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# Analysis of Gas Turbine Off-Design Safe Operation Using Variable Geometry Compressor

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***Abstract.*** *The conditions at which a compressor will work when installed in a gas turbine engine are determined by the ability of the components, in the gas stream behind it, to swallow the flow that it originates. Therefore running lines drawn over the compressor map may intersect the surge line. If this occurs, the engine would not run. In many cases the engine would not even accelerate to idle speed because the compressor would be required to work past the surge line. To avoid surge air could be bled at a mid compressor position. Surge is caused by misalignment of the flow with the blade passages. To avoid surge, one may reset the blade staggers in order to bring the flow aligned with the blade passages. A model for the performance prediction of variable geometry (IGV - Inlet Guide Vanes and stators) axial flow compressor has been implemented into a computer program for the estimation of engines performance. Therefore, all major important parameters, such as specific thrust or power, maximum cycle temperature, efficiency and fuel consumption, for example, can be obtained for selected range of IGV, stators settings. The results may be used for the choice of the operating point as a function of load, aiming the most economical and safety application.*

***Keywords***. *Variable Geometry, Compressor, Performance Maps, Simulation*

## 1. Introduction

Modern gas turbines are required to operate over a wide range of power to satisfy the whole engine operating envelope, like: altitude, flight speed, thrust or power, ground idle, take-off, climb, cruise, and so forth. High performance gas turbines with fixed geometry are not capable to satisfy all power requirements, due to the reduction of stability margin at lower speeds, for example. In order to satisfy all load requirements, either air bleed valve or variable geometry in the compressor, or both, must be used.

The compressor is an important component of the gas turbine because it must provide the maximum of high-pressure air to be heated in the limited volume of the combustion chamber and then expanded through the turbine. The energy that can be released in the combustion chamber is proportional to the mass of air provided by the compressor. Its efficient operation is the key to high overall engine performance. Efficient operation is achieved at maximum compression with minimum temperature rise. The power necessary to create the pressure rise of a given airflow is determined by the compressor efficiency and will affect the temperature change that can take place in the combustion chamber (Treager, 1998).

Two types of compressors are used: axial and centrifugal. The overall characteristic and applications of these two types of compressors are different. Axial flow compressor has the potential for both higher-pressure ratio and higher efficiency than centrifugal compressor. For jet engines applications, another advantage is the much larger flow rate possible for a given frontal area. Axial flow machine dominates the field for large powers and the centrifugal compressor is restricted to the lower end of the power spectrum where the flow is too small to be handled efficiently by axial blading (Cohen et al, 1996)

The importance of variable geometry is well documented by several authors (Toll et al, 1963; Allely and Eberhardt, 1964; Rahnke, 1969; Armstrong et al, 1976; Johnson, 1976; May et al, 1976; Healy, 1979; Steinmetz, 1987; Roy-Aikins, 1988; Sirinoglou, 1992; Cox et al, 1995; Rand and Wright, 2000; Cohen et al, 1996; Treager, 1998; Walsh and Fletcher, 1998; Bringhenti and Barbosa, 2001, 2002; Bringhenti et al, 2001). Gas turbines are designed to operate as efficiently as possible at design point, but the engine is required to work away from these conditions when compressor speed, altitude, ambient conditions, etc, change. At design point all components are optimized and flow is aligned with the blades, resulting high engine cycle efficiency. At off-design condition one or more components may be operating away from its design conditions. In the case of the flow either in the compressor or at the turbine, misalignments with the blades results in undesirable incidence angles and therefore decrease of performance. Variable geometry can be used to bring the flow alignment to the component and improvement to the component efficiency, what contributes to the overall gas turbine efficiency.

