIT versus Gas Generator Speed

Fig. 16(g)

Ngg (rpm $\times 10^{-3}$)

Fig. 16(h)

PERFORMANCE CURVES FOR GAS GENERATOR TURBINE EROSION, COKING AND MECHANICAL DAMAGE

ITT versus Gas Generator Speed

$N_{gg} (\text{rpm} \times 10^{-3})$

ITT (K)

PERFORMANCE CURVES FOR GAS GENERATOR TURBINE EROSION, COKING AND MECHANICAL DAMAGE
EGT versus Gas Generator Speed

$N_{gg} (\text{rpm} \times 10^{-3})$

PERFORMANCE CURVES FOR GAS GENERATOR TURBINE EROSION, COKING AND MECHANICAL DAMAGE

Power versus Gas Generator Speed

Fig. 16(j)

Ngg (rpm x 10⁻³)

PERFORMANCE CURVES FOR POWER TURBINE EROSION, COKING AND MECHANICAL DAMAGE

Inlet Air Flow versus Gas Generator Speed

Ngg (rpm x 10⁻³)

Fig. 17(a)

Fig. 17(b)

PERFORMANCE CURVES FOR POWER TURBINE EROSION, COKING AND MECHANICAL DAMAGE

Fuel Flow versus Gas Generator Speed

$N_{gg} (rpm \times 10^{-3})$

PERFORMANCE CURVES FOR POWER TURBINE EROSION, COKING AND MECHANICAL DAMAGE

Fig. 17(c)

$N_{gg} \text{ (rpm} \times 10^{-3}\text{)}$

Fig. 17(d)

PERFORMANCE CURVES FOR POWER TURBINE EROSION, COKING AND MECHANICAL DAMAGE

PERFORMANCE CURVES FOR POWER TURBINE EROSION, COKING AND MECHANICAL DAMAGE

EPR versus Gas Generator Speed

Fig. 17(e)

N_{gg} ($\text{rpm} \times 10^{-3}$)

PERFORMANCE CURVES FOR POWER TURBINE EROSION, COKING AND MECHANICAL DAMAGE

CDT versus Gas Generator Speed

Fig. 17(f)

PERFORMANCE CURVES FOR POWER TURBINE EROSION, COKING AND MECHANICAL DAMAGE

Fig. 17(g)

Fig. 17(h)

PERFORMANCE CURVES FOR POWER TURBINE EROSION, COKING AND MECHANICAL DAMAGE

ITT versus Gas Generator Speed

PERFORMANCE CURVES FOR POWER TURBINE EROSION, COKING AND MECHANICAL DAMAGE

EGT versus Gas Generator Speed

$\text{Ngg} (\text{rpm} \times 10^{-3})$

Fig. 17(i)

PERFORMANCE CURVES FOR POWER TURBINE EROSION, COKING AND MECHANICAL DAMAGE

Power versus Gas Generator Speed

Ngg (rpm x 10⁻³)

Fig. 17(j)

FIG. 13(a)

PERFORMANCE CURVES FOR COMBINED COMPRESSOR FOULING AND GAS GENERATOR TURBINE EROSION

Inlet Air Flow versus Gas Generator Speed

Ngg (rpm x 10⁻³)

PERFORMANCE CURVES FOR COMBINED COMPRESSOR FOULING AND GAS GENERATOR TURBINE EROSION

Fuel Flow versus Gas Generator Speed

$\text{NGG} (\text{rpm} \times 10^{-3})$

Fig. 18(b)

Fig. 18(c)

PERFORMANCE CURVES FOR COMBINED COMPRESSOR FOULING AND GAS GENERATOR TURBINE EROSION

CPR versus Gas Generator Speed

N_{gg} (rpm × 10⁻³)

PERFORMANCE CURVES FOR COMBINED COMPRESSOR FOULING AND GAS GENERATOR TURBINE EROSION

CTPR versus Gas Generator Speed

$N_{gg} (\text{rpm} \times 10^{-3})$

Fig. 18(d)

PERFORMANCE CURVES FOR COMBINED COMPRESSOR FOULING AND GAS GENERATOR TURBINE EROSION

EPR versus Gas Generator Speed

N_{gg} (rpm × 10⁻³)

Fig. 18(e)

Fig. 18(f)

PERFORMANCE CURVES FOR COMBINED COMPRESSOR FOULING AND GAS GENERATOR TURBINE EROSION

PERFORMANCE CURVES FOR COMBINED COMPRESSOR FOULING AND GAS GENERATOR TURBINE EROSION
TIT versus Gas Generator Speed

$N_{gg} (\text{rpm} \times 10^{-3})$

Fig. 18(g)

Fig. 18(h)

PERFORMANCE CURVES FOR COMBINED COMPRESSOR FOULING AND GAS GENERATOR TURBINE EROSION

ITT versus Gas Generator Speed

$\text{Ngg} (\text{rpm} \times 10^{-3})$

PERFORMANCE CURVES FOR COMBINED COMPRESSOR FOULING AND GAS GENERATOR TURBINE EROSION

EGT versus Gas Generator Speed

Fig. 18(1)

$N_{gg} \text{ (rpm} \times 10^{-3}\text{)}$

Fig. 18(j)

PERFORMANCE CURVES FOR COMBINED COMPRESSOR FOULING AND GAS GENERATOR TURBINE EROSION

Power versus Gas Generator Speed

Fault Matrix for the Basic Engine Faults Modeled

$R_{Ld} = 100\%$

	ma	m _f	IPL	CPR	CTPK	ELR	CDT	TIT	ITT	IGT	Power
Compressor Fouling	↓	↓	↓	↓	—	↓	↓	↓	↓	↓	↓
Gas Generator Turbine Erosion	—	↓	—	↓	↓	↓	↓	↓	↓	↓	↓
Gas Generator Turbine Coking	—	↓	—	↓	↓	↓	↓	↓	↓	↓	↓
Gas Generator Turbine Damage	—	↓	—	↓	↓	↓	↓	↓	↓	↓	↓
Power Turbine Erosion	—	↓	—	↓	↓	↓	↓	↓	↓	↓	↓
Power Turbine Coking	—	↓	—	↓	↓	↓	↓	↓	↓	↓	↓
Power Turbine Damage	—	—	—	—	—	—	—	—	—	—	—
Combined Fault	↓	↓	↓	↓	↓	↓	↓	↓	↓	↓	↓

