Proceedings of the 10th Brazilian Congress of Thermal Sciences and Engineering -- ENCIT 2004
Braz. Soc. of Mechanical Sciences and Engineering -- ABCM, Rio de Janeiro, Brazil, Nov. 29 -- Dec. 03, 2004

Paper CIT04-0089

PERFORMANCE STUDY OF A 1 MW GAS TURBINE

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Abstract. *The conception of a gas turbine of 1MW power generation is the aim of this work. Design parameters are obtained from cycle calculations based on target component performances appropriate for such an engine. Using compressor and turbine maps specially designed for the engine, the gas turbine overall performance is studied. Part-load performance is studied aiming at operation away from design at acceptable efficiency levels. Results from engine performance calculations are fed back to the compressor and turbine studies, in order to incorporate the engine overall performance needs.*

Keywords*. gas turbine, performance, power generation, turbine.*

1. Introduction

In this work attention is focused on industrial gas turbines for power generation. Distinction between 'aircraft gas turbine' and 'industrial gas turbine' has to be made for three main reasons: Firstly, the life required of an industrial plant without overhaul is much larger than that for aeronautical power plant. Secondly, limitation of the size and weight of an aircraft power plant is much more important than in most other applications. Thirdly, the aircraft power plant can make use of the kinetic energy of the gases leaving the turbine, whereas it is wasted in other types and consequently must be kept as low as possible.

When gas turbines were originally proposed for industrial applications, unit sizes tended to be 10 MW or less and, even with heat exchangers, the cycle efficiency was only about 28-29%. The availability of fully developed aircraft engines offered the attractive possibility of higher powers. The early aero-derivative engines, produced by substituting the exhaust nozzle for a power turbine, produced about 15 MW with a cycle efficiency of about 25 per cent. Modifications required included strengthening of the bearings, changes to the combustion system to enable it to burn natural gas or diesel fuel, the addition of a power turbine and a de-rating of the engine for longer life. In some cases a reduction gearbox was required to match the power turbine speed to that of the driven load, like a marine propeller. For other types of load, such as alternators or pipeline compressors, the power turbine could be designed to drive the load directly.

The use of gas turbines for electrical power generation has changed dramatically in recent years. In the 1970s, gas turbines were primarily used for peaking and emergency applications; aero-derivative units with a heavy-duty power turbine were widely used. One of the outstanding advantages of this engine type was its ability to produce full power from cold in under two minutes, although this capability should be used only for emergencies because thermal shock would greatly reduce the time between overhauls. The aero-derivative units had a maximum rating of about 35MW, their efficiency was about 28 per cent and they burned expensive fuel so they were not considered for base load applications.

In many countries electric power generation is not possible by hydroelectric power plants generation. Gas turbines are then the main source of electric power generation. On these days most attention is placed on the development of gas turbines for power generation, not only to match the demand for new resources but also to make them more efficient to respond to the increasing fuel costs.

Great effort has been made around the world to improve gas turbine performance. The effort is concentrated mainly on advanced materials research, such as composite ceramics and thermal barrier coatings, to increase maximum cycle temperature, therefore improving cycle efficiency. In addition, low emissions technologies research and development will improve the combustion system by greatly reducing the NOx and CO produced without negatively impacting turbine performance. NOx emissions levels must be kept below regulating limits due to the environmental impact. Gas turbines are incorporating new technologies in the design of the combustions systems, what make them conform to those limits.

Gas turbines may be used for emergency applications, due to high reliability and quick starting, characteristics not found in other engines. For a given power output the gas turbines are more compact than piston engines, a characteristic that make gas turbines appropriate for installation in buildings. Other advantages of using gas turbines are the absence of vibration and attenuable noise.

The Brazilian short-term needs for gas turbine based power plants is mainly due to the generation capacity has been kept constant for many years while the power consumption has been increasing steadily. Several power plants are being considered so that in the near future many gas turbines will be continuously operating. It is not always possible to operate gas turbines at full load, conditions at which they are most efficient. At part-load the efficiency is always lower so that a means to increase their part-load performance would significantly contribute to lower the costs of electricity (May et al, 1976; Rahnke, 1969; Hourmouziadis et al, 1976; Bringhenti and Barbosa, 2001, 2002 a, b and c).

In a gas turbine the process of compression, combustion and expansion does not occur in the same component, like in piston engines. Therefore, these components may be designed, built, tested and developed isolated, later assembled in a convenient manner to form a gas turbine. The main components that form a gas turbine are compressor, combustion chamber and turbine. Many types of gas turbines may be assembled by a combination of several compressors and turbines.

Gas turbine performance is affected, mainly, by loss in general (friction, flow) and limitation imposed by material development. In general, the higher the combustion chamber outlet temperature the higher the gas turbine efficiency.

The development of gas turbine was difficult due to the lower efficiency of compressors and turbines. These low efficiencies were due to the lack of the best knowledge of the flow in these components. The development of gas turbines was only possible with the application of the aerodynamic concepts in the components design. The great problem for the gas turbine production was the huge difficulty to get compressors working efficiently in high pressures.