To satisfy the off-design conditions, variable geometry and/or bleed valves can be used. Variable geometry in the compressors may include IGV (Inlet Guide Vane) and/or VSV (Variable Stator Vane). For high-pressure compressors several rows of IGVs or VSVs at the front of the compressor stages can be used. Beyond 5 compressor stages, 1 additional stage would require the use of IGVs and/or VSVs to provide a satisfactory part speed surge line (Walsh and Fletcher, 1998). There is an improvement in surge line at low engine speeds when variable geometry compressor is used. This is particularly important during starting and operation at idle conditions. At a first order, the working line in terms of pressure ratio versus flow is unaffected by IGV and/or VSV setting. Although at off-design efficiency can be improved when IGV and/or VSV are used it is necessary to optimize the settings of each row by rig-testing the compressor (Cohen, 1996). For each IGV and/or VSV setting the compressor map is unique. Bleed valves located downstream of a compressor does not affect the compressor map but the working line shows a step change when it is open. On the other hand, when bleed valves are located part way along a multi-stage compressor, the internal geometry of the compressor changes, not just the boundary conditions imposed upon it, hence the map itself changes. To choose between VIGVs (Variable Inlet Guide Vanes) and air bleed valves is a complex matter. Bleed valves are cheaper, lighter and usually more reliable than variable vanes. However they incur a far more severe SFC (Specific Fuel Consumption) penalty since the bleed valve flow, which can be up to 25% of the mainstream and has had considerable work input, is either dumped overboard or into a bypass duct (Walsh and Fletcher, 1998). In this work it will not be considered the use of bleed valves and will not go deeper in the subject of choosing between the two options. This subject is considered well documented in the works of Steinke et al (1967), Broichhausen et al (1988), Roy-Aikins (1988), Muir et al (1989), Sirinoglou (1992), Riegler et al (2001) and Tsalavoutas et al (2001).

For the compressor geometry optimized at design (IGV and VSV fixed) the compressor map may be generated to quantify its performance under all off-design conditions. The compressor map is a set of constant corrected speed curves that represent pressure ratio and isentropic efficiency as functions of corrected mass flow. These curves are limited in the lower end of mass flow by surge (stall) and in the higher end by choke. The compressor usually is required to operate close to the surge because the high efficiency is there. Surge must be avoided at any circumstances so that the study of variable geometry is important because it is a feasible means to control the engine operation, keeping it away from surge.

Variable geometry engine analysis usually relies on computer codes for cycle simulation. This method of analysis is more flexible, faster and cheaper than experimental tools. Simulation can be considered the basis of all engine assessment. The most complete codes for engine simulation are based on blocks representing each of the engine components. In order to perform the simulation it is required that the performance characteristics of those components are available. The accuracy of the engine model is a function of the quality of these data. Usually these codes adopt some typical characteristic maps of the components available in the literature. This approach produces a fairly good simulation of the engine behavior. The problem appears when a variable geometry gas turbine needs being simulated, since the set of curves representing the component performance for each geometric change is necessary. These maps are very difficult to find in the open literature, since, in general, they are obtained by very expensive test campaign. An alternative is to introduce some sort of interpolation to the maps available in the literature for a fixed geometry, but this introduces a further approximation in the simulation and may not be adequate. Since tests are very expensive another possibility is to simulate the engine component numerically using, for example, in the case of turbomachinery, the streamline curvature method (Denton, 1973). The characteristic maps of a compressor are very time consuming to obtain in this way. The use of semi-empirical one-dimensional approaches seems to be a better option when very high precision is not an important issue.

In this work only variable geometry compressor is considered. In past works (Bringhenti et al, 2001; Bringhenti and Barbosa, 2001, 2002) the authors already reported the use of variable geometry turbine and its importance. Variable geometry nozzle application will be object of future works. The results shown in this work were obtained using the GTAnalysis computer program, written in FORTRAN (Bringhenti, 1999). It is able to simulate at steady state operation variable NGV (Nozzle Guide Vanes), variable VSV, and variable nozzle, of virtually any gas turbine of interest. This work indicates a methodology for the study of the effects of variable geometry compressor on engine performance, using GTAnalysis.

Electricity power plants usually operate at part-load, condition at which the cost of electricity may be prohibitive due to low engine efficiency. This paper aims at the study of operation at part-load condition in the search of safe and more efficient engine, through the appropriate scheduling of the low and high-pressure compressors stators.