DATA TRENDING SIMULATION OF COMPRESSOR FOULING

Fig. 20

DATA TRENDING SIMULATION OF COMPRESSOR FOULING WITH INCREASED DATA SCATTER

FIG. 21

DATA TRENDING SIMULATION OF GAS GENERATOR TURBINE EROSION

Fig. 23

EPR versus Ngg for another FT12 Engine (Sept 1980)

EPR versus N_{gg} for another FT12 Engine (Oct 1980)

Fig. 25

EPR versus N_{gg} for another FT12 Engine (Jan 1981)

Fig. 26

EGT versus Ngg for another FT12 Engine (Sept 1980)

Fig. 27

EGT versus N_{gg} for another FT12 Engine (Oct 1980)

Fig. 28

EGT versus N_{gg} for another FT12 Engine (Jan 1981)

Fig. 29

CPR versus Ngg for another FT12 Engine (Sept 1980)

Fig. 30

CPR versus Ngg for another FT12 Engine (Oct 1980)

Fig. 31

CPR versus Ngg for another FT12 Engine (Jan 1981)

MATHEMATICAL EQUATIONS DESCRIBING ENGINE COMPONENTS

Inlet losses

$$Po_1 - Po_2 = A0 + A1*ma + A2*ma^2 + A3*ma^3 \quad (\text{kPa})$$

where the values of the coefficients are;

$$A0 = 1.965151$$

$$A1 = -0.2798635$$

$$A2 = 0.1568426*10^{-1}$$

$$A3 = -0.2202442*10^{-3}$$

Generalised Compressor Characteristics

(Refer to Figure 9)

For any non-dimensional speed line the equations are of the general form;

$$1. \frac{(m_1 \sqrt{To_1 / Po_1})}{(m_1 \sqrt{To_1 / Po_1})_{ref}} = A0(N/\sqrt{\theta}) + A1(N/\sqrt{\theta}) * K + A2(N/\sqrt{\theta}) * K^2 + A3(N/\sqrt{\theta}) * K^3$$

$$2. \eta / \eta_{ref} = B0(N/\sqrt{\theta}) + B1(N/\sqrt{\theta}) * K + B2(N/\sqrt{\theta}) * K^2 + B3(N/\sqrt{\theta}) * K^3$$

$$\text{where } K = \frac{(CPR-1)}{(CPR-1)_{ref}}$$

For $N/\sqrt{\theta} = 0.7$

$$A0 = 0.5487640$$

$$B0 = 0.9216795$$

$$A1 = -0.8936507*10^{-1}$$

$$B1 = -3.519248$$

$$A2 = 0.2508529$$

$$B2 = 2.504212*10^{-1}$$

$$A3 = -3.042285$$

$$B3 = -4.537829*10^{-1}$$

For $N/\sqrt{\theta} = 0.8$

$$A0 = 0.7248287$$

$$A1 = -0.4812978$$

$$A2 = 1.909652$$

$$A3 = -2.793296$$

$$B0 = 0.6822179$$

$$B1 = -0.2593836$$

$$B2 = 5.065521$$

$$B3 = 0.8212690 \times 10^{-1}$$

For $N/\sqrt{\theta} = 0.9$

$$A0 = 0.9398656$$

$$A1 = -0.6542046$$

$$A2 = 1.561598$$

$$A3 = -1.281060$$

$$B0 = 0.2599827$$

$$B1 = 2.098693$$

$$B2 = -1.536920$$

$$B3 = 0.8212690$$

For $N/\sqrt{\theta} = 0.95$

$$A0 = 1.163797$$

$$A1 = -1.320362$$

$$A2 = 2.384240$$

$$A3 = -1.413468$$

$$B0 = 1.534540$$

$$B1 = -4.727178$$

$$B2 = 9.413535$$

$$B3 = -5.279266$$

For $N/\sqrt{\theta} = 1.0$

$$A0 = 1.376208$$

$$A1 = -1.666216$$

$$A2 = 2.391709$$

$$A3 = -1.109940$$

$$B0 = -0.3838063$$

$$B1 = 3.989964$$

$$B2 = -3.851028$$

$$B3 = 1.264317$$

For $N/\sqrt{\theta} = 1.05$

$$A0 = 1.035463$$

$$A1 = 0.4991890 \times 10^{-1}$$

$$A2 = 0.3769934 \times 10^{-1}$$

$$B0 = -0.6103197$$

$$B1 = 4.163795$$

$$B2 = -3.659106$$

$$A3 = -0.7235995 \times 10^{-1}$$

$$B3 = 1.061413$$

Combustor Pressure Loss and Efficiency

$$1. \frac{Po_2 - Po_3}{Po_2} = 0.064038 * \frac{m_2 / To_2}{Po_2} + 0.001892 * \frac{To_3}{To_2} - 1$$

$$2. \eta_c = 1.05239 - 2.67865 * (Po_2)^{-2}$$

Generalised Turbine Characteristics

(Refer to Figure 11)

For any non-dimensional speed line the equations are of the following form;

$$1. \frac{(m_3 / To_3 / Po_3)}{(m_3 / To_3 / Po_3)_{ref}} = A0(N/\sqrt{\theta}) + A1(N/\sqrt{\theta}) * K + A2(N/\sqrt{\theta}) * K^2 + A3(N/\sqrt{\theta}) * K^3$$

$$2. \eta / \eta_{ref} = B0 + B1(N/\sqrt{\theta}) * K + B2(N/\sqrt{\theta}) * K^2 + B3(N/\sqrt{\theta}) * K^3$$

