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AXIAL FLOW COMPRESSOR MODELLING
FOR
ENGINE HEALTH MONITORING STUDIES
by

© David E. Muir, B.Eng., P.Eng.

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

Department of Mechanical and Aeronautical Engineering
Carleton University
Ottawa, Ontario
Canada

January 1988

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H. H. Leran

Thesis Supervisor

ABSTRACT

The Department of National Defence has identified a need for improved Engine Health Monitoring procedures for the new Canadian Patrol Frigate. The Canadian Patrol Frigate propulsion system includes two General Electric LM2500 gas turbines, a modern high pressure ratio engine which employs multiple stages of compressor variable geometry. Since significant variations in overall engine performance can result from the tolerances allowed on variable geometry setup or from a malfunction of the variable geometry system in service, a means of estimating the influence of changes in compressor variable geometry on LM2500 performance is required.

A general method for predicting the thermodynamic performance of variable geometry axial compressors has been developed. The method has been successfully applied to the General Electric LM2500-30 marine gas turbine compressor and the variation in overall engine performance due to changes in compressor variable geometry setting has been estimated.

The new compressor performance modelling technique is based on a meanline stage-stacking analysis, relies only on the limited performance data typically made available by engine manufacturers and represents a considerable advance over previous methods used for Engine Health Monitoring studies.

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NOMENCLATURE

A_N	-	stage inlet effective annulus area
C_{a1}	-	stage inlet axial air velocity
C_1	-	stage inlet absolute air velocity
C_p	-	specific heat at constant pressure
M_1	-	stage inlet absolute Mach number
N_1	-	compressor rotational speed
N_2	-	power turbine rotational speed
P_i	-	total pressure at station i
P_{os}	-	stage inlet total pressure
PR	-	compressor pressure ratio
PR_s	-	stage pressure ratio
Q_1	-	stage inlet flow function as defined by equation (11)
r_m	-	stage inlet mean radius
SHP	-	shaft horsepower
T_i	-	total temperature at station i
T_{os}	-	stage inlet total temperature
ΔT_{os}	-	total temperature rise across compressor stage
U	-	compressor rotor tangential blade speed at mean radius
$\dot{W}_i$	-	mass flow rate at station i
$\dot{W}_f$	-	engine fuel flow rate
α_1	-	stage inlet absolute air angle
$\Delta \alpha_1$	-	change in stage inlet absolute air angle relative to match speed conditions
β_1	-	stage inlet relative air angle
γ	-	ratio of specific heats
δ_1	-	corrected pressure at station i, $P_i/14.7$ psia

- ζ - stage temperature rise coefficient, $C_p \Delta T_{OS} / U^2$
 ζ_{ref} - stage temperature rise coefficient at maximum stage efficiency
 η - stage isentropic efficiency,

$$\frac{T_{OS} (PR_S^{\frac{\gamma-1}{\gamma}} - 1)}{\Delta T_{OS}} = \psi / \zeta$$

- η_{ref} - maximum stage efficiency
 θ_i - corrected temperature at station i, $T_i / 519^\circ R$
 λ - variable geometry setting angle
 ϕ - stage flow coefficient, C_{a1} / U
 ϕ_{ref} - stage flow coefficient at maximum stage efficiency
 ψ - stage pressure rise coefficient,

$$C_p T_{OS} (PR_S^{\frac{\gamma-1}{\gamma}} - 1) / U^2$$

 ψ_{ref} - stage pressure rise coefficient at maximum stage efficiency

Station Numbers

- a - ambient
 1 - compressor inlet
 2 - compressor exit
 3 - gas generator turbine inlet
 4 - power turbine inlet
 5 - power turbine exit

Chapter 1

INTRODUCTION

Since early 1982, the Canadian Navy, with specialist support from GastOPS Ltd., has been developing on-board gas turbine performance monitoring and fault diagnosis procedures for its Tribal class warships [1]. To date, a ruggedized computer installation has been developed which provides the ship's Engineering Officer with an automated tool for collecting and trending engine thermodynamic and mechanical vibration data and an interactive procedure by which likely engine faults can be systematically investigated using computer models of the main (Pratt and Whitney FT4), cruise (Pratt and Whitney FT12) and auxiliary (Solar Saturn) gas turbine engines. The experience gained from the Tribal class engine performance monitoring and fault diagnosis program is intended to provide a basis for the design and development of a similar system for the new Canadian Patrol Frigate (CPF).

The CPF propulsion system includes two General Electric LM2500 gas turbines, a modern high pressure ratio engine which employs multiple stages of variable compressor geometry. In order to accurately predict the performance characteristics of the variable geometry compressor and to provide a more fundamental approach to axial compressor fault modelling, a new compressor modelling technique has been developed. The new technique includes individual compressor stage performance modelling and uses a meanline stage-stacking analysis to synthesize overall compressor performance. Notably, the modelling procedures rely only on the limited performance data typically made available by engine

manufacturers. From these data, representative stage characteristics are inferred.

This thesis describes the development of the new axial compressor performance modelling technique and the application of this technique to the LM2500-30 marine gas turbine.

Chapter 2

BACKGROUND

2.1 Gas Turbines for Naval Propulsion

Gas turbine engines, in particular aero-derivative units, have assumed a dominant role in naval propulsion; so much so, that virtually all modern naval surface vessels include gas turbines in their propulsion machinery. This predominance arises from several factors [2]:

1. High specific power, low specific weight and low specific volume leads to a relatively compact, lightweight powerplant.
2. Fast response and flexible operating characteristics.
3. Repair of the gas generator by replacement reduces on-board maintenance requirements and improves availability.
4. High reliability.
5. Remote and/or automated engine control reduces manning requirements.