To take into account performance losses in a gas turbine is essential that, during the design, the performance of all possible points of operation were carefully evaluated. This performance evaluation may be done using performance information of gas turbine main components (compressor, combustion chamber, turbine, etc.).The performance calculation procedure at off-design point are the same used at the design point, except that the operating points of each components are unknown beforehand (efficiencies, flows, rotations, etc.), requiring iterative calculations, for which are required the performance characteristics of the components, usually compressor map, combustion chamber map, turbine map and other relations that simulate the component performance at desired conditions. Requiring, therefore, an enormous quantity of calculation that will be more convenient to use computational programs for automatically do it (Stamatis et al, 1990; Barbosa and Bringhenti, 1999 and 2000; Cohen et al, 1996).

Gas turbines in the power range of 1 MW are compact, lightweight, with quick starting capability, and simple to operate. With these characteristics these gas turbines may be used widely in industries, universities, colleges, hospitals and commercial buildings to produce electricity, heat or steam.

This work deals with performance calculation at off-design point of a gas turbine in the range of 1 MW for power generation application using the GTAnalysis computer program. GTAnalysis is a specially written computer program, using the FORTRAN language, to simulate at steady state almost all gas turbines existing in the market. For off-design performance calculation, GTAnalysis has a library of compressor, combustion chamber and turbine maps (Bringhenti, 1999 and 2003). In this work specially designed compressor and turbine maps are used, which are generated by other specially developed computer programs (Tomita and Barbosa, 2004; Bringhenti et al, 2001). Those programs are able to design compressors and turbines and generate their performance maps.

GTAnalysis is a modular computer program that can easily accommodate any component performance map, either produced from test bench or calculated theoretically. Due to the modular characteristic, any required modification can be incorporated to the program, making it very friendly.

The results shown in the figures presented in this work indicate the main parameters that need to be monitored during the off-design performance.

The gas turbine studied in this work is an industrial engine. Its gas generator was designed to fit also a turbojet in the range of 5 kN thrust (Bringhenti and Barbosa, 2004).

2. Main gas turbine parameters

A computer program that numerically simulates the steady state performance of gas turbines was used to analyze their part-load performance. The program is based on engine functional blocks build-up, that easily model almost all gas turbines, numbered according to Fig. (1).

For this work, it is required to design an industrial gas turbine in the power range of 1 MW that would be adequate for distributed power generation. It is also desirable that its gas generator could be used in a 5 kN turbojet engine. Performance targets were chosen based on achievable technology, as shown in Tab. (1). The procedure developed in this work may be also used after component testing results become available in order to achieve quantitatively better results.

Table 1. Reference engine design point characteristics

parameters	values
mass flow (kg/s)	8.104
compressor pressure ratio	5.0
maximum cycle temperature (K)	1173.0
shaft output (MW)	1.2215
cycle efficiency	0.194
Compressor isentropic efficiency	0.85
combustor chamber pressure loss	0.05
combustion chamber efficiency	0.99
gas generator turbine isentropic efficiency	0.85
mechanical efficiency gas generator shaft	0.99
exhaust gas temperature (K)	859
isentropic efficiency of free power turbine	0.85
mechanical efficiency of free power turbine shaft	0.99

The engine is a single-shaft gas generator, free power turbine, as sketched in Fig. (1).

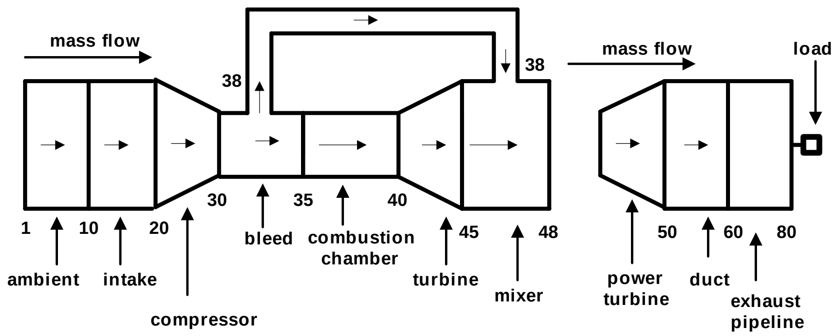

Figure 1. Sketch of a single shaft free power turbine unit

The mains engine functional blocks used for this analysis were: ambient, intake, compressor, combustion chamber, turbine, power turbine, and exhaust pipe. The compressor spool turns at N1 rpm at design. At part-load N1 is lowered from 100% down to the point at which the surge margin vanishes. The power turbine is directly coupled to the generator, so that speed is fixed at 60Hz or 3600 rpm. For each chosen off-design point all thermodynamic parameters were calculated, from which a selection of appropriate data was taken to produce the graphs shown.

The main performance parameters that impose restrictions on engine operations, considered in this work, were: maximum cycle temperature or combustion chamber outlet temperature (due to material limits), surge margin (safe operation or surge-free operation) and compressor corrected speed (mechanical integrity due to the stresses caused by rotational speed).