$$\text{where } k = \frac{(Po_3 / Po_4)}{(Po_3 / Po_4)_{ref}}$$

For $N/\sqrt{\theta} = 0.4$

$$A0 = -3.811603$$

$$B0 = 1.929788$$

$$A1 = 1.574581 \times 10^{-1}$$

$$B1 = -2.073717$$

$$A2 = -1.711266$$

$$B2 = 0.617205$$

$$A3 = 6.175344$$

$$B3 = 0.1514428$$

For $N/\sqrt{\theta} = 0.6$

$$B0 = -3.717332$$

$$B1 = 1.738859$$

$$B2 = -2.083325 \times 10^{-1}$$

$$B3 = 7.994243$$

$$\text{For } N/\sqrt{\theta} = 0.8$$

$$B0 = -5.666920$$

$$B1 = 2.294447$$

$$B2 = -2.595043 \times 10^{-1}$$

$$B3 = 9.631515$$

$$\text{For } N/\sqrt{\theta} = 1.0$$

$$A0 = -2.396513$$

$$B0 = -8.193138$$

$$A1 = 1.003396 \times 10^{-1}$$

$$B1 = 3.145051 \times 10^{-1}$$

$$A2 = -9.957943$$

$$B2 = -3.584433 \times 10^{-1}$$

$$A3 = 3.320434$$

$$B3 = 1.359551 \times 10^{-1}$$

INITIAL DESIGN POINT CALCULATIONS FOR FT12

Given: CPR = 6.75

$$m_a = 23.182 \text{ kg/s}$$

$$P_{wr} = 3581 \text{ kW}$$

$$sfc = 0.4206 \text{ kg/kW/hr}$$

Thus: $m_f = sfc * P_{wr} = 0.4184 \text{ kg/s}$

Assumed: $\eta_c = 0.80$

$$\eta_{m1} = 0.98 = \eta_{m2}$$

$$\eta_{ggt} = 0.86$$

$$b = 0.04$$

$$P_{o3}/P_{o2} = 0.96$$

$$T_{o1} = 288 \text{ K}$$

Considering isentropic compression over the compressor gives;

$$\Delta T_{o12} = 1/\eta_c T_{o1} [(CPR)^{0.2871} - 1]$$

$$\Delta T_{o12} = 261.2 \text{ K}$$

From conservation of energy within the gas generator;

$$m_a * C_{p12} * \Delta T_{o12} = \eta_{m1} * m_a * (1 - b) * (1 + f/a) * C_{p34} * \Delta T_{o34}$$

$$1.021 * 261.1 = 0.98 * (1 - 0.04) * (1 + 0.01805) * 1.1358 * \Delta T_{o34}$$

$$\Delta T_{o34} = 245.1 \text{ K}$$

Shaft power is given by;

$$P_{wr} = \eta_{m2} * m_a * (1 + f/a) * C_{p45} * \Delta T_{o45}$$

$$\Delta T_{o45} = 3581 / [23.182 * (1 + 0.01805) * 1.102]$$

$$\Delta T_{o45} = 140.5 \text{ K}$$

If $T_{o3} = 1175 \text{ K}$

$$\Delta T_{34} = T_{o3} * \eta_{ggt} * [1 - (P_{o4}/P_{o3})^{(0.2509)}]$$

Rearranging, and substituting actual values gives;

$$P_{o3}/P_{o4} = 3.0527$$

$$\begin{aligned} P_{o4}/P_{o5} &= (CPR) * (P_{o3}/P_{o2}) * (P_{o4}/P_{o3}) * (P_{o5}/P_{o1}) \\ &= (6.75) * (0.96) * (1/3.0527) * (1.0) \\ &= 2.123 \end{aligned}$$

Calculating power turbine isentropic efficiency;

$$\begin{aligned} \Delta T_{45} &= T_{o4} * \eta_{pt} * [1 - (P_{o5}/P_{o4})^{(0.2604)}] \\ \eta_{pt} &= 0.849 \end{aligned}$$

With a combustor inlet temperature of 549.2 K and a temperature rise of 625.8 K the ideal (f/a) is 0.01715.

Thus,

$$\eta_b = (0.01715)/(0.01805) = 0.95$$

These conditions were taken to be at 105% gas generator speed. Consequently,

$$\frac{m_1 \sqrt{T_{o1}}}{P_{o1}} = \frac{23.182 \sqrt{288}}{101.3} = 3.884$$

$$\frac{m_3 \sqrt{T_{o3}}}{P_{o3}} = \frac{23.182 (1 + 0.01805 - 0.04) \sqrt{1175}}{6.48 * (101.3)} = 1.1839$$

$$\frac{m_4 \sqrt{T_{o4}}}{P_{o4}} = \frac{22.182 (1 + 0.01805) \sqrt{929.9}}{2.12 * (101.3)} = 3.3511$$

$$\frac{N}{\sqrt{T_{o3}}} = \frac{16750}{\sqrt{1175}} = 488.65$$

COMBUSTION CHAMBER CALCULATIONS

The following shows the development of final equations used to model the FT12 combustor. From the previous design point calculations the following engine parameters were determined.

$$\dot{m}_a = 23.182 \text{ kg/s}$$

$$b = 0.04$$

$$T_{o_2} = 549.2 \text{ K}$$

$$T_{o_3} = 1175.0 \text{ K}$$

$$P_{o_2} = 683.8 \text{ kPa}$$

$$\text{burner pressure loss} = 4\% \text{ (of } P_{o_2} \text{)}$$

$$\text{burner efficiency} = 95\%$$

Combustor Pressure Loss

The total burner pressure loss is composed of the cold and hot losses. The required equations to be considered are 2 and 5 of Section 4.2. Repeated below,

$$\frac{\Delta P_{23\text{cold}}}{P_{o_2}} = \frac{\Delta P_{23}}{q_{\text{ref}}} \frac{R}{2} \left[\frac{\dot{m}_2 \sqrt{T_{o_2}}}{A_{\text{ref}} P_{o_2}} \right]^2$$

$$\frac{\Delta P_{23\text{hot}}}{q_{\text{ref}}} = K_1 \frac{T_{o_3}}{T_{o_2}} - K_2$$