The first major gas turbine powered naval vessels were the Royal Navy "Tribal" class frigates and "County" class destroyers which entered service in the late 1950s. These ships featured a gas turbine engine designed specifically for the warship/frigate application as the boost engine in a

COSAG (Combined Steam And Gas turbine) powerplant arrangement. The development of gas turbine engine technology in the 1950s and 1960s, however, was directed towards the large and growing markets for aircraft engines. The advantages of the small, light, high performance aero-engines for marine application were quickly recognized and by the 1970s the aero-derivative marine gas turbine had become the standard for naval gas turbine applications. The primary advantage of the aero-derivative gas turbine, apart from proven performance and reliability, is its cost-effectiveness. Although the marinization of an aero-engine requires modifications to the gas generator to include corrosion resistant materials, the addition of a power turbine, modifications to rotor bearing systems and changes to the fuel and combustion systems to cater for the heavier distillate fuels used at sea, the basic aerodynamic and mechanical design and development work done for the aero-engine remains valid, with resultant savings in both cost and development time. Additionally, the market for a naval gas turbine is typically small and the large costs associated with a "new" engine development program cannot reasonably be spread over the number of units produced.

The similarity of airborne and naval powerplant requirements (compactness, lightweight, flexibility of operation, high reliability, etc.) has enabled propulsion system designers to benefit directly from successive generations of aero-engine technologies. Modern naval gas turbines such as the LM2500 have taken full advantage of demonstrated performance improvements achieved through higher pressure ratios, increased turbine temperatures and variable turbomachinery geometry. Similarly, improvements in the operational and maintenance characteristics of naval gas turbines, including

increased component lives, reduced times between overhaul, modular construction and improved inspection and monitoring techniques, have also been realized.

2.2 LM2500 Engine

The LM2500 gas turbine is a direct outgrowth of the TF39 turbofan engine used in the U.S. Airforce C5 transport aircraft. A commercial derivative of this engine, the CF6, is used to power the McDonald Douglas DC-10, Airbus Industries A300 and A310 and Boeing 747 wide body aircraft. The LM2500 incorporates 16 stages of axial compression with an overall pressure ratio of approximately 18 to 1. The first seven stages of the compressor are provided with variable stators which alleviate the part speed stability problems normally associated with high pressure ratio compressors. The remainder of the gas generator section consists of a single annular combustor and a two-stage, air-cooled turbine. The free power turbine has six stages, a rated speed of 3600 RPM and nominally provides 25,000 hp of shaft power. A cross-sectional view of the LM2500 engine is shown in Figure 1.

The LM2500 marine engine was first demonstrated in 1969 as a retrofit powerplant for the GTS Adm. Callaghan, a U.S. Navy cargo vessel [3]. Based on the success of these initial sea trials, the U.S. Navy selected the LM2500 as the powerplant for its DD963 "Spruance" class destroyers, an all-gas turbine powered warship incorporating four LM2500s in a COGAG (Combined Gas And Gas) arrangement. The U.S. Navy has subsequently chosen the LM2500 for other new ships, notably their FFG class frigates, CG47 class cruisers and PHM class hydrofoils [4].

As indicated in Table 1, many navies in addition to the U.S. Navy have incorporated or plan to incorporate LM2500 engines in the propulsion systems for their warships, including the Canadian Navy's new Canadian Patrol Frigate. Apart from marine applications, the LM2500 is also utilized for pipeline gas compression [5], offshore platform drive applications [6], and industrial power generation [7].

2.3 Gas Turbine Performance Monitoring

In recent years, the gas turbine industry has expended a considerable effort in developing equipment for the purposes of monitoring engine condition and providing operators with more detailed information upon which to base their maintenance decisions. Concurrent with these developments, the manufacturers have offered engines of modular construction, enabling operators to adopt a "repair by replacement" maintenance practice. The current trend in gas turbine engine maintenance philosophy, particularly amongst military operators, is towards an on-condition maintenance (OCM) concept which takes advantage of both the modular engine construction and improved engine monitoring equipment. Under an OCM program, engine maintenance requirements are identified strictly according to actual equipment condition.

An important element of the OCM concept is Engine Health Monitoring (EHM), whereby special surveillance techniques are employed to gather and process condition data, assess engine health, and diagnose engine faults before a failure occurs. There is an extremely wide variety of EHM techniques either in use or under development at the present time. A general review of these techniques is presented by MacIsaac [8] and the application of certain techniques to naval gas turbines

is discussed in Reference [9]. In general, EHM systems may be considered under three main categories:

1. Mechanical Health - The most widely used approach is vibration monitoring. Overall level checks and level tracking according to rotor speed are both common. Detailed frequency/amplitude content analysis, if necessary, is usually performed by a specialist group.
2. Oil Analysis - A number of techniques ranging from simple magnetic chip detectors to more sophisticated methods such as Spectrometric Oil Analysis and Ferrography have been employed with varying degrees of success.
3. Performance Analysis - These procedures range from simple Trend Analysis to the more advanced Gas Path Analysis techniques. Performance analysis techniques are predicated on the assumption that the mechanical health of an engine is reflected in its thermodynamic behaviour. As such, an engine degradation causes changes in monitored performance parameters away from their healthy or baseline values. Trend Analysis merely allows the operator to track the deviations in directly measured parameters, whereas Gas Path Analysis generally uses the instrument readings to deduce additional important cycle parameters and to isolate engine faults to a specific component or module.

The general approach to gas turbine performance monitoring and performance related fault diagnosis adopted by the

Canadian Forces is outlined in Figure 2. Within a specific data capture "window", a data acquisition system conditions and samples—engine sensor signals and provides the data reduction system with raw data in engineering units. As a minimum, the data processing function usually includes data validity checks, correction for non-standard engine inlet conditions, comparison of the measured performance parameters to "healthy" or baseline values, evaluation of performance deviations (Engine Fault Signature) and performance deviation display. Performance related fault diagnosis is accomplished by comparing the Engine Fault Signature to a "library" of known or predicted fault signatures. The Fault Library is, in turn, established using a "diagnostic" engine model. The diagnostic engine model concept is described briefly in the following section.

2.4 Diagnostic Engine Modelling

Engine manufacturers normally provide operators with performance specifications, in the form of expected power and specific fuel consumption specifications, performance limits for safe operation or computer decks for steady state performance estimation, all for a nominal engine. This information is essential to the user to ensure that each engine is capable of meeting its anticipated performance requirements. In the event of a performance deterioration, however, it is of minimal help in identifying the cause of the engine degradation. In order to isolate performance related engine problems to a given component, the operator must be able to predict both overall and component performance under degraded engine conditions.