Cycle efficiency, maximum cycle temperature, mass flow, N1, pressure ratio and surge margin were monitored during the calculations, because they are important for engine handling and life of its components.

The engine is required to operate off-design due to load variation. Performance deteriorates because at off-design the components operate at regions of lower efficiencies, caused by bad component matching. The bad-matching results from the passage areas, calculated at design point conditions, not being enough to accommodate the flow at those operating conditions.

To improve off-design efficiency it is usually adopted the flow control by means of variable stator geometry (to improve matching of the components). In this work, variable geometry would not be used, since one seeks simple and cheap engine, but the results shown that variable geometry compressor have to be incorporated to increase part-load safe operation (Bringhenti and Barbosa, 2003).

Surge margin at design point was set to 21% and was monitored, during the calculations, at all part-load conditions. Maximum cycle temperature was set to 1173 K at design point to avoid blade cooling.

3. Results

The gas turbine model studied in this work is for power generation application, so that the power turbine must run at constant speed because generator works at either 50 Hz or 60 Hz. At part-load operation only the gas generator speed is changed for the engine with the free power turbine layout.

The gas generator used in this gas turbine model was carefully designed. It must attend the restrictions imposed by cycle performance and thrust required in a turbojet application (Bringhenti and Barbosa, 2004). The compressor need to be designed and tested, therefore, the power absorbed by the compressor cannot exceed 1.50 MW because this is the test bench capacity intended to be built in Brazil.

The surge margin used in this work is given by Eq. (1),

$$sm = \frac{\Pr\_surge - \Pr\_design}{\Pr\_surge - \Pr\_choke} \tag{1}$$

Where: Pr_surge is the pressure ratio at surge point; Pr_design is the pressure ratio chosen at design point; and Pr_choke is the pressure ratio at choke point. These points can be seen in Fig. (2).

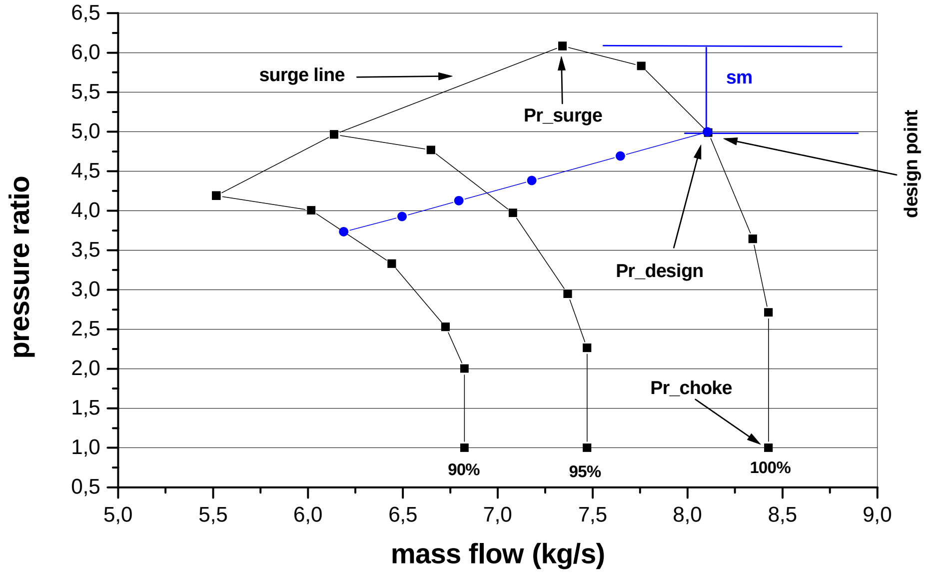

Figure 2. Compressor characteristics – surge margin definition

Part-load conditions were obtained varying N1 from 100% (relative to design) down to the point at which the surge margin vanishes, maintaining the power turbine speed constant. Figure (3) through Fig. (9) show the simulation results at several off-design conditions varying gas generator speed N1. In this study the main cycle parameters were analyzed, such as: stagnation temperatures, pressure ratio, surge margin, air mass flow, power output, isentropic efficiency of compressor and cycle efficiency.

Figure (3) shows the specially designed compressor maps for this application. For the sake of space only two designed compressor maps are shown in Fig. (3). In magenta, the first designed compressor map without variable inlet guide vane (IGV) and in black, the last designed compressor map with variable IGV. In this figure are shown the running lines, surge margins and design point for these two compressors.

The chosen design point for the compressor was defined by Bringhenti and Barbosa (2004). In that work, the gas generator was designed to attend either a turbojet or turboshaft performance characteristics with low development costs. To attend this requirement, IGV was not incorporated in the first design compressor.

The designed compressor, as well as the performance calculation, generated maps, was developed with an auxiliary computer program (Tomita and Barbosa, 2004). The designed compressor has 5 stages and pressure ratio of 5. Without variable IGV the surge margin at design point was set as 15%. Incorporating variable IGV in compressor, for same pressure ratio and stage number, surge margin may be set as 21%. Walsh and Fletcher (1998) suggest that required surge margin for power generation application is 15-20%, and is dependent upon acceleration and deceleration times required, engine configuration, whether centrifugal or axial compressors are applied, whether bleed valves or variable stator vanes are employed at part-load, etc.