For tubular combustors Cohen et al [13] suggest a value of 35 for $\Delta P_{23\text{cold}}/q_{\text{ref}}$. Initially, assume the entire combustor pressure loss to be due to the cold loss. Since the diffuser and combustor are of a fixed geometry, q_{ref} and P_{o_2} will be related. Therefore q_{ref} in equation 5 can be replaced by P_{o_2} . Taking K_2 to be equal to 1.0 and substituting into the above

equation gives

$$\frac{\Delta P_{23\text{hot}}}{P_{O_2}} = K_1 \frac{1175. - 1.0}{549.2}$$

Lefebvre [16] suggests for a moderate pressure rise and a cold loss of 7% the hot loss is approximately 0.5% of P_{O_2} . scaling the hot loss for a cold loss of approximately 4% gives

$$0.002745 = K_1 (1.13475)$$

Thus,

$$K_1 = 0.001892$$

The cold loss is now 3.736%, substituting into equation 2 gives;

$$0.03736 = (35)(287/2/1000.) \frac{23.182*(0.96) \sqrt{549.2}}{(6.75)*101.3}^2 (1/A_{\text{ref}})^2$$

Thus,

$$A_{\text{ref}}^2 = 78.43$$

Equation 2 takes the final form of

$$\frac{\Delta P_{23\text{cold}}}{P_{O_2}} = 0.064038 \frac{m_2 \sqrt{T_{O_2}}}{P_{O_2}}^2$$

Finally,

$$\frac{\Delta P_{23}}{P_{O_2}} = 0.064038 \frac{m_2 \sqrt{T_{O_2}}}{P_{O_2}}^2 + 0.001892 \frac{T_{O_3}}{T_{O_2}} - 1.0$$

Combustor Efficiency

From Section 4.2 the combustion efficiency is given by equation 9.

$$\eta_b = f \frac{P_{O_2} A_{ref}}{m_2 \sqrt{T_{O_2}}} \frac{\Delta P_L}{P_{O_2}}^{0.5}$$

or

$$\eta_b = C - \frac{P_{O_2} A_{ref}}{m_2 \sqrt{T_{O_2}}} \frac{\Delta P_L}{P_{O_2}}^{0.5}$$

In the above equation the unknowns C, ΔP_L have to be determined. The design point conditions will solve for one equation and the assumption of a combustion efficiency of 90% at a CPR of 3.05 will serve for another equation.

Therefore,

$$0.95 = C - [\Delta P_L / (6.75 * 101.3)]^{0.5} / 0.009725$$

$$0.90 = C - [\Delta P_L / (3.05 * 101.3)]^{0.5} / 0.009725$$

Solving the above two equations gives,

$$C = 1.05239$$

$$\Delta P_L = 0.00687$$

Consequently,

$$\eta_b = 1.05239 - 2.67865 / (1/P_{O_2})$$

The following is a documentation of the digital computer program used to calculate the results presented in this thesis. To augment the computer code listing is a program flow chart and a discription of the variables used in the program. Figures 9, 11, and 12 serve as references for the program. A complete program listing is provided and the Appendix is terminated with sample inputs to the program and the computed results.

PROGRAM SYMBOLS TABLE

C-3

VARIABLE	DESCRIPTION
BPL	Burner Pressure Loss
CNRF	Compressor Reference Speed
CP	Specific Heat at Stations (1-5)
CPR	Compressor Pressure Ratio
CPRG	Guessed Compressor Pressure Ratio
CPRRF	Reference Compressor Pressure Ratio
CPXY	Average Specific Heat Between Stations X and Y
DT12T1	$((T_{02}-T_{01})/T_{01})$
DT34T3	$((T_{03}-T_{04})/T_{03})$
ETB	Burner Efficiency
ETC	Compressor Efficiency
ETCRF	Reference Compressor Efficiency
ETGT	Gas Generator Turbine Efficiency
ETGTRF	Reference Gas Generator Turbine Efficiency
ETPT	Power Turbine Efficiency
ETPTRF	Reference Power Turbine Efficiency
FA	Fuel to Air Ratio
FILE	Data Filename Variable
G	Initial Guesses for Compressor Pressure Ratio
GTPR	Gas Generator Turbine Pressure Ratio
GTPRG	Guessed Gas Generator Turbine Pressure Ratio
GTPRRF	Reference Gas Generator Turbine Pressure Ratio
GTNRF	Reference Gas Generator Turbine Speed
HA2	Enthalpy at Station 2
IPL	Inlet Pressure Loss
KCPR	Scaling Factor for Compressor Pressure Ratio
KETC	Scaling Factor for Compressor Efficiency

PROGRAM SYMBOLS TABLE

C-3

VARIABLE	DESCRIPTION
BPL	Burner Pressure Loss
CNRF	Compressor Reference Speed
CP	Specific Heat at Stations (1-5)
CPR	Compressor Pressure Ratio
CPRG	Guessed Compressor Pressure Ratio
CPRRF	Reference Compressor Pressure Ratio
CPXY	Average Specific Heat Between Stations X and Y
DT12T1	$((T_{02}-T_{01})/T_{01})$
DT34T3	$((T_{03}-T_{04})/T_{03})$
ETB	Burner Efficiency
ETC	Compressor Efficiency
ETCRF	Reference Compressor Efficiency
ETGT	Gas Generator Turbine Efficiency
ETGTRF	Reference Gas Generator Turbine Efficiency
ETPT	Power Turbine Efficiency
ETPTRF	Reference Power Turbine Efficiency
FA	Fuel to Air Ratio
FILE	Data Filename Variable
G	Initial Guesses for Compressor Pressure Ratio
GTPR	Gas Generator Turbine Pressure Ratio
GTPRG	Guessed Gas Generator Turbine Pressure Ratio
GTPRRF	Reference Gas Generator Turbine Pressure Ratio
GTNRF	Reference Gas Generator Turbine Speed
HA2	Enthalpy at Station 2
IPL	Inlet Pressure Loss
KCPR	Scaling Factor for Compressor Pressure Ratio
KETC	Scaling Factor for Compressor Efficiency