A representative twin-shaft aero-derivative gas turbine in the range of 50 MW chosen and was modeled. Major working parameters were taken from the published literature; others were assumed based on cycle calculations and inferred engine state of the art.

## 2. Main gas turbine parameters and numerical model used

This work deals with the study of a gas turbine with variable geometry incorporated to the low (LPC) and high-pressure (HPC) compressors. At design point this engine generates the ISA power output of 40.725 MW.

Since the available published data for such engine are scarce, guesses were made in order to gather all the information for the engine simulation, as shown in Tab. (1). It must be stressed that the data contained in that table may not represent the actual engine parameters, but are enough to proceed with the study. If more reliable information is gathered in the future, the procedure developed in this work can be applied and the calculations redone, resulting in quantitatively better results. The adoption of approximated values for the parameters in Tab. (1) does not invalidate the quality of the results (Bringhenti and Barbosa, 2002).

Table 1. Gas turbine model design point characteristics

| | |
|---|---|
| Mass Flow (kg/s) | 124.561 |
| Overall Compressor Pressure Ratio | 29.48 |
| LPC – Low Pressure Compressor Pressure Ratio | 2.56 |
| HPC – High Pressure Compressor Pressure Ratio | 11.48 |
| Maximum Cycle Temperature (K) | 1530 |
| Shaft Power Output (MW) | 40.725 |
| Isentropic Efficiency of Low Pressure Compressor | 0.88 |
| Isentropic Efficiency of High Pressure Compressor | 0.87 |
| Surge Margin of Low Pressure Compressor | 0.20 |
| Surge Margin of High Pressure Compressor | 0.20 |
| Combustor Chamber Pressure Loss | 0.04 |
| Combustion Efficiency | 0.99 |
| Isentropic Efficiency of Low Pressure Turbine | 0.89 |
| Isentropic Efficiency of High Pressure Turbine | 0.90 |
| Mechanical Efficiency of Low Pressure Turbine Shaft | 0.99 |
| Mechanical Efficiency of High Pressure Turbine Shaft | 0.99 |
| Pressure Loss – Intake | 0.0 |
| Pressure Loss – Exhaust | 0.0 |
| Bleed – Percentage of total mass flow (kg/s) | 10% |
| Pressure Loss – Bleed | 0.04 |
| Fuel Flow (kg/s) | 2.295 |
| SFC – Specific Fuel Consumption (kg/s)(fuel)/kW | $0.05635 \times 10^{6}$ |
| Exhaust Gas Temperature (K) | 712.1 |
| Exhaust Gas Temperature of High Pressure Turbine (K) | 1087.9 |
| Exhaust Gas Temperature of Low Pressure Turbine (K) | 712.1 |
| Inlet Pressure (Pa) | 101325 |
| Inlet Total Temperature (K) | 288.15 |
| VSV angle of low pressure compressor at design point (degree) | 0. |
| VSV angle of high pressure compressor at design point (degree) | 0. |
| Cycle efficiency | 0.4127 |

A computer program that numerically simulates the steady state performance of complex gas turbines was used (Barbosa and Bringhenti, 1999). For a correct engine simulation by GTAnalysis computer program, the gas turbine under study was divided into component blocks, whose calculating stations were numbered according to Fig. (1).