The specially developed compressor performance map was incorporated into GTAnalysis computer program. In Fig. (3), it can be seen that at part-load operation the running line, without IGV, intercepts the surge line at approximately 88% N1, relative to design. When variable IGV is used this value can be decreased down to 74% N1, relative to design. For this engine configuration and application the use of variable IGV must be incorporated in the compressor for safest operation at part-load. Instead of variable IGV to solve the surge margin problem at part-load operation, a blow-off valve could be used as an alternative method. In this method air is bled from some intermediate stage of the compressor. Blow-off clearly involves a waste of turbine work, so that the blow-off valve must be designed to operate only over the essential part of the running range.

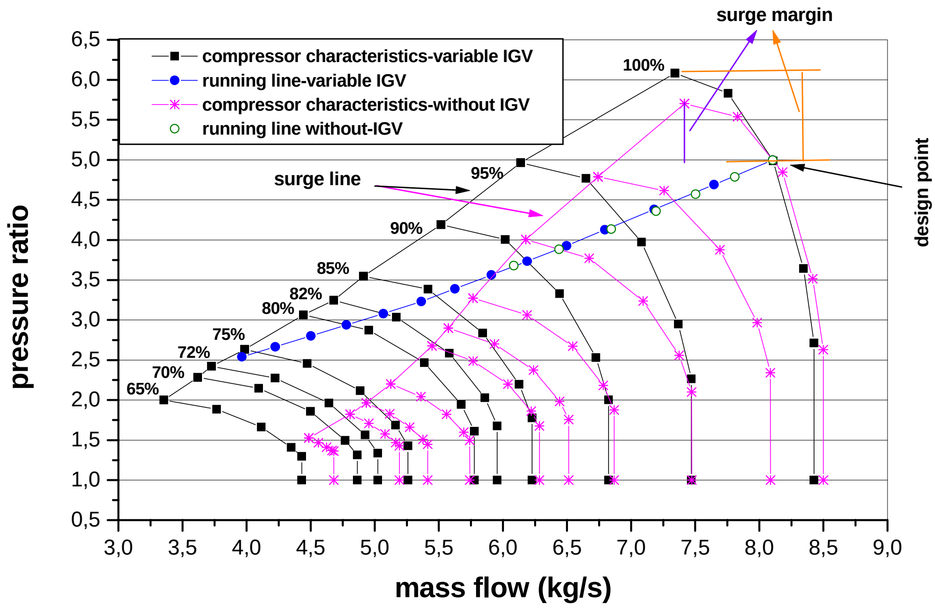

Figure 3. Compressor characteristics – pressure ratio

Table (2) shows the required IGV angles for each corrected speed, relative to design. The compressor characteristics shown in Fig. (3) and Fig. (4) have 10 constant corrected speed curves, and the values of each constant corrected speed are shown in Tab. (2). The minus sign indicate that the IGV are closing, decreasing the blade passage area.

Table 2. Compressor IGV angles

%N1	100	95	90	85	82	80	75	72	70	65
IGV (°)	0	-15	-20	-25	-25	-25	-30	-30	-30	-30

Figure (4) shows the isentropic efficiency of the two developed compressors. The isentropic efficiency chosen for these two compressors at design point are shown on the figure, 85%. When variable IGV is used, the isentropic efficiency for each constant corrected speed can be maintained at highest value at part-load. This characteristic cannot be observed without IGV, where the isentropic efficiency for each constant corrected speed falls significantly, and indicating bad part-load performance. For this particular application and engine configuration the isentropic efficiency of compressor stayed practically unaffected, so the use of variable IGV was very important in improving safe operation and increasing the range of part-load operation.