KM1	Scaling Factor for Compressor Flow
M	Mass Flow at a Station
MA	Absolute Mass Flow
MB2	Seal Bleed at Station 2
MB3,MB4	Casing Bleeds at Stations 3 and 4
MF	Fuel Mass Flow
M1EFF,M2EFF	Shaft Mechanical Efficiencies
M1RF	Compressor Reference Mass Flow
M3RF,M4RF	Turbine Reference Mass Flows
M4F	Power Turbine Flow Calculated from Continuity
M4W	Power Turbine Flow Calculated from Pressure Ratio
M4R	$(M4F/M4W)$
NGG	Gas Generator Speed
NS	Number of Compressor Speed Lines
PAMB	Ambient Pressure
PO	Total Pressure at Stations (1-5)
PTN	Power Turbine Speed
PTNRF	Reference Power Turbine Speed
PTPR	Power Turbine Pressure Ratio
PTPRRF	Reference Power Turbine Pressure Ratio
PWR	Shaft Power
SFC	Specific Fuel Consumption
TAMB	Ambient Temperature
TITLE	Problem Title
TO	Total Temperature at Stations (1-5)
T03	Combustion Temperature
T3R	$(T3T1W/T3T1F)$

T3T1F (T03/T01) Calculated from Continuity

T3T1W (T03/T01) Calculated from Work

PROGRAM LISTING

C*****

C

C This program performs component matching for a single spool
C free power turbine engine. The compressor and turbines are
C characterized by general component maps which must be scaled
C to simulate a specific engine.

C

C*****

C

```

IMPLICIT INTEGER*1 (I,J,L)
IMPLICIT REAL*8 (A-H,K,M-Z)
CHARACTER*15 FILE
CHARACTER*64 TITLE
LOGICAL IOREAD,IOCLOS
DIMENSION CP(5),CPRG(3),GTPRG(3),G(6,2),KCPR(6),MB2(6),
*M(5),N(6),PO(5),TO(5),T3R(2),M4R(2)
COMMON/CMPSR/ETCRF,K2,KETC(6),KM1(6),M1RF
COMMON/CMBSTN/FA
COMMON/GSTRBN/ETGTRF,GTNRF,M3RF,NGG,TO3,K3
COMMON/PTRBN/M4RF,ETPTRF,PTPR,PTPRF
EQUIVALENCE(TO3,TO(3))

```

C

C

Open data input file

C

10

```

WRITE(1) " FILENAME?"
READ(1) FILE
IF(.NOT.IOREAD(5,0,0,FILE)) GO TO 20
WRITE(1) " NO SUCH FILE!"
GO TO 10

```

C

C

Read input data

C

20

```

READ(5,1000) PAMB,TAMB
READ(5,1000) ETCRF,CNRF,CPRF,M1RF
READ(5,1000) (N(I),KM1(I),KETC(I),KCPR(I),G(I,1),G(I,2),MB2(I),
* I=1,5)
READ(5,1000) ETGTRF,GTNRF,GTNRF,M3RF
READ(5,1000) ETPTRF,PTNRF,PTPRF,M4RF
READ(5,1000) M1EFF,M2EFF,MB3,MB4
READ(5,1100) TITLE

```

C

C

Initialize variables

C

```

TO(1)=TAMB
CP(1)=1.005
CP34=1.147
TO(3)=800.0
PO(1)=PAMB
THETA=16.97056/DSQRT(TAMB)
DELTA=101.3/PAMB
NS=5

```

C

C

Select two points on a compressor speed line and start iterations

C

C

DO 30 IS=1,NS

3 3
OF / DE

```

      NGG=N(1)*DSQRT(TO(1))*CNRF
      CPRG(1)=G(IS,1)*(CPRRF-1)+1
      CPRG(2)=G(IS,2)*(CPRRF-1)+1
      JG=1
C      Set the working value of CPR equal to the guessed value
40      CPR=CPRG(JG)
      K2=(CPR*KCPR(IS)-1)/(CPRRF-1)
C      Get compressor mass flow and efficiency
      CALL CPCHIC(M(1),ETC,IS)
C      Calculate the temperature rise across the compressor
      DO 50 I=1,5
          DT12T1=(CPR**(.287/CP12)-1.0)/ETC
          TO(2)=TO(1)*(1+DT12T1)
          CP(2)=CPA(TO(2))
50      CP12=(CP(1)+CP(2))/2
C      Calculate inlet pressure loss
      DO 60 I=1,5
          MA=M(1)*PO(1)/DSQRT(TO(1))
          IPL=((2.204420475D-4*MA+1.568425731D-2)*MA-0.279863521569)
          *      *MA+1.9651509
60      PO(1)=PAMB-IPL
      PO(2)=PO(1)*CPR
C      Calculate burner inlet flow and efficiency
      M(2)=M(1)*DSQRT(TO(2)/TO(1))*(1-MB2(IS))/CPR
      HA2=HA(TO(2))
      ETB=ETBA(PO(2))

C
C      Start gas generator turbine matching by guessing the turbine
C      pressure ratio. Two guesses are used, one half the CPR and
C      the square root of CPR
C
      GTPRG(1)=DSQRT(CPR)
      GTPRG(2)=CPR/2
      LG=1
70      GTPR=GTPRG(LG)
      K3=GTPR/GTPRRF
C      Calculate the burner pressure loss, the f/a ratio, TIT based on
C      continuity
      DO 80 I=1,5
C          Get the turbine flow and efficiency
          CALL GTCHIC(M(3),ETGT)
          BPL=BP(M(2),TO(2),TO3)
          FA=(HA(TO3)-HA2)/(43540.8-HA(TO3)-THETHA(TO3))/ETB
          T3T1F=(M(3)*CPR*BPL/M(1)*(1+FA-MB2(IS)))*2
80      CP(3)=CPB(TO3)
C      Calculate temperature drop across the gas generator turbine
C      from pressure ratio
      DO 90 I=1,5
          DT34T3=ETGT*(1-(1/GTPR)**(.287/CP34))
          TO(4)=TO3*(1-DT34T3)
          CP(4)=CPB(TO(4))
90      CP34=(CP(3)+CP(4))/2
C      Calculate TIT from gas generator work balance
      T3T1W=DT12T1*CP12/M1EFF/CP34/DT34T3/(1+FA-MB2(I)-MB3)
      TO3W=T3T1W*TO(1)
C      Do values of TIT from flow and work match