The overall performance of a gas turbine engine is governed by the performance characteristics of its constituent components and by the laws of compatibility of mass flow, work and rotational speed which determine the interaction or "matching" between these components. The procedures for gas turbine component matching are adequately described in Reference [10] and, if suitable component performance representations can be acquired or estimated, engine performance over a wide range of operating conditions can be predicted. Having established a component-based engine model, the influence of specific modes of component degradation on overall engine performance can then be investigated by appropriately modifying the individual component performance characteristics.

A major impediment to the development of component-based engine models is the lack of available component data. These data are usually proprietary to the engine manufacturer and with the scant information which is normally available, the estimation of suitable component performance maps remains, at best, a difficult task. The most troublesome component in this regard is the compressor, because of the strong dependency of its performance on rotational speed. Relatively simple methods of estimating compressor performance such as the scaling techniques described in Reference [11] have been used successfully to model certain fixed geometry compressors; however, because of the assumption of geometric similarity which underlies these methods, they are unsuitable for estimating the performance of compressors which incorporate variable stator geometry.

Until very recently, component performance data under degraded gas path conditions were virtually non-existent.

Initial attempts at component fault modelling for engine health monitoring studies involved adjusting the component performance maps within a component-based engine model to match observed field data [11]. For multi-stage turbomachinery components, however, this approach lacks generality. Most controlled investigations of turbomachinery performance deterioration have dealt with a single compressor or turbine stage. In order to make use of the findings of these investigations, it is necessary to establish component models based on individual stage performance.

The remainder of this thesis describes the development and application of a more fundamental approach to multi-stage component performance modelling; specifically, multi-stage axial compressors. The new modelling technique utilizes individual stage performance characteristics to synthesize overall compressor performance. As such, the performance of variable geometry axial compressors can be estimated accurately.

Notably, the new compressor modelling technique relies only on the limited performance data typically made available by the engine manufacturer.

Chapter 3

COMPRESSOR MODEL DEVELOPMENT

The development of a modelling technique for estimating the overall performance of an axial compressor which includes one or more rows of variable stator geometry is described in this chapter. The basis of the modelling technique is a meanline stage-stacking analysis which incorporates individual stage performance modelling. Since stage performance data and compressor aerodynamic design details are usually proprietary to the engine manufacturer, estimated stage performance curves are inferred from available overall compressor performance data.

3.1 Axial Compressor Stage Characteristics

The overall performance of a multi-stage axial compressor depends on the performance of its constituent stages. Assuming that the air is compressed adiabatically and behaves as a perfect gas with a constant specific heat, the performance of an axial compressor stage can be presented in terms of non-dimensional coefficients for:

flow: $\phi = C_{al}/U$ (1)

pressure ratio: $\psi = \frac{C_p T_{os} (PR_S^{\frac{\gamma-1}{\gamma}} - 1)}{U^2}$ (2)

temperature rise: $\zeta = \frac{C_p \Delta T_{os}}{U^2}$ (3)

and efficiency: $\eta = \frac{T_{os} (PR_S^{\frac{\gamma-1}{\gamma}} - 1)}{\Delta T_{os}} = \psi/\zeta$ (4)

Figure 3 illustrates typical variations of pressure, temperature and efficiency coefficient with flow coefficient for an axial compressor stage. These relationships, commonly referred to as stage characteristics, can be estimated from two-dimensional cascade data and appropriate correction factors for endwall effects [12]. However, most engine manufacturers rely on experimental measurements taken in a multi-stage environment (compressor rig) to accurately define stage performance.

A reference flow condition, arbitrarily defined as the point corresponding to the maximum stage efficiency, is indicated on the stage characteristic curves of Figure 3. The use of this reference point in generalizing axial compressor stage performance is outlined in Section 3.2.

The effect of variable stator vane stagger on the stage characteristics of a single axial compressor stage can be examined theoretically by considering the relationship between the stage temperature rise coefficient and the velocity diagram parameters for a given streamline. Figure 4 defines the velocity diagram parameters. For a stage with equal inlet and exit axial air velocities, the relationship between tangential blade speed, U , and axial velocity is given by;

$$\frac{U}{C_{al}} = \tan \alpha_1 + \tan \beta_1 \quad (5)$$

and the specific work input of the stage can be expressed as:

$$C_p \Delta T_{os} = U C_{al} (\tan \beta_1 - \tan \beta_2) \quad (6)$$

By combining equations (5) and (6) and dividing throughout by U^2 , the stage temperature rise coefficient can be expressed as a function of the flow coefficient, the rotor inlet absolute air angle, α_1 , and the rotor exit relative air angle, β_2 , as follows:

$$\frac{C_p \Delta T_{os}}{U^2} = 1 - \frac{C_{a1}}{U} (\tan \alpha_1 + \tan \beta_2) \quad (7)$$

$$\text{or,} \quad \zeta = 1 - \phi (\tan \alpha_1 + \tan \beta_2) \quad (8)$$

Since α_1 and β_2 are determined largely by the stator and rotor blade geometries, the temperature rise characteristic for a fixed geometry compressor stage can be approximated by a straight line with a negative slope of $(\tan \alpha_1 + \tan \beta_2)$. By changing α_1 with variable stagger stator vanes, however, the slope or location of the temperature rise characteristic also changes as illustrated in Figure 5.

The air angles α_1 and β_2 are not entirely independent of flow coefficient and the actual temperature rise characteristic will deviate from its linear approximation, particularly at low flow coefficient values where rotor stall effects have a strong influence on the effective flow angles. Nevertheless, by the principle described above, flow instabilities at low ϕ values can be avoided by using variable compressor geometry to shift the location of the stage characteristics.

3.2 Generalized Stage Characteristics

For modelling purposes, it is useful to develop generalized relationships between ψ/ψ_{ref} , η/η_{ref} , and ϕ/ϕ_{ref} . The values of ψ_{ref} and ϕ_{ref} are arbitrarily chosen as those corresponding to the maximum stage efficiency, η_{ref} . As well, since:

$$\eta = \psi/\zeta \quad (9)$$

it follows that

$$\zeta/\zeta_{\text{ref}} = \frac{\psi/\psi_{\text{ref}}}{\eta/\eta_{\text{ref}}} \quad (10)$$

The importance of the generalized stage characteristics representation is that, if the maximum efficiency point (ϕ_{ref} , ψ_{ref} , η_{ref}) of a stage is known or assumed, the complete ψ , η , and ζ stage characteristics can be deduced readily.