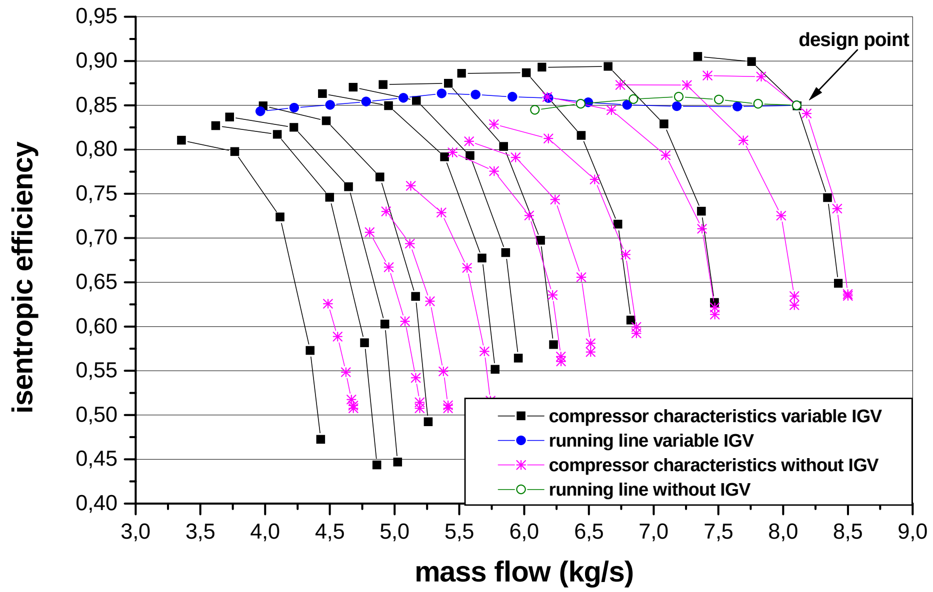

Figure 4. Compressor characteristics – isentropic efficiency

The Fig. (5) to Fig. (9) show the performance calculations using compressor map with variable IGV. The main control parameters are plotted in these figures. Figure (5) shows how cycle efficiency is affected at part-load condition, in blue on the left hand side. In the same way the power output decrease is shown, in black on the right hand side. The specified design point is show in the red circle. Cycle efficiency can be improved at part-load by using variable geometry in the power turbine as discussed in the works by Bringhenti and Barbosa (2004 b, 2003 and 2002 c), Sirinoglou (1992) and Roy-Aikins (1988, 1990). One manner to improve cycle efficiency at design point is permitting an increase in maximum cycle temperature. In this work maximum cycle temperature was limited at 1173 K to avoid blade cooling, lowering the development costs.

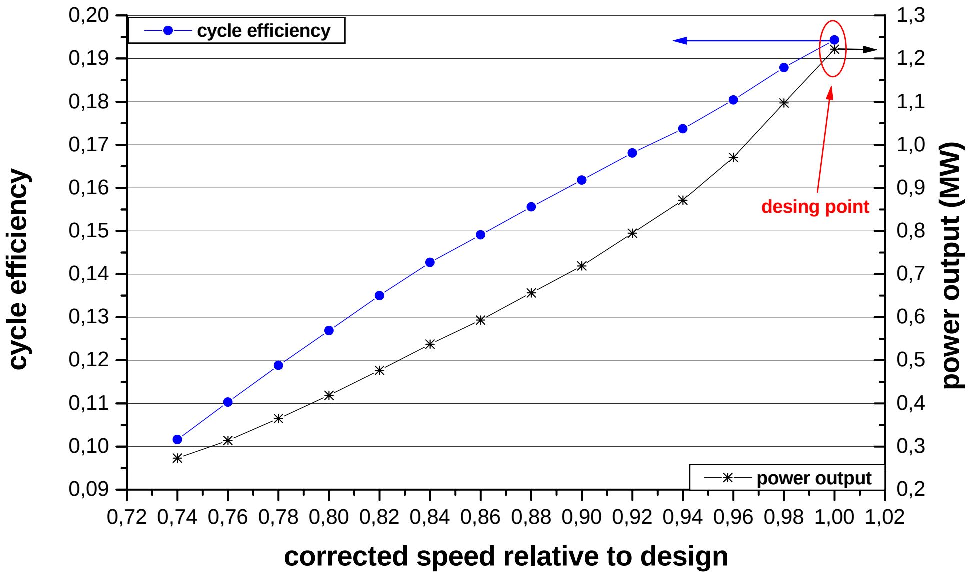

Figure 5. Cycle efficiency

Figure (6) shows the surge margin at design point, in the circle in red, and the surge margin at part-load, in blue on the left hand side. For each corrected speed is possible to find a surge margin value and the corresponding power output, in black on right hand side. At design point the surge margin was set as 21%, this value was in agreement with Walsh and Fletcher (1998) quoted above. As can be seen in Fig. (6) the surge margin value at 74% N1 almost vanished, the power output at this value correspond to 24.6%, relative to design point. Generally, the minimum required power output at part-load operation corresponds to 50% power output, relative to design point. The power output that corresponds to 50%, relative to design point, is 0.61 MW and the surge margin value correspondent to this power output is approximately 8%. Therefore, part-load operation is possible with a well designed system control; the engine can operate safety with 8% surge margin. The starting operation is out of the scope in this work and will not be discussed.