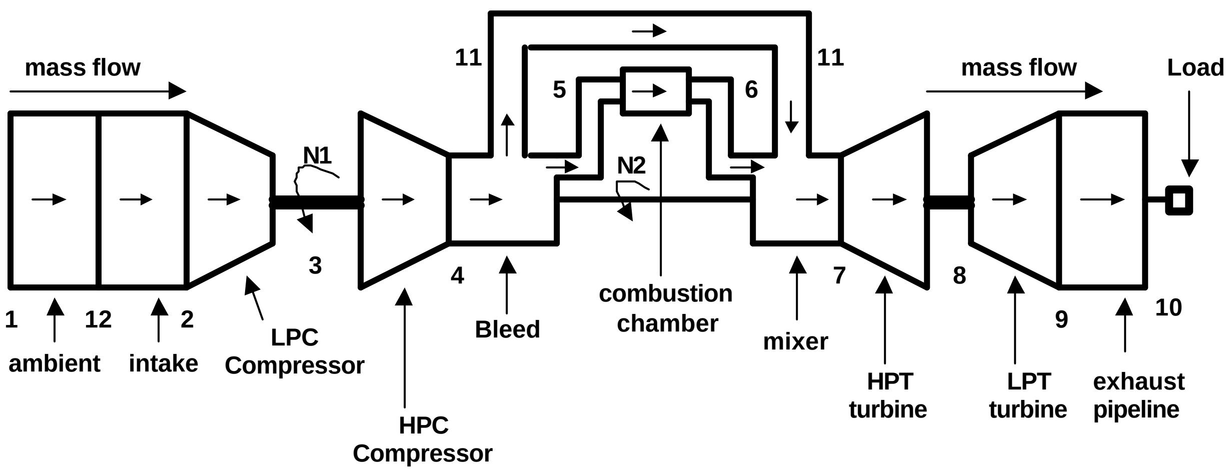


Figure 1. Model of a twin-shaft gas turbine for power generation - generator driven by the LP shaft at the turbine end

The gas turbine has two concentric and independent shafts. The external is the high pressure spool, linking the HPC and the HPT. The internal shaft is the low pressure spool, linking the LPC and LPT. Power is extracted from the LP shaft at the turbine end. High pressure air is bled for HPT cooling. The low pressure spool turns at N1 rpm and the high pressure spool at N2 rpm. The engine was divided into the following blocks: ambient, LPC (Low Pressure Compressor), HPC (High Pressure Compressor), bleed, combustion chamber, mixer, HPT (High Pressure Turbine), LPT (Low Pressure Turbine). The load is linked at the LPT turbine. Since the LPT turbine is directly coupled to the generator, N1 is fixed at 60 Hz or 3600 rpm.

Variable geometry compressors (VGCs) were used, with stator stagger settings varying in steps of $5^o$, from $-25^o$ to $+25^o$. To evaluate the number of variable stators needed, an auxiliary simulation was used to determine the speed at which the engine operating line would intersect the surge line. Restrictions were imposed on the maximum cycle temperature (combustion chamber outlet temperature), surge margin (safe operation) and compressor corrected speed (mechanical integrity).

## 3. Fixed geometry compressor

During the design of an axial flow compressor, the geometry is defined according to the specified mass flow, pressure ratio and efficiency, at which the efficiency is optimized as result of good blade alignment with the incoming flow. Since an axial flow compressor is made of several stages, and each stage behaves as a single compressor, the overall compressor characteristics are dependent on the characteristics of the individual stages and their matching. Therefore it is important that at each stage the blades are staggered correctly. At part speed, flow alignment is poor since air mass flow changes with the compressor speed and pressure with its speed squared. As consequence, density varies much more than the mass flow at the rear stages, causing them to choke and the front stages to stall. A remedy for this undesirable condition is the re-stagger of the blades in order to decrease de mass flow entering the compressor. Simulating the engine before it is manufactured would reduce cost and time of development, since major operating problems may be foreseen.

Figure (2) shows the LPC (fixed geometry) map to which is superimposed the engine operating line (shown in red), clearly indicating the need for surge margin improvement at part-load operation. Low-pressure compressor operates at constant speed N1. It is also shown the compressor efficiency variation at part-load, indicating small decrease in efficiency.

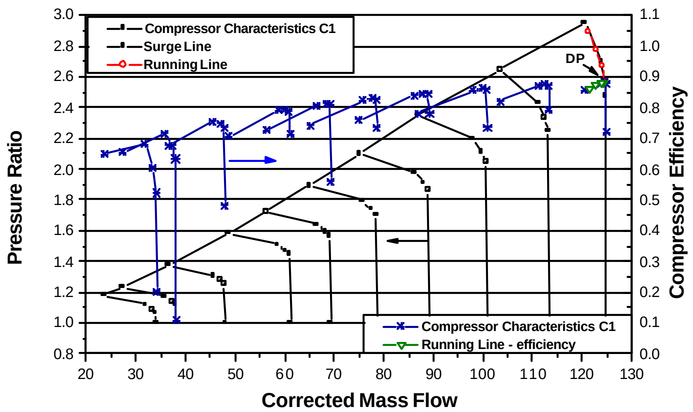


Figure 2. LPC characteristics and engine running line (shown in red)