Figure 6 shows a plot of ψ/ψ_{ref} vs. ϕ/ϕ_{ref} assembled using stage pressure rise data from a number of publications [13, 14, 15, and 16]. The bold line represents the curve which best models the LM2500-30⁰ compressor stage pressure rise characteristics, as determined by the stage characteristic estimation procedures described in Section 3.4. A generalized stage efficiency curve developed by Howell and Bonham [12] is given in Figure 7. The curve is based on fifteen stage characteristics, seven theoretical and eight

experimental, and was found to adequately model the off-design variation in compressor stage efficiency for the LM2500-30.

3.3 Stage-Stacking

If the individual performance characteristics of each axial compressor stage are known, overall compressor performance over a range of rotational speeds and mass flows can be estimated by a "stage-stacking" procedure. The stage-stacking method is a simple one-dimensional or meanline calculation procedure which makes use of the dimensionless stage characteristics, along with certain gas path geometry data, to evaluate the overall compressor pressure ratio and temperature rise. Details of the stage-stacking procedure can be found in References 17, 18, 19 and 20.

In addition to the performance characteristics of each stage, the following data are also required:

- 1) Effective annulus areas at the inlet to each stage, A_N .
- 2) Mean radii at the inlet to each stage, r_m .
- 3) Absolute flow angles at the inlet to each stage, α_1 .

The stage-stacking procedure begins by solving the continuity of flow equation at the inlet to the first compressor stage. From this calculation, the first stage inlet axial velocity and flow coefficient can be determined. For example, at assigned values of compressor mass flow, speed,

and inlet conditions (P_{OS} , T_{OS}), the calculation proceeds as follows:

$$1. \text{ Evaluate } Q_1 = \frac{w_1 \sqrt{T_{OS}}}{A_N \cos \alpha_1 P_{OS}} \quad (11)$$

$$2. \text{ From } Q_1 = f(M_1, \gamma) \quad (12)$$

evaluate M_1 by iteration

$$3. \text{ From } \frac{C_1}{\sqrt{T_{OS}}} = f(M_1, \gamma) \quad (13)$$

evaluate C_1

$$4. \text{ Evaluate } \phi_1 \text{ by:}$$

$$\phi_1 = \frac{C_{a1}}{U} = \frac{C_1 \cos \alpha_1}{\frac{2\pi N_1}{60 r_m}} \quad (14)$$

The first stage ϕ , in turn, determines the first stage pressure rise and temperature rise coefficients from which the inlet conditions to the second stage are obtained. Thus, using the equations given above, the stage-by-stage calculation is continued throughout the compressor until the performance of each stage has been determined. It should be noted that a uniform variation of axial velocity across the annulus at the inlet to each stage is assumed.

3.4 Development of LM2500 Stage Characteristics

LM2500-30 compressor stage characteristics have been estimated using a numerical search procedure which makes use of the stage-stacking analysis to match overall compressor performance to a known running line. The basic method is summarized in Figure 8. At a given compressor speed and mass flow (established by the operating line data points) the numerical search procedure begins by assuming specific values for the reference point (η_{ref} , ϕ_{ref} and ψ_{ref}) of each stage. The generalized stage characteristic representations are then used to obtain the performance maps for each stage and the stages are "stacked" to get the overall compressor pressure ratio, PR, and temperature rise ratio, $\Delta T/T$. If the calculated PR and $\Delta T/T$ values do not agree with the operating line data, then the assumed values of η_{ref} , ϕ_{ref} and ψ_{ref} for each stage are modified.

In order to implement the above search procedure at various points along the LM2500-30 compressor operating line, certain data are required and a number of simplifying assumptions must be made. These data requirements and assumptions are described in the following sections.

3.4.1 Compressor Operating Line Data

Reference 21 provides tabulated LM2500-30 engine performance data over a wide range of inlet temperatures (-65°F to 130°F) and power turbine speeds (1200 to 3600 RPM) at an inlet pressure of 14.7 psia. Included in these data are the compressor rotational speed, N_1 , inlet air mass flow rate, W_1 , exit pressure, P_2 , and exit temperature, T_2 , from which the corrected compressor operating characteristics can be

derived. Table 2 presents the LM2500-30 operating line data obtained from Reference 21 and used for the stage characteristic development work.

3.4.2 Gas Path Dimensions

The stage-stacking analysis requires that the mean radius and effective annulus area at the inlet to each stage be known. These dimensions were estimated from a cross-sectional layout of the LM2500 presented in a General Electric product brochure [22] and subsequently refined in the course of the stage characteristic development work. Table 3 summarizes the estimated LM2500-30 compressor gas path geometry.

3.4.3 Flow Angles

The stage-stacking analysis also requires that the absolute air flow angle at the inlet to each stage be known. Additionally, for a variable geometry compressor, the variation in these angles with different variable geometry positions must be known or assumed.

Figure 9 presents the LM2500-30 variable geometry schedule obtained from field measurements on a gas compression unit. The stagger angle of the inlet guide vanes (IGVs) and first six rows of stator vanes are varied by seven rings under the control of a single actuating lever. As shown in Figure 9, the angular position of the actuating lever is scheduled as a function of corrected compressor speed, $N_1/\sqrt{\theta_1}$. As compressor speed is reduced, the stagger of the

variable vanes is increased to limit the amount of air flow and thus avoid unstable operating conditions.

Since details of the compressor blading and the variable geometry actuating mechanism are proprietary to the engine manufacturer, a number of assumptions were made concerning the stage inlet absolute flow angles. Firstly, it was assumed that at the fully open actuator position ($\lambda = -3$ degrees) the flow angle at the inlet to each stage was 15 degrees. This value is typical of axial compressor designs for gas turbine applications. Secondly, it was assumed that the IGVs are rotated such that the relative flow angle at the compressor inlet is kept nearly constant. The relative flow angle, β_1 , can be approximated by:

$$\tan \beta_1 = \frac{U}{C_{a1}} - \tan \alpha_1 \quad (15)$$

from which a total IGV travel of 50 degrees can be estimated. Finally, it was assumed that the total travel of the variable stator vanes decreased linearly as follows:

Total IGV travel:	50 degrees
Total Stage 1 Stator travel:	42 degrees
Total Stage 2 Stator travel:	35 degrees
Total Stage 3 Stator travel:	28 degrees
Total Stage 4 Stator travel:	21 degrees
Total Stage 5 Stator travel:	14 degrees
Total Stage 6 Stator travel:	7 degrees

Based on the above assumptions, the stage inlet absolute flow angles can be related to the variable geometry position as shown in Figure 10.