```

```

C      If so, continue on to power turbine matching
C      If not, interpolate a new gas generator turbine pressure ratio
      IF(DABS(TO3W-TO3).LT.0.1) GO TO 130
      T3R(LS)=TO3W/TO3
      IF(LS.GE.2) GO TO 100
      LS=LS+1
      GO TO 70
100    GTPRG(3)=(GTPRG(2)-GTPRG(1))*(1-T3R(1))/(T3R(2)-T3R(1))+GTPRG(1)
      IF(GTPRG(3).LT.GTPRG(1).OR.GTPRG(3).GT.GTPRG(2)) GO TO 110
      LS=2
      GO TO 70
110    IF(GTPRG(3).LT.GTPRG(1)) GO TO 120
      GTPRG(1)=GTPRG(2)
      GTPRG(2)=GTPRG(3)
      T3R(1)=T3R(2)
      LS=2
      GO TO 70
120    GTPRG(2)=GTPRG(1)
      GTPRG(1)=GTPRG(3)
      LS=1
      GO TO 70

C
C      Power turbine matching
C
130    GTPR=GTPRG(2)
      PTPR=CPR*BPL*PO(1)/PAMB/GTPR
C      Get power turbine mass flow from continuity
      M4F=M(3)*GTPR*DSQRT(TO(4)/TO3)*(1+FA-MB3-MB4)/(1+FA-MB2(IS)-MB3)
C      Get power turbine mass flow and efficiency from pressure ratio
      CALL PTCHIC(M(4),ETPT)
C      Do the mass flows agree
C      If so, output results
C      If not, interpolate a new compressor pressure ratio
      IF(DABS(M4F-M(4)).LT.2.0D-3) GO TO 170
      M4R(JG)=M4F/M(4)
      IF(JG.GE.2) GO TO 140
      JG=JG+1
      GO TO 40
140    CPRG(3)=(CPRG(2)-CPRG(1))*(1-M4R(1))/(M4R(2)-M4R(1))+CPRG(1)
      IF(CPRG(3).LT.CPRG(1).OR.CPRG(3).GT.CPRG(2)) GO TO 150
      CPRG(2)=CPRG(3)
      JG=2
      GO TO 40
150    IF(CPRG(3).LT.CPRG(1)) GO TO 160
      CPRG(1)=CPRG(2)
      CPRG(2)=CPRG(3)
      M4R(1)=M4R(2)
      JG=2
      GO TO 40
160    CPRG(2)=CPRG(1)
      CPRG(1)=CPRG(3)
      JG=1
      GO TO 40
170    CP45=CP(4)
C      Calculate exhaust gas temperature
      DO 180 I=1,5

```

```

      TO(5)=TO(4)*(1-ETPT*(1-(1/PTPR)**(0.287/CP45)))
      CP(5)=CPG(TO(5))
180    CP45=(CP(4)+CP(5))/2
      IPL=IPL/PO(1)
      MF=MA*(1-MB2(15))*FA*THETA*DELTA
      NGG=NGG*THETA
      PWR=MA*(1+FA-MB3+MB4)*CP45*(TO(4)-TO(5))*M2EFF*THETA*DELTA
      SFC=MF*3600/PWR
      TO(2)=TO(2)*THETA
      TO3=TO3*THETA
      TO(4)=TO(4)*THETA
      TO(5)=TO(5)*THETA
      MA=MA*DELTA/THETA
C    Output Results
      WRITE(4,1200) TITLE
      WRITE(4,1300) TAMB,PAMB,TO(2),MB2(15),TO3,MB3
      WRITE(4,1400) TO(4),MB4,TO(5),IPL,CPR,ETC,BPL,ETB,GTPR,ETGT
      WRITE(4,1500) PTPR,ETPT,MA,MF,FA,PWR,SFC
      WRITE(4,1600) NGG,PTN
30    CONTINUE
      IF(IOCLOS(5)) STOP
      STOP
1000  FORMAT(7G0.0)
1100  FORMAT(A65)
1200  FORMAT(1X, //A0, /8('-----'), //)
1300  FORMAT(7X, 'Tamb= ', F5.1, ' (K)', 7X, 'Pamb= ', F5.1, ' (kPa)', /, 7X,
* 'TO2= ', F5.1, ' (K)', 7X, 'Mb2= ', F4.3, /, 7X, 'TO3= ', F6.1, ' (K)',
* 7X, 'MB3= ', F4.3)
1400  FORMAT(7X, 'TO4= ', F5.1, ' (K)', 7X, 'Mb4= ', F4.3, /, 7X, 'TO5= ',
* F5.1, ' (K)', /, 7X, 'IPL= ', F5.4, /, 7X, 'CPR= ', F5.3, 11X, 'CETA= ',
* F5.3, /, 7X, 'BPL= ', F5.4, 11X, 'BETA= ', F5.3, /, 7X, 'GTPR= ', F5.3,
* 11X, 'GTETA= ', F5.3)
1500  FORMAT(7X, 'PTPR= ', F5.3, 11X, 'PTETA= ', F5.3, /, 7X, 'Ma= ', F6.2,
* ' (kg/S)', 4X, 'Mf= ', F6.3, ' (KG/S)', 8X, 'f/a= ', F5.4, /, 7X,
* 'Power= ', F5.0, ' (kW)', 6X, 'SFC= ', F6.3, ' (KG/KW-hr)')
1600  FORMAT(7X, 'Ngg= ', F6.0, ' (rpm)', 5X, 'Npt= ', F6.0, ' (rpm)', /,
* 16('====='))
      END