The operating point on the low-pressure compressor map moves towards surge as load is decreased. If no remedial action is taken at part-load, the low-pressure compressor cannot operate because it may surge due to the inherent reduction of mass flow. It is the point where blade stall becomes so severe that the blading can no longer support the adverse pressure gradient, and with a lower pressure ratio rise now being produced the flow instantaneously breaks down (Walsh and Fletcher, 1998). When the compressor would be operating at stall condition, either the working line needs to be lowered or the surge line moved upwards. Actions may be taken like opening bleed valves or reducing fuel flow. The latter is unacceptable since it is fixed by the power demand. The solution would be moving upwards the surge line; therefore increasing surge margin, while the engine running line stays practically unaffected.

Figure (3) shows the high-pressure compressor characteristics, to which is superimposed the engine running line (shown in red). For the HPC it can be seen that operation at part-load is not critical because the running line is always far from surge line. For this study, surge margin at design point was set at 20%.

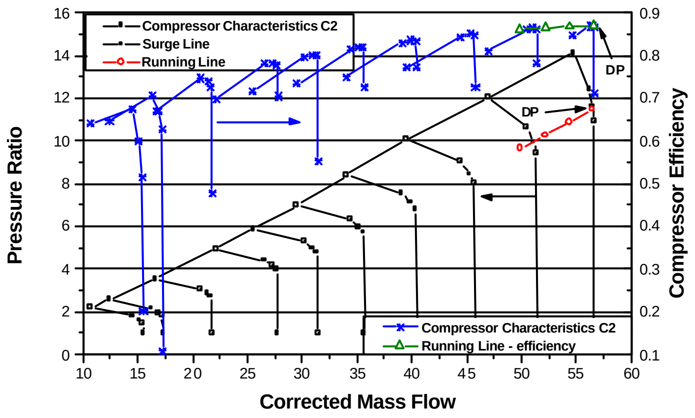


Figure 3. HPC characteristics and engine running line (shown in red)

## 4. Influence of variable geometry of the low pressure compressor on the performance parameters

Variable geometry is incorporated to the LPC in order to improve surge margin at part-load operation. Engine performance is shown in Fig. (4) to Fig. (6). The calculations were performed by GTAnalysis computer program. Intermediate results were interpolated using indigenous techniques and a specially developed computer program, written in FORTRAN. Surge margins of 10%, 15% and 20% were chosen for this study because these values may give indication of how the engine control must be, in order to accommodate sudden load changes. For each VSV setting design speed was decreased to the point that surge margin vanished. At this point the VSV was re-staggered in the direction of reducing mass flow (closing), from 0° to –25°, in steps of 5°.

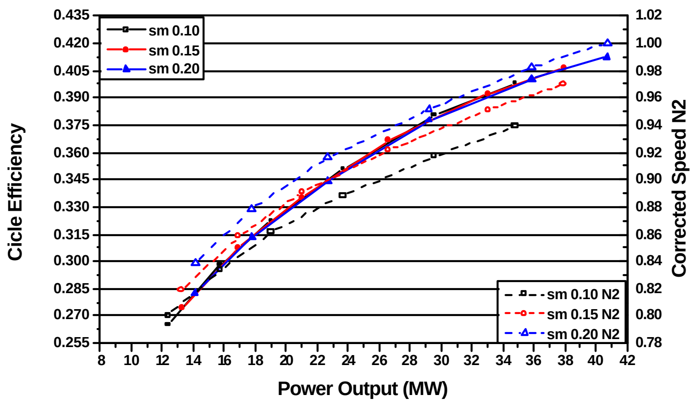


Figure 4. VG LPC influence on cycle efficiency

As can be seen from Fig. (4), the cycle efficiency (continuous line) stays unaffected at all off-design conditions, when VG LPC is used, independently of surge margin settings (dotted lines). At this same figure the corrected speed N2 is plotted, indicating how the HP spool would turn at part-load. Corrected speed is associated with mechanical integrity and must be kept within limits of available material, manufacturing and production technology.