3.4.4 Stage Characteristics

As previously described, the LM2500-30 compressor stage characteristics were developed by a trial and error procedure whereby the stage characteristics for each stage were guessed, a stage-stacking analysis was then performed and finally, the computed overall compressor performance was compared to known operating data. Since the LM2500-30 compressor is comprised of 16 stages, 16 different sets of stage characteristics are possible. In order to make the stage characteristic development process more manageable, three simplifying assumptions were made:

1. The shape of each stage characteristic was defined by the generalized pressure coefficient and efficiency curves shown in Figures 6 and 7 respectively and summarized in Tables 4 and 5.
2. At a corrected speed referred to as the "match" speed, it was assumed that the performance of all stages could be modelled by a single set of "stage-averaged" stage characteristics. This speed was determined by trial and error to be $N_1/\sqrt{\theta_1} = 9160$ RPM.
3. At each speed other than the match speed, the fixed geometry stages were modelled by their match speed characteristics and the variable geometry stages were modelled by a series of stage characteristics, each representing a different change in rotor inlet absolute flow angle relative to the match speed angle.

Figure 11 and Table 6 give the LM2500-30 stage characteristics which were derived from the procedures described above. It is worth noting that, although the numerical

search procedure was automated or computerized, the generalized pressure coefficient curve (Figure 6) and the match speed value noted above were determined largely by trial and error.

Figure 12 shows an overall compressor pressure ratio versus flow map obtained from the stage stacking analysis using the stage characteristics of Figure 11. The influence of the variable geometry setting on compressor performance is clearly visible; hence, it can be expected that problems with the variable geometry system in service will result in significant performance deviations. It is further noted that the compressor characteristics presented in Figure 12 show good agreement with the only published LM2500 compressor map data known to the author [23].

Chapter 4

ENGINE MODEL STUDIES

The sensitivity of LM2500-30 overall engine performance to compressor variable geometry setting has been investigated using a component matching program for simple cycle, two shaft gas turbine engines [24, 25, 26]. This chapter presents a brief description of the engine model and summarizes the results obtained from the model studies.

4.1 Engine Model Description

As depicted in Figure 13, the General Electric LM2500-30 is comprised of four major components; the compressor, the combustor, the gas generator turbine and the power turbine. If suitable thermodynamic descriptions of these components can be obtained, the steady state performance of the engine can be predicted using well-established component matching procedures [10]. For a single spool, free turbine engine such as the LM2500, this amounts to determining the operating points which satisfy the following compatibility conditions:

1. compatibility of flow between the compressor and the gas generator turbine
2. compatibility of flow between the gas generator turbine and the power turbine
3. balance of work between the compressor and its driving turbine

4. balance of work between the power turbine and the propeller load

These operating points are found by a trial and error procedure using data obtained from the various component characteristics [24].

The development of LM2500-30 compressor characteristics as a function of compressor variable geometry setting was described in Chapter 3. Table 7 summarizes the estimated compressor performance data and Table 8 gives the variable geometry schedule which governs the position of the variable stator vanes under normal operating conditions.

The gas generator turbine and power turbine corrected flow and temperature drop maps were derived from the tabulated data presented in Reference 21. These data, along with reasonable assumptions concerning the secondary engine cycle parameters (i.e. combustor pressure loss, bleed flow fractions, gas properties, etc.) can be used to evaluate each of the major cycle parameters, including the turbine performance characteristics, over a wide range of engine operating conditions. Tables 9 and 10 summarize the estimated LM2500-30 gas generator and power turbine performance data and Table 11 gives the secondary engine cycle parameters.

The combustor performance is determined by a simple energy balance relationship which establishes the quantity of fuel required to raise the compressor delivery air to gas generator turbine inlet conditions [27].

4.2 Engine Model Validation

The LM2500-30 engine model was validated by comparing the model predictions to the performance data published in Reference 21 for a power turbine speed of 3600 RPM. Figure 14 presents the results of the model validation runs and indicates that the simulated performance predictions are in excellent agreement with the published data.

4.3 Prediction of Variable Geometry Effects on Engine Performance

Following the manufacture or overhaul of a gas turbine engine, a number of adjustments are normally made to the engine's power management systems to ensure that minimum performance specifications are met and that the limits of safe engine operation will not be exceeded in service. For engines with variable compressor geometry, these adjustments include set up of the variable geometry actuation system. It is possible that these set up procedures may be performed incorrectly or that wear or failure of actuating mechanism during field service can lead to a misadjustment of the compressor stator vanes.

The effects of variable geometry misadjustment on overall engine performance are illustrated in Figure 15. Superimposed on the nominal performance curves predicted by the engine model are the simulated running lines for various fixed variable geometry positions, λ . The simulated results indicate that significant changes in engine performance can be expected if the variable geometry system is not tracking properly. An increase in the variable geometry position at a

given speed is equivalent to closing the variable stator vanes, thus reducing the engine air flow, cycle pressures, fuel flow, turbine temperatures and power. For example, at the rated speed of 9160 RPM, the engine model predicts that a 1 degree increase in the variable geometry position from $\lambda = -3$ degrees to $\lambda = -2$ degrees will result in a 3% reduction in air flow, a 4% decrease in fuel flow and a power loss of 5%. Opening the variable stator vanes at a given speed has the opposite effect. It is also evident from the compressor pressure ratio versus speed curves that a reduction of the variable geometry position at lower speeds results in higher than normal pressure ratios and could possibly lead to unstable compressor operating conditions.