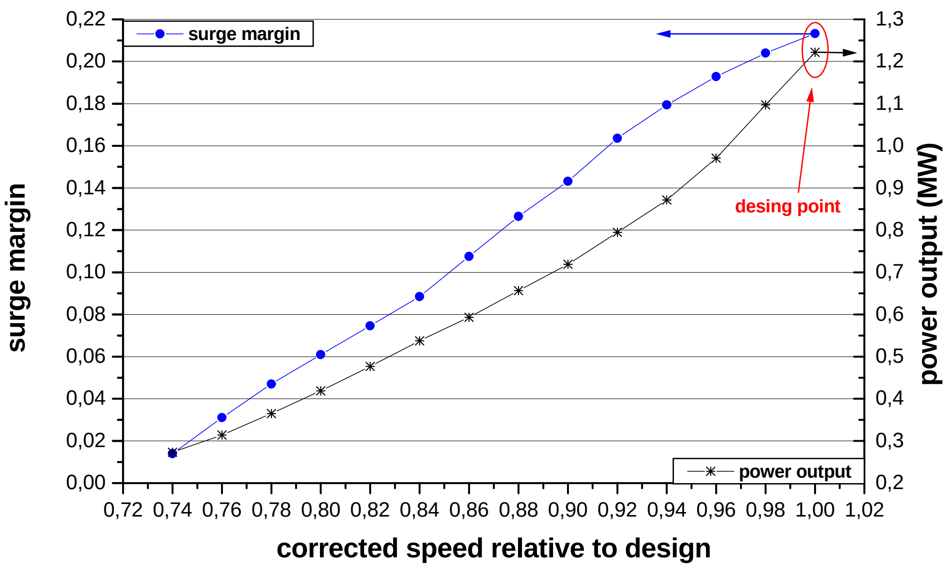

Figure 6. Surge margin

Figure (7) shows pressure ratio and power output set at design point, in the red circle, as well as pressure ratio and power output variation at off-design conditions. Pressure ratio and power output at design point were chosen beforehand by Bringhenti and Barbosa (2004). At design point the pressure ratio was set 5 and power output was set 1.2215 MW. At off-design, for each chosen corrected speed, the corresponding pressure ratio can be found at the left hand side in Fig. (7), blue curve. The corresponding power output can be found on the right hand side in the same figure, black curve.

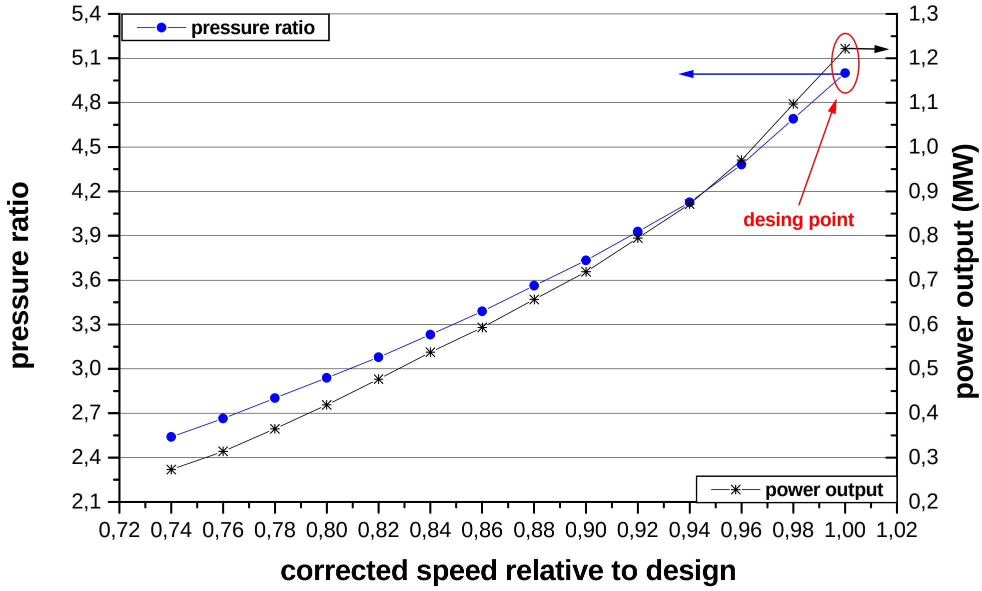

Figure 7. Pressure ratio

Figure (8) shows air mass flow and power output set at design point, in the red circle, as well as air mass flow and power output variation at off-design conditions. Air mass flow and power output at design point were chosen beforehand by Bringhenti and Barbosa (2004), the air mass flow is the same used in a turbojet engine, to attend the required thrust of 5 kN. At design point the air mass flow was set 8.104 kg/s. At off-design, for each chosen corrected speed the corresponding air mass flow can be found at the left hand side in Fig. (8), blue curve.