The information is useful for the design of the engine control system, since they provide the needed data to control HP spool speed and HP blade settings. Final adjustments ought to be done at test bench, during engine performance development program.

Figure (5) shows the turbine inlet temperature (maximum cycle temperature). This parameter is of much importance because it represents one of the engine critical operating limits. Turbine inlet temperature is associated with cycle life, where cycle life (also known as low cycle fatigue life) is dictated by thermal stress levels; therefore the maximum cycle temperature never can be exceeded for long periods of time. In this work the maximum cycle temperature considered was the design point temperature (1530 K). It can also be seen that, for a given surge margin and power output, the maximum cycle temperature would vary about 40 K for all part-load excursions. The importance of the lowest maximum cycle temperature possible arises from engine life improvement.

Information like the ones shown in Fig. (5) may be used during engine control system design, with appropriate schedule of the VG stators.

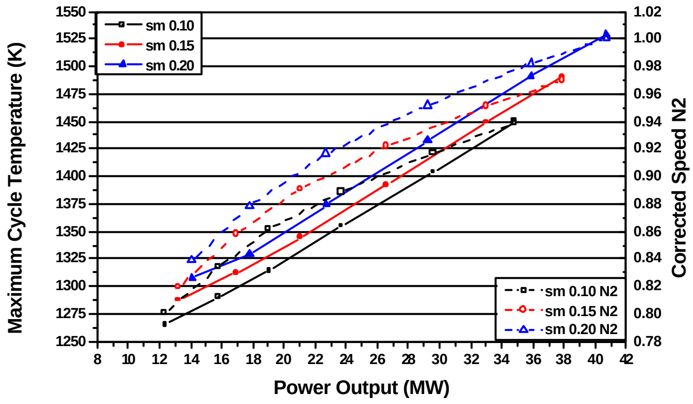


Figure 5. VG LPC influence on maximum cycle temperature

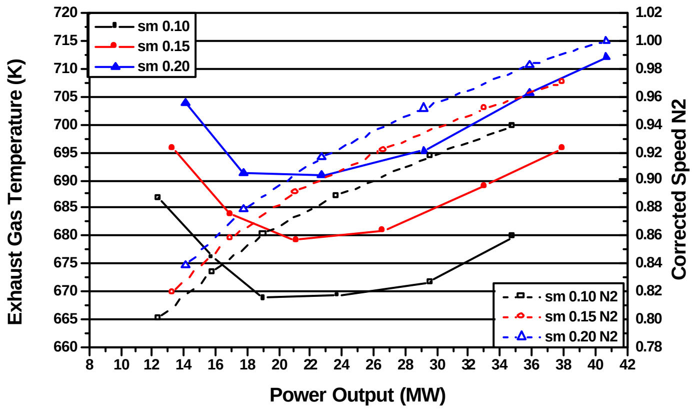


Figure 6. VG LPC influence on exhaust gas temperature

Figure (6) shows the exhaust gas temperature as another important parameter under analysis. Exhaust gas temperature is important when combined cycle gas turbine for cogeneration is used, although it is not focus of study in the present work. Increasing exhaust gas temperatures may be obtained at increasing surge margin settings for a given power, a feature that may be explored for cogeneration application.

Figure (7) shows the IGV/VSV scheduling for the LPC, for given surge margin and power output. It is important to notice that, for a given power and for each surge margin, the IGV/VSV scheduling is not the same. For a given power, the higher the surge margin the more the IGV/VSV must be closed (continuous lines). On the other hand, the higher the surge margin the higher the corrected speed (N2) lines.