Chapter 5

CONCLUSIONS

The use of variable compressor geometry in high performance gas turbines, such as the LM2500 engines of the new Canadian Patrol Frigate, introduces new problems in determining the health of these units. The relationships between corrected engine performance parameters which provide useful indicators of developing problems for fixed geometry engines are no longer adequate; rather, for each different position of the variable geometry system, a unique engine running line exists.

An axial compressor modelling technique based on a meanline stage-stacking analysis has been developed which can predict the performance of variable geometry compressors over a wide range of variable geometry positions. The non-dimensional stage performance curves required by the stage stacking method are derived from generalized stage characteristic representations and a trial and error procedure which matches predicted compressor performance to data provided by the engine manufacturer.

The new modelling technique has been successfully applied to the General Electric LM2500-30 marine gas turbine compressor, a 16 stage unit with variable inlet guide vanes and 6 rows of variable stator vanes. The estimated LM2500-30 compressor performance data has, in turn, been implemented in a component-based simulation of the engine and the effects of variable geometry position on overall engine performance have been quantified.

The simulation of axial compressor performance by stage-stacking techniques represents a significant advance in modelling for Engine Health Monitoring applications and is particularly valuable for engines with variable compressor geometry. The individual stage modelling approach enables generalized representations of stage performance to be developed under both healthy and degraded compressor conditions. For example, in addition to the variable geometry studies of this thesis, compressor models based on stage-stacking procedures have also been used to investigate the effects of compressor fouling on pipeline gas turbines [28] and compressor tip clearance changes on an airborne engine [29].

The LM2500 engine is due to be installed in the first ship of the CPF class early in 1988. It is significant that a comprehensive and validated performance model of the engine has been developed before the CPF actually enters service. The judicious use of this model in conjunction with performance data obtained from initial sea trials and subsequent operations of the LM2500 will provide the Canadian Forces with a systematic means of establishing the health of their equipment and scheduling maintenance on an on-condition basis.

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Table 1
LM2500 Naval Installations [4]

Class	Type	Country	Propulsion System	Rating (HP)	Displacement (Tons)
PHM-1/3	Hydrofoil	USA	↑	16,000/19,500	230
PSK	Gunboat	Indonesia		23,000	290
PGG-511	Gunboat	Saudi Arabia		23,000	460
PCG-612	Gunboat	Saudi Arabia		23,000	870
KCX	Corvette	Korea	CODOG	26,800	1000/1350
PCX	Corvette	Korea		26,800	1000
KV-72	Corvette	Denmark		26,800	1330
V-50	Corvette	Brazil		27,500	1930
KFX	Frigate	Korea		26,800	1950
LUPO	Frigate	Italy, Peru		25,000	2500
Maestrale	Frigate.	Venezuela	↓	25,000	3000
F-122	Frigate	Italy		25,500	3000
CPF	Frigate	W. Germany		23,500	4000
		Canada			
FFG-7	Frigate	USA, Spain	↑	20,500	3600
		Australia			
DD 963/DDG 993	Destroyer	USA		21,500	7800/8225
CG47	Cruiser	USA		21,500	9050
Garibaldi	Carrier	Italy	↓	20,000	10000
Principe De Asturias	Carrier	Spain		23,200	14000
Callaghan	Cargo	USA		20,000	24000

Table 2

LM2500-30 Compressor Operating Line Data [21]

$N_1/\sqrt{\theta_1}$	P_2/P_1	$W_1/\theta_1/\delta_1^*$	$\Delta T_{12}/T_1$
9450	18.06	147.5	1.530
9160	17.21	144.0	1.439
8971	16.25	137.8	1.389
8813	15.30	131.6	1.343
8660	14.37	125.5	1.307
8508	13.44	119.2	1.262
8364	12.45	112.2	1.220
8105	10.35	96.7	1.112
7772	7.88	76.6	.983

* lbm/sec

Table 3
Estimated LM2500-30 Compressor Gas Path Geometry

Stage	Inlet Mean Radius (in)	Effective Inlet Annulus Area (in ²)
1	11.00	553.
2	11.16	513.
3	11.31	471.
4	11.47	428.
5	11.63	383.
6	11.78	338.
7	11.94	291.
8	12.09	242.
9	12.25	213.
10	12.29	188.
11	12.32	164.
12	12.36	144.
13	12.39	127.
14	12.43	112.
15	12.46	101.
16	12.50	90.

Table 4

Generalized Pressure Coefficient Curve

Φ/Φ_{ref}	Ψ/Ψ_{ref}
0.60	0.930
0.65	0.990
0.70	1.035
0.75	1.065
0.80	1.077
0.85	1.080
0.90	1.070
0.95	1.045
1.00	1.000
1.05	0.925
1.10	0.800
1.15	0.650
1.20	0.450
1.25	0.225
1.30	0.000

Table 5
Generalized Efficiency Curve

<u>$(\zeta/\zeta_{\text{ref}})/(\phi/\phi_{\text{ref}})$</u>	<u>η/η_{ref}</u>
0.213	0.235
0.253	0.395
0.283	0.530
0.320	0.625
0.354	0.707
0.391	0.768
0.466	0.859
0.543	0.920
0.626	0.958
0.715	0.979
0.810	0.988
0.905	0.995
1.000	1.000
1.103	0.997
1.213	0.989
1.337	0.972
1.483	0.944
1.695	0.885

Table 6
Estimated LM2500-30 Compressor Stage Characteristics

$\Delta\alpha_1 = 0.0^\circ$			$\Delta\alpha_1 = 2.5^\circ$			$\Delta\alpha_1 = 7.6^\circ$			$\Delta\alpha_1 = 18.6^\circ$		
ϕ	ψ	ζ	ϕ	ψ	ζ	ϕ	ψ	ζ	ϕ	ψ	ζ
.319	.298	.395	.294	.275	.372	.273	.255	.359	.220	.206	.309
.345	.317	.399	.319	.292	.376	.296	.271	.366	.238	.219	.312
.372	.331	.400	.343	.306	.375	.319	.284	.369	.256	.229	.313
.399	.341	.398	.368	.315	.373	.341	.292	.368	.274	.235	.311
.425	.345	.393	.392	.318	.369	.364	.295	.366	.293	.238	.307
.452	.346	.385	.417	.319	.362	.387	.296	.362	.311	.239	.301
.478	.342	.378	.441	.316	.352	.409	.293	.355	.329	.236	.295
.505	.334	.368	.466	.309	.342	.432	.287	.345	.348	.231	.287
.531	.320	.350	.490	.295	.326	.455	.274	.329	.366	.221	.273
.558	.296	.326	.515	.273	.304	.478	.254	.306	.384	.204	.255
.584	.256	.286	.539	.236	.266	.500	.219	.268	.402	.177	.223
.611	.208	.240	.564	.192	.224	.523	.178	.225	.421	.144	.187
.638	.144	.189	.588	.133	.176	.546	.123	.178	.439	.099	.148
.664	.072	.095	.613	.066	.088	.569	.062	.089	.457	.050	.074
.691	.000	.000	.637	.000	.000	.591	.000	.000	.476	.000	.000

