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Industrial Twin-Shaft Gas