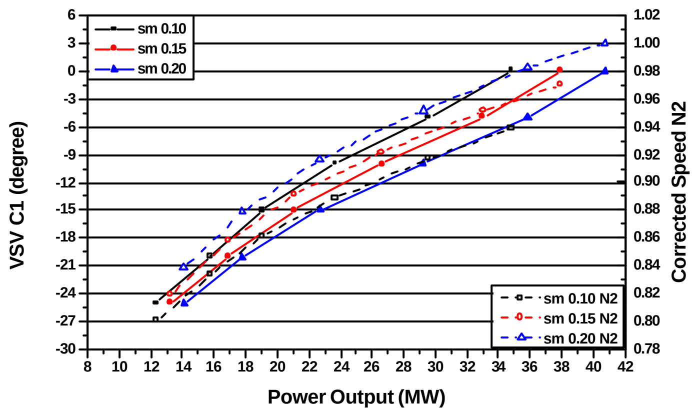


Figure 7. Variable geometry low-pressure compressor – VSV (Variable Stator Vane and/or IGV)

When VG LPC was incorporated to the engine, three different surge margin settings were studied. At the 10% setting, for a given power the smallest turbine inlet temperature was calculated. This would be a convenient control mode as far as the cycle life, associated if thermal stress level, is concerned, that is, the number of times that the engine is started, accelerated to full power, and eventually shut down, between overhauls (Walsh and Fletcher, 1998). If surge margin is set to 20%, one is in the safe operation side, easy to control engine transients. The operation of gas turbines is obviously dependent on such factors as the rotating group polar moment of inertia and the maximum temperature, which the turbine blades can withstand for short periods. Usually the limiting factor on acceleration it the proximity of the surge line to the equilibrium running line, and this is particularly critical at the start of acceleration from low powers (Cohen et al, 1996).

## 5. Influence of variable geometry of the low and high pressure compressors on the performance parameters

Variable geometry was incorporated to the low and high-pressure compressors as means to control important gas turbine performance parameters. It is shown in Fig. (8) that cycle efficiency is practically unaffected, at a given power and low-pressure compressor surge margin (sm1). However other important parameters may vary significantly.

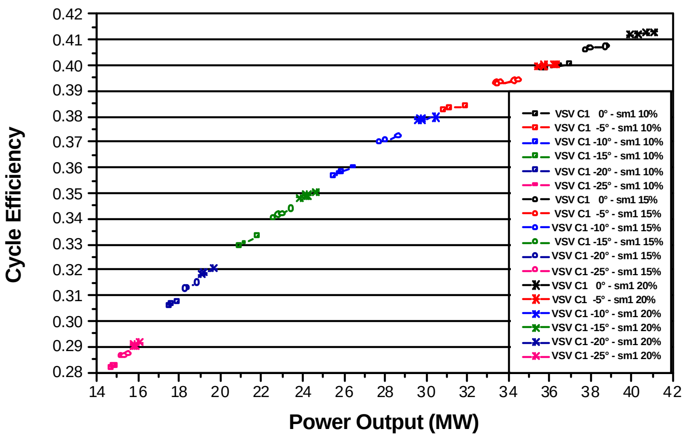


Figure 8. VG LPC and HPC influence on engine cycle efficiency

Figure (8) shows cycle efficiency plotted as function of power output, for three different LPC surge margins (sm1): 10%, 15% and 20%. These values were chosen having in mind the cycle life and LPC safe operation. Although it is not considered in this work the transient operation, surge margin levels can give indication of possible operation of the engine under sudden load variation.

## 6. Power generation application

The fundamental requirement is to maintain the safety of the engine, regardless of how the load is applied or how the inlet conditions (e.g. ambient conditions) are changing. The control system must ensure that the critical operating limits of rotational speed and turbine inlet temperature are never exceeded, and that compressor surge is avoided. One method must be used to provide the control system designer with information about the fuel flow required for steady-state operation over the entire range of operating conditions. If transient behavior is under study, its analysis can predict the maximum fuel flow, which can be used for acceleration without encountering surge or exceeding temperature limits (Cohen et al, 1996).