Table 7

Estimated LM2500-30 Compressor Performance
Variable Geometry Position (λ) = -3 degrees

$N_1/\sqrt{\theta_1}$	P_2/P_1	$W_1/\theta_1/\delta_1^*$	$\Delta T_{1-2}/T_1$
8105	9.61	127.8	1.020
	12.52	127.3	1.175
	13.69	126.8	1.232
	14.24	126.2	1.259
	14.54	125.7	1.275
	14.81	125.2	1.289
	15.02	124.7	1.301
	15.18	124.2	1.310
8508	9.87	134.7	1.054
	14.89	134.4	1.301
	16.06	134.1	1.355
	16.54	133.9	1.378
	16.92	133.6	1.396
	17.21	133.3	1.410
	17.43	133.0	1.421
	17.62	132.7	1.431
8813	10.47	139.3	1.114
	16.12	139.2	1.370
	17.15	139.1	1.416
	17.66	138.9	1.439
	18.09	138.8	1.458
	18.42	138.7	1.473
	18.68	138.6	1.485
	18.90	138.5	1.495
9160	11.62	144.0	1.196
	16.40	144.0	1.405
	17.49	144.0	1.451
	18.11	143.9	1.477
	18.55	143.9	1.495
	18.90	143.9	1.509
	19.20	143.9	1.522
	19.45	143.8	1.533
9450	12.88	147.7	1.272
	16.29	147.7	1.421
	17.44	147.7	1.468
	18.20	147.7	1.499
	18.73	147.7	1.521
	19.10	147.6	1.536
	19.43	147.6	1.550
	19.72	147.6	1.561

* lbm/sec

Table 7 (Cont'd)

Estimated LM2500-30 Compressor Performance
Variable Geometry Position (λ) = -0.9 degrees

$N_2/\sqrt{\theta_1}$	P_2/P_1	$W_1/\sqrt{\theta_1}/\delta_1^*$	$\Delta T_{1-2}/T_1$
8105	8.83	120.5	0.976
	12.21	120.0	1.163
	13.12	119.5	1.209
	13.68	119.0	1.239
	14.07	118.6	1.260
	14.32	118.1	1.273
	14.48	117.6	1.283
	14.63	117.1	1.292
8508	9.16	126.9	1.017
	14.05	126.6	1.269
	15.01	126.4	1.315
	15.60	126.2	1.344
	16.02	125.9	1.365
	16.28	125.7	1.377
	16.47	125.5	1.387
	16.65	125.2	1.397
8813	10.71	131.2	1.129
	14.91	131.1	1.325
	15.92	131.0	1.371
	16.52	130.9	1.398
	16.94	130.9	1.418
	17.18	130.8	1.429
	17.39	130.7	1.439
	17.58	130.6	1.448
9160	12.48	135.7	1.236
	15.38	135.7	1.367
	16.28	135.7	1.407
	16.86	135.7	1.433
	17.22	135.7	1.449
	17.55	135.6	1.463
	17.80	135.6	1.474
	18.03	135.6	1.484
9450	13.84	139.3	1.320
	15.51	139.3	1.394
	16.55	139.3	1.438
	16.97	139.3	1.456
	17.48	139.3	1.479
	17.72	139.3	1.489
	18.10	139.3	1.505
	18.29	139.3	1.514

* lbm/sec

Table 7 (Cont'd)

Estimated LM2500-30 Compressor Performance
Variable Geometry Position (λ) = 3.4 degrees

$N_2/\sqrt{\theta_1}$	P_2/P_1	$W_1\sqrt{\theta_1}/\delta_1^*$	$\Delta T_{1-2}/T_1$
8105	8.16	112.5	0.951
	11.58	112.0	1.151
	12.48	111.6	1.200
	12.95	111.1	1.226
	13.30	110.7	1.247
	13.54	110.2	1.262
	13.68	109.8	1.271
	13.81	109.3	1.279
8508	8.66	118.5	1.008
	13.09	118.3	1.244
	14.04	118.1	1.293
	14.57	117.9	1.321
	14.94	117.7	1.340
	15.22	117.5	1.355
	15.43	117.3	1.367
	15.58	117.1	1.375
8813	8.94	122.6	1.044
	13.67	122.6	1.288
	14.66	122.5	1.336
	15.24	122.5	1.365
	15.64	122.4	1.384
	15.91	122.3	1.397
	16.11	122.3	1.407
	16.30	122.2	1.417
9160	11.12	127.0	1.184
	13.89	127.0	1.320
	14.89	127.0	1.367
	15.52	127.0	1.396
	15.93	127.0	1.416
	16.27	127.0	1.431
	16.50	126.9	1.442
	16.68	126.9	1.451
9450	10.10	130.5	1.155
	13.43	130.5	1.319
	15.07	130.5	1.395
	15.72	130.5	1.424
	16.10	130.5	1.441
	16.50	130.5	1.460
	16.80	130.5	1.473
	17.00	130.5	1.482

* lbm/sec

Table 7 (Cont'd)

Estimated LM2500-30 Compressor Performance
Variable Geometry Position (λ) = 12.6 degrees