```



```

C*****
C
C      Compressor characteristics
C
C      This routine returns mass flow and isentropic efficiency for a
C      given speed line and pressure ratio. Each characteristic
C      curve has an associated scaling factor and is multiplied by
C      the appropriate reference value.
C*****
C
C      SUBROUTINE CPCHIC(M1,ETC,I)
C      IMPLICIT INTEGER*1 (I,J,L)
C      IMPLICIT REAL*8 (A-H,K,M-Z)
C      COMMON/CMFRSP/ETCRF,I2,KETC(6),KM1(6),M1RF
C
C      Go to the appropriate curves for the selected speed line
C      GO TO (10,20,30,40,50), I
C
C      10      M1=((( -3.04228500765*K2+0.25085293198)*K2-0.08936506634)*K2
C      *      +0.548764)*M1RF*KM1(I)
C      ETC=((( -45.378293733*K2+25.042148914)*K2-3.5192481463)*K2
C      *      +0.9216795)*ETCRF*KETC(I)
C      RETURN
C
C      20      M1=((( -2.7932956947*K2+1.9096515391)*K2-0.4812977875)*K2
C      *      +0.724828715)*M1RF*KM1(I)
C      ETC=((( -6.91853447859*K2+5.065520804)*K2-0.259385705)*K2
C      *      +0.6822178658)*ETCRF*KETC(I)
C      RETURN
C
C      30      M1=((( -1.2810597089*K2+1.561598056)*K2-0.654204456172)*K2
C      *      +0.9398656279)*M1RF*KM1(I)
C      ETC=((( 0.0821268997429*K2-1.5369198301)*K2+2.09869263)*K2
C      *      +0.2599827389)*ETCRF*KETC(I)
C      RETURN
C
C      40      M1=((( -1.413468493*K2+2.3842399999)*K2-1.3203624416)*K2
C      *      +1.1637965632)*M1RF*KM1(I)
C      ETC=((( -5.2792664202*K2+9.41353520785)*K2-4.7271782052)*K2
C      *      +1.5345339807)*ETCRF*KETC(I)
C      RETURN
C
C      50      M1=((( -1.10994025421*K2+2.39170931443)*K2-1.666216432)*K2
C      *      +1.37620844866)*M1RF*KM1(I)
C      ETC=((( 1.2643174263*K2-3.8510282169)*K2+3.9699636808)*K2
C      *      -0.3838063211)*ETCRF*KETC(I)
C      RETURN
C      END

```

```

C*****
C
C      Combustion Chamber Characteristics
C
C      These routines calculate the burner pressure loss and
C      efficiency.
C*****
C
C      REAL*8 FUNCTION ETBA(P02)
C      IMPLICIT REAL*8 (A-Z)
C      ETBA=1.05239-2.67865*DSORT(1.0/P02)
C      RETURN
C      END
C
C      REAL*8 FUNCTION BP(M2,T02,T03)
C      IMPLICIT REAL*8 (A-Z)
C      BP=1.0-(0.064038*M2+0.001892*(T03/T02-1.0))
C      RETURN
C      END

```

```

C*****
C
C      Gas Generator Turbine Characteristics
C
C      This routine calls the general turbine routine, applies
C      the appropriate scaling factors and returns the
C      non-dimensional mass flow and isentropic efficiency for a
C      given pressure ratio
C
C*****
C
      SUBROUTINE GTCHIC(M3,ETGT)
      IMPLICIT REAL*8 (A-H,K,M-Z)
      COMMON/GBTRBN/ETGTRF,GTNRF,M3RF,NGB,T03,K3
      K5=NGB/DBQRT(T03)/GTNRF
C      Is the pressure ratio less than the reference value?
      IF(K3.LT.1) GO TO 10
C      If the pressure ratio is equal or equal to the reference
C      value set the mass flow and efficiency equal to their
C      reference values
      M3=M3RF
      ETGT=ETGTRF
      RETURN
C      Interpolate to obtain mass flow at pressure ratio and
C      rotational speed given
10  M3=(M100(K3)-M040(K3))*(K5-0.4)/0.6+M040(K3)*M3RF
      IF(K5.LT.1) GO TO 20
      ETGT=ETGTRF
      RETURN
C      Interpolate to obtain efficiency at pressure ratio and
C      rotational speed given
20  IF(K5.GE.0.6) GO TO 30
      ETGT40=ETT40(K3)
      ETGT60=ETT60(K3)
      ETGT=((ETGT60-ETGT40)*(K5-0.4)/0.2+ETGT40)*ETGTRF
      RETURN
30  IF(K5.GE.0.8) GO TO 40
      ETGT60=ETT60(K3)
      ETGT80=ETT80(K3)
      ETGT=((ETGT80-ETGT60)*(K5-0.6)/0.2+ETGT60)*ETGTRF
      RETURN
40  ETGT80=ETT80(K3)
      ETGT10=ETT100(K3)
      ETGT=((ETGT10-ETGT80)*(K5-0.6)/0.2+ETGT80)*ETGTRF
      RETURN
      END

```

```

C*****
C
C      Power Turbine Characteristics
C
C      This routine calls the general turbine routine, applies
C      the appropriate scaling factors and returns the
C      non-dimensional mass flow and isentropic efficiency for a
C      given pressure ratio
C*****
C
      SUBROUTINE PTCHIC(M4,ETPT)
      IMPLICIT REAL*8 (A-H,K,M-Z)
      COMMON/PTRBN/M4RF,ETPTRF,PTPR,PTPRRF
      K4=PTPR/PTPRRF
      IF (K4.LT.1) GO TO 10
      M4=M4RF
      ETPT=ETPTRF
      RETURN
10    M4=M100(K4)*M4RF
      ETPT=ETT100(K4)*ETPTRF
      RETURN
      END

```

```

C*****
C
C   General Turbine Characteristics
C
C   This routine computes the general turbine characteristics.
C   Mass flow for the 100 and 40 % rotational speed lines are
C   given by M100 and M040 respectively. Likewise, for efficiency
C   by ETT100, ETT80, ETT60 and ETT40. K is pressure ratio
C*****
C
C   REAL*8 FUNCTION M100(K)
C   IMPLICIT REAL*8 (A-Z)
C   M100=((3.3204338306*K-9.9579431389)*K+10.0339571147)*K
C   *      -2.3965131155
C   RETURN
C   END
C
C   REAL*8 FUNCTION M040(K)
C   IMPLICIT REAL*8 (A-Z)
C   M040=((6.1753435615*K-17.1126611847)*K+15.7458116916)*K
C   *      -3.8116034929
C   RETURN
C   END
C
C   REAL*8 FUNCTION ETT40(K)
C   IMPLICIT REAL*8 (A-Z)
C   ETT40=((0.15144275226*K+0.61720456253)*K-2.07371766676)*K
C   *      +1.9297876067
C   RETURN
C   END
C
C   REAL*8 FUNCTION ETT60(K)
C   IMPLICIT REAL*8 (A-Z)
C   ETT60=((7.9942426206*K-20.8332535034)*K+17.3885879643)*K
C   *      -3.17173320520
C   RETURN
C   END
C
C   REAL*8 FUNCTION ETT80(K)
C   IMPLICIT REAL*8 (A-Z)
C   ETT80=((9.6315153951*K-25.9504276913)*K+22.9444679172)*K
C   *      -5.666920001517
C   RETURN
C   END
C
C   REAL*8 FUNCTION ETT100(K)
C   IMPLICIT REAL*8 (A-Z)
C   ETT100=((13.5955097544*K-35.8443261515)*K+31.450509)*K
C   *      -8.19313816092
C   RETURN
C   END