Figure (9) shows some important parameters when cycle performance (engine operation) is analyzed. The results shown are for fixed LPC surge margin at 10%. The power output at design was set as 40.725 MW and surge margin as 20% for both compressors. Design power output is not shown in Fig. (9) because surge margin set at 10% the operation would be impossible, due to the constraints imposed on maximum temperature and spool speed. Surge margin and corrected speed for high-pressure compressor are plotted as function of power output. Continuous lines for VG LPC and for HPC surge margins are indicated. Similarly, dashed lines represent VG PC and N2 HPC. These information are important for the engine control system design. On the curves are also shown the settings for the LPC and HPC. Horizontal lines (shown in magenta) indicate the HPC settings, with decrease of N2 and HPC surge margin (sm2). Vertical lines indicate the LPC settings for decreasing power. Interpolation from the calculates values for steps of $5^o$ in the VG was needed.

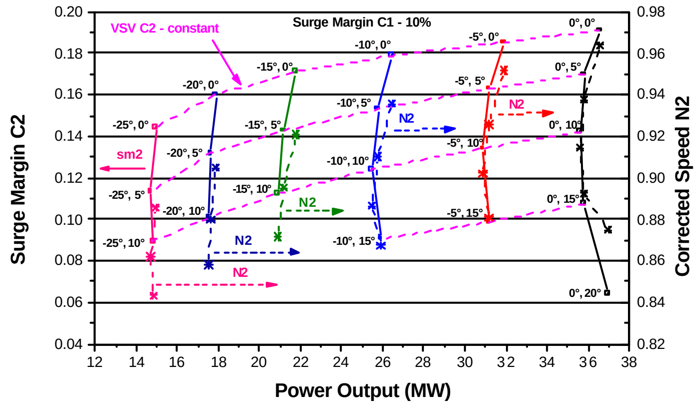


Figure 9. Influence of VG LPC and HPC on important performance parameters for sm1 set at 10%

Figure (10) has similarly been produced for surge margin of 20%, now set for both LPC and HPC, as indication of safe operation at part-loads.

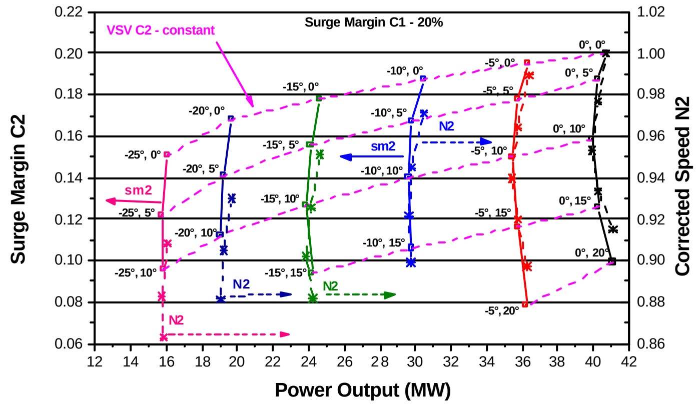


Figure 10. Influence of VG LPC and HPC on important performance parameters for sm1 set at 20%

Usually the VG accounts for the power output decrease from 100%, at the design point, to 50% at part-load. Lower loads would require bleed valve between LPC and HPC (Horner, 1994). Since in this work bleed valve between LPC and HPC compressors was not take into account, the calculations for powers lower than 20 MW (50% output power) were obtained and analyzed without bleed.

## 7. Conclusions

A very complex engine has been modeled and the engine performance calculations have been done using the GTAnalysis computer program. All key parameters were fully determined and vital information concerning engine safe operation, cycle life and control system design were determined. Means of improving part load performance, like VGC, were analyzed. A viable methodology for such complex analysis has been developed whose results conform with actual engine behavior, at least qualitatively. These results may be quantitatively correct provided more reliable engine data are available to support the components performance assumptions that had to be made.

An Aeroderivative engine that incorporates variable geometry in the form of compressor inlet guide vanes that direct air at the optimum flow angle, and variable stator vanes to insure ease of starting and smooth, efficient operation over the entire engine operating range (Horner, 1994) could be successfully modeled.

## 8. Acknowledgement

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