$N_2/\sqrt{\theta_1}$	P_2/P_1	$W_1\sqrt{\theta_1}/\delta_1^*$	$\Delta T_{1-2}/T_1$
8105	6.66	95.6	0.868
	9.91	95.4	1.085
	10.69	95.2	1.133
	11.09	94.9	1.159
	11.35	94.7	1.176
	11.57	94.5	1.190
	11.76	94.3	1.202
	11.90	94.1	1.212
8508	6.86	100.2	0.904
	10.86	100.2	1.160
	11.63	100.1	1.206
	12.08	100.0	1.232
	12.39	99.9	1.250
	12.62	99.9	1.264
	12.81	99.8	1.276
	12.97	99.7	1.285
8813	7.30	103.5	0.963
	10.82	103.5	1.176
	11.75	103.5	1.228
	12.25	103.5	1.256
	12.61	103.5	1.276
	12.87	103.4	1.291
	13.10	103.4	1.304
	13.30	103.4	1.315
9160	8.21	107.2	1.046
	10.92	107.2	1.205
	11.93	107.1	1.260
	12.46	107.1	1.288
	12.87	107.1	1.310
	13.16	107.1	1.326
	13.39	107.1	1.339
	13.60	107.1	1.349
9450	8.72	110.1	1.097
	10.96	110.1	1.277
	11.95	110.1	1.281
	12.51	110.1	1.311
	12.96	110.1	1.334
	13.35	110.1	1.355
	13.57	110.1	1.367
	13.79	110.1	1.378

* 1bm/sec

Table 8

Estimated LM2500-30 Compressor Variable Geometry Schedule

Corrected Speed ($N_1/\sqrt{\theta_1}$)	Variable Geometry Position (λ)
5000.	39.0
5600.	39.0
6000.	38.5
7000.	32.0
8000.	15.0
8500.	3.5
8700.	0.0
9000.	-2.5
9160.	-3.0
9500.	-3.0

Table 9

Estimated LM2500-30 Gas Generator Turbine Performance Data

Pressure Ratio (P_3/P_4)	Flow Rate ($W_3\sqrt{T_3/P_3}$)	Temperature Drop ($\Delta T_{3-4}/T_3$)
1.50	29.32	0.065
2.00	29.32	0.130
2.50	29.32	0.167
3.00	29.32	0.198
3.50	29.32	0.225
4.00	29.32	0.248
4.50	29.32	0.270
5.00	29.32	0.293

Table 10
Estimated LM2500-30 Power Turbine Performance Data

Pressure Ratio (P_4/P_5)	$N_2/\sqrt{T_4} = 60.$		$N_2/\sqrt{T_4} = 80.$		$N_2/\sqrt{T_4} = 100.$	
	Flow Rate ($W_4/\sqrt{T_4}/P_4$)	Temperature Drop ($\Delta T_{4-5}/T_4$)	Flow Rate ($W_4/\sqrt{T_4}/P_4$)	Temperature Drop ($\Delta T_{4-5}/T_4$)	Flow Rate ($W_4/\sqrt{T_4}/P_4$)	Temperature Drop ($\Delta T_{4-5}/T_4$)
1.00	0.0	0.000	0.0	0.000	0.0	0.000
1.50	86.0	0.082	80.0	0.041	72.1	0.010
2.00	101.9	0.138	96.5	0.122	92.1	0.076
2.50	109.4	0.182	104.4	0.175	99.9	0.139
3.00	113.2	0.209	108.1	0.215	104.6	0.187
3.50	115.2	0.227	110.6	0.242	107.5	0.226
4.00	116.6	0.239	112.1	0.263	109.0	0.253
4.50	117.4	0.251	112.9	0.275	110.0	0.272

Table 11
Estimated LM2500-30 Thermodynamic Cycle Constants

<u>Parameter</u>	<u>Value</u>
Specific Heat - Compressor	0.247 BTU/lbm-°R
Specific Heat - Gas Generator Turbine	0.291 BTU/lbm-°R
Specific Heat - Power Turbine	0.274 BTU/lbm-°R
Ratio of Specific Heats - Compressor	1.387
Ratio of Specific Heats - Gas Generator Turbine	1.307
Ratio of Specific Heats - Power Turbine	1.329
Mechanical Efficiency of Gas Generator Spool	98%
Fraction of Compressor Discharge Air Used for Turbine Cooling	4%
Combustor Pressure Loss	4%
Combustion Efficiency	98%

LM2500 MARINE GAS TURBINE ENGINE

Figure 1 Cross-Sectional View of LM2500 Engine

Figure 2 Gas Turbine Performance Monitoring and Fault Diagnosis

Figure 3 Axial Compressor Stage Characteristics

Velocity triangles for one stage

Cascade notation

Figure 4 Axial Compressor Velocity Triangles and Cascade Notation [10]

Figure 5 Influence of Variable Stator Vane Stagger on Axial Compressor Stage Performance

Figure 6 Generalized Stage Pressure Coefficient Curve

Figure 7 Generalized Stage Efficiency Curve [12]

Figure 8 Stage Characteristic Development

Figure 9 Estimated LM2500-30 Variable Geometry Schedule

Figure 10 Estimated LM2500-30 Compressor Stage Inlet Flow Angles

Figure 11(a) Estimated LM2500-30 Compressor Stage Characteristics

Figure 11(b) Estimated LM2500-30 Compressor Stage Characteristics

Figure 12 Estimated LM2500-30 Compressor Performance

Figure 13 LM2500-30 Engine Configuration

Figure 14(a) Engine Model Validation
($N_2 = 3600$ RPM)

Figure 14(b) Engine Model Validation
($N_2 = 3600$ RPM)

Figure 14(c) Engine Model Validation
($N_2 = 3600$ RPM)

Figure 14(d) Engine Model Validation
($N_2 = 3600$ RPM)

Figure 14(e) Engine Model Validation
($N_2 = 3600$ RPM)

Figure 15(a) Influence of Compressor Variable Geometry Setting on Engine Performance ($N_2 = 3600$ RPM)

Figure 15(c) Influence of Compressor Variable Geometry Setting on Engine Performance ($N_2 = 3600$ RPM)

COMPRESSOR SPEED, $N_1/\sqrt{\theta_1}$ (RPM)

Figure 15(d) Influence of Compressor Variable Geometry Setting on Engine Performance ($N_2 = 3600$ RPM)

Figure 15(e) Influence of Compressor Variable Geometry Setting on Engine Performance ($N_2 = 3600$ RPM)

END

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