```

```

C*****
C
C      Working Fluid Properties (From Chappell, M.S. and
C      Cockshutt, E.P.)
C*****
C
C      Specific heat for air (KJ/kg/K)
C
C      REAL*8 FUNCTION CPA(T)
C      IMPLICIT REAL*8 (A-Z)
C      If temperature > 800K use 2nd polynomial fit
C      IF(T.GT.800) GO TO 10
C      CPA=(((8.93788182D-12*T-5.4651988D-8)*T+1.2754298D-4)*T
C      *      -3.2921648D-5)*T+0.24336328)*4.18679
C      RETURN
10    CPA=(((8.93788182D-12*T-5.4651988D-8)*T+1.2754298D-4)*T
C      *      +0.19075549)*4.18679
C      RETURN
C      END
C
C      Specific heat for combustion products (KJ/kg/K)
C
C      REAL*8 FUNCTION CPG(T)
C      IMPLICIT REAL*8 (A-Z)
C      COMMON/CMBSTN/FA
C      CPG=CPA(T)+FA*THETCP(T)/(1+FA)
C      RETURN
C      END
C
C      Variation of specific heat due to fuel added (KJ/kg/K)
C
C      REAL*8 FUNCTION THETCP(T)
C      IMPLICIT REAL*8 (A-Z)
C      If temperature > 800K use 2nd polynomial fit
C      IF(T.GT.800) GO TO 10
C      THETCP=(((((-2.9202103D-15*T+6.8521981D-12)*T-5.1850414D-9)*T
C      *      +6.7154771D-7)*T+1.0787235D-3)*T-8.5863702D-2)*4.18679
C      RETURN
10    THETCP=(((((-6.2379644D-18*T+7.3252744D-14)*T-2.9619121D-10)*T
C      *      +4.5763254D-7)*T-3.3817793D-5)*T+0.26004518)*4.18679
C      RETURN
C      END
C
C      Enthalpy of air (KJ/kg)
C
C      REAL*8 FUNCTION HA(T)
C      IMPLICIT REAL*8 (A-Z)
C      If temperature > 800K use 2nd polynomial fit
C      HA=(((((-1.7976731D-14*T+2.53172212D-11)*T+1.579838D-8)*T
C      *      -1.6460824D-5)*T+0.24336328)*T-0.405671)*4.18679
C      RETURN
10    HA=(((2.2344545D-12*T-1.8217329D-8)*T+6.376249D-5)*T+0.19075549
C      *      *T+11.31763)*4.18679
C      RETURN
C      END

```

```
C
C      Variation of enthalpy due to fuel added (KJ/kg)
C
      REAL*8 FUNCTION THETHA(T)
      IMPLICIT REAL*8 (A-Z)
C      If temperature > 800K use 2nd polynomial fit
      THETHA=((((8.3337678D-14*T+5.95612D-11)*T-5.175189D-7)*T
*          +7.67971033D-4)*T-0.12635675)*T+14.969069)*4.18679
      RETURN
10    THETHA=((((3.9185699D-15*T-2.9954469D-11)*T+6.0082756D-8)
*          +8.7121923D-5)*T+0.19989167)*T-42.237153)*4.18679
      RETURN
      END
```

101.3,288.0
 .842,940.0,6.448,3.65229
 0.70,1.123,1.020,0.7000,0.380,0.36,0.033
 0.80,1.159,0.999,0.8000,0.575,0.55,0.015
 0.90,1.136,0.995,0.9200,0.770,0.75,0.017
 0.95,1.125,0.958,0.9600,0.920,0.85,0.020
 1.00,1.115,1.020,0.9295,1.040,0.97,0.000
 0.870,465.37,1.90,1.22272
 0.870,295.61,1.85,3.27
 0.97,0.97,0.0,0.0

Reference Conditions

Reference Conditions

Tamb= 288.0 (K)	Pamb= 101.3 (kPa)	
T02= 432.5 (K)	Mb2= .033	
T03= 792.8 (K)	Mb3= .000	
T04= 652.2 (K)	Mb4= .000	
T05= 628.0 (K)		
IPL= .0100		
CPR= 3.053	CETA= .743	
BPL= .9578	BETA= .899	
GTPR= 2.388	GTETA= .870	
PTPR= 1.212	PTETA= .740	
Ma= 12.44 (kg/s)	Mf= .121 (kg/s)	f/a= .0101
Power= 317. (kW)	sfc= 1.378 (kg/kW-hr)	
Ngg= 11167. (rpm)	Npt= 7549. (rpm)	

=====

Reference Conditions

Tamb= 288.0 (K)	Pamb= 101.3 (kPa)	
T02= 463.2 (K)	Mb2= .015	
T03= 831.3 (K)	Mb3= .000	
T04= 664.5 (K)	Mb4= .000	
T05= 617.7 (K)		
IPL= .0201		
CPR= 4.052	CETA= .800	
BPL= .9560	BETA= .919	
GTPR= 2.736	GTETA= .870	
PTPR= 1.388	PTETA= .842	
Ma= 16.24 (kg/s)	Mf= .163 (kg/s)	f/a= .0102
Power= 802. (kW)	sfc= .731 (kg/kW-hr)	
Ngg= 12762. (rpm)	Npt= 7620. (rpm)	

=====

END

21-02-86

FIN