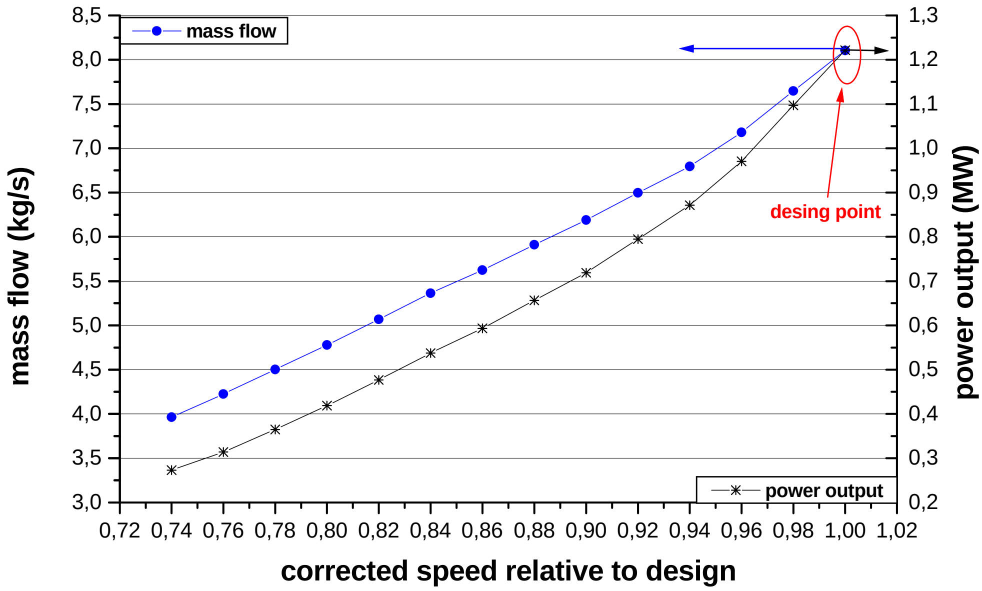

Figure 8. Air mass flow

Figure (9) shows maximum cycle temperature and power output set at design point, in the red circle, as well as the stagnation temperatures and power output variation at off-design conditions. At design point maximum cycle temperature, station 40, was set 1173 K.

At off-design, for each chosen corrected speed the stagnation temperatures can be found at the left hand side in Fig. (9), four curves are shows, each one corresponding to different temperatures. Turbine inlet temperature, blue curve, corresponds to the maximum cycle temperature, station 40. Turbine exit temperature, magenta curve, corresponds to gas generator exit temperature, station 45. Power turbine inlet temperature, red curve, corresponds to the temperature at station 48. Power turbine exit temperature, dark cyan curve, corresponds to the temperature at station 50. Power output can be found on the right hand side in the same figure, black curve. Exhaust gas temperature may be a control parameter at off-design when combined cycle or cogeneration application is used; this type of control is out of the scope of this work.

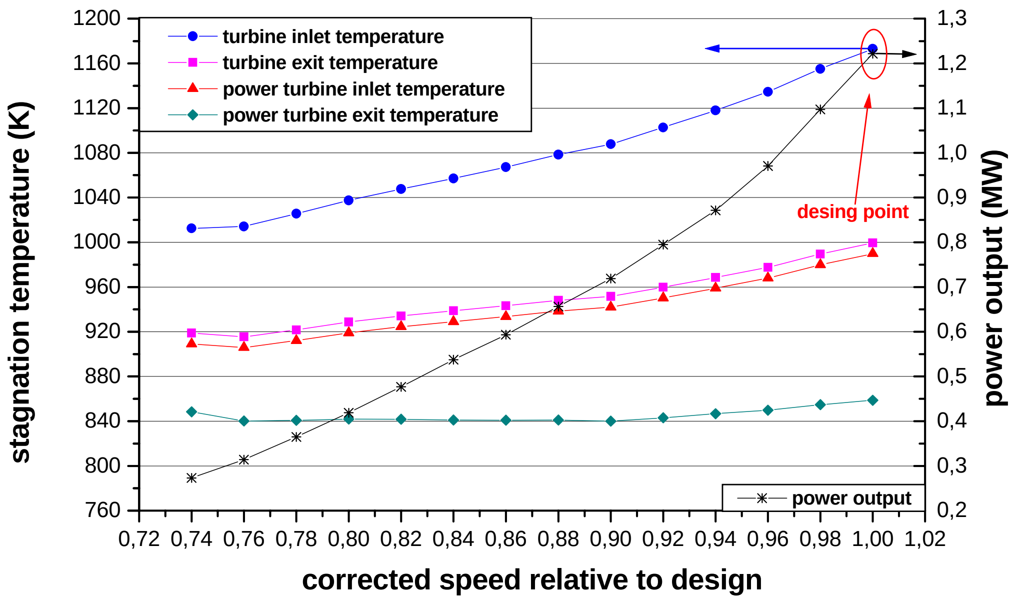

Figure 9. Stagnation temperatures

4. Conclusions

GTAnalysis is a computer program that describes engine by blocks, it has been used in this work to simulate a single-shaft free power turbine and the engine performance calculations are shown on Fig. (3) to Fig. (9). Specially developed maps were generated by external computer programs and incorporated into GTAnalysis, proving that, due to its modular characteristics, to incorporate designed maps for a given engine is easy.

One of the aims of the gas generator design is to lower the development costs. To comply with this requirement, in the first design, it was determined that the maximum cycle temperature was limited at 1173 K, avoiding blade cooling. Five stages compressor without variable geometry and pressure ratio of 5 was used. The results showed that, to increase the range of part-load operation, with safety, is necessary to incorporate on the compressor design variable IGV, as shown on Fig. (3) and on Tab. (2). Incorporating variable IGV in compressor the power output at part-load may be decreased down to 0.61 MW, which corresponds to 50% power output at design point, with a surge margin of 8%.

The gas generator was firstly designed to attend the thrust required for a turbojet engine with 5 kN, and the same gas generator was used for the power generation application in a single-shaft free power turbine. At the gas generator design some important parameters were taken into account to lower the development costs and test bench capacity. The power absorbed by the compressor can not exceed 1.5 MW.

5. Acknowledgements

The authors would like to thank El Paso Rio Claro Ltd. and ANEEL for their support. This research work was developed at *Centro El Paso de Referência em Turbinas a Gás*, ITA, and was possible due to the Brazilian federal law 9991/2000.

6. References

Bringhenti, C., and Barbosa, J.R. 2004, “Estudo de Um Gerador de Gases para Utilização em um Turboeixo e em um Turbojato”, proceedings of CONEM 2004, paper 41042.

Bringhenti, C., and Barbosa, J.R. 2004 b, “Part-Load Versus Downrated Industrial Gas Turbine Performance”, proceedings of ASME TURBO EXPO 2004, Power for Land, Sea, and Air, GT2004-54147, June 14-17, Austria Center Vienna, Vienna, Austria.

Bringhenti, C., and Barbosa, J.R. 2003, “Analysis of Gas Turbine Off-Design Safe Operation Using Variable Geometry Compressor”, proceedings of the COBEM, COBEM2003-0230, São Paulo– SP, November 10-14, Brazil.

Bringhenti, C., and Barbosa, J.R. 2002 a, “Effects Of Variable-Area turbines Stators Over The Important Parameters Of Gas Turbine Performance”, proceedings of the CONEM, CPB0272, João Pessoa – PB, Brazil.

Bringhenti, C., and Barbosa, J.R. 2002 b, “Performance Evaluation Of Different Types Of Gas Turbines For Power Generation”, proceedings of the CONEM, CPB0292, João Pessoa – PB, Brazil.

Bringhenti, C., and Barbosa, J.R. 2002 c, “Study Of An Industrial Gas Turbine With Turbine Stators Variable Geometry”, proceedings of the ENCIT, CIT02-0885, Caxambu – MG, Brazil.

Bringhenti, C. 2003, “Variable Geometry Gas Turbine Performance Analysis”, Ph. D. Thesis, ITA, Brazil.

Bringhenti, C., and Barbosa, J.R., 2001, “An Overview Of Variable Geometry Gas Turbines”, Proceedings of the COBEM, Energy and Thermal Systems, Uberlândia – MG, Brazil, Vol. 4, pp. 97-105.

Bringhenti, C., Barbosa, J.R., and Carneiro, H.F. de França Mendes, 2001, “Variable Geometry Turbine Performance Maps For The Variable Geometry Gas Turbines”, Proceedings of the COBEM, Energy and Thermal Systems, Uberlândia – MG, Brazil, Vol. 4, pp. 87-96.

Barbosa, J.R., Bringhenti, C., 2000, “Simulação Numérica em Sala de Aula”, CONEM, paper EC8840, Ensino de Engenharia Mecânica, pp. 2-9, Natal - RN, Brasil.

Barbosa, J. R., Bringhenti, C., 1999, “Simulação Numérica do Desempenho de Turbinas a Gás”, Proceedings of COBEM 1999, Engenharia Aeroespacial S8, Águas de Lindóia - SP, 22 a 26 de novembro, Brazil.

Bringhenti, C. 1999, “Analysis of Steady State Gas Turbine Performance”, M. Sc. Thesis, ITA, Brazil (In Portuguese).

Cohen, H., Rogers, G.F.C., and Saravanamuttoo, H.I.H., 1996, “Gas Turbine Theory”, 4th Edition, Longman Group Limited.

Hourmouziadis, J., Hogemeister, K., Rodemacher, O., and Kolben, H., September 1976, “Experience With a One Stage Variable Geometry Axial Turbine”, Session IV, Ref. 31, AGARD-CP-205.

May Jr., R. J., Tall, W. A., and Bush, H. I., 1976, “Potential Improvements in Engine Performance Using a Variable Geometry Turbine”, AGARD, cpp-205.

Rahnke, C. J., 1969, “The Variable Geometry Power Turbine”, International Automotive Engineering Congress, Detroit, Society of Automotive Engineers.

Roy-Aikins, J. E. A., 1988, “A study of variable geometry in advanced gas turbines”, Ph. D. Thesis, Cranfield Institute of Technology.

Roy-Aikins, J. E. A., 1990, “Considerations for the Use of Variable Geometry in Gas Turbines, ASME, 90-GT-271.

Sirinoglou, Alexander A. 1992 “Implementation of Variable Geometry for Gas Turbine Performance Simulation Turbomatch Improvement”, Cranfield Institute of Technology.

Stamatis, A., Mathioudakis, K., and Papailiou, K. D., 1990, “Adaptative Simulation of Gas Turbine Performance”, Transactions of the ASME, v.112, p. 68-175.

Tomita, J. T., and Barbosa, J. R., 2004, “Design and Analysis of an Axial Flow Compressor for a 1 MW Gas Turbine”, proceedings of the ENCIT, CIT04-0062, Rio de Janeiro – RJ, November 29 - December 03, Brazil.

Walsh, P. P., and Fletcher, P., 1998, “Gas Turbine Performance”, Blackwell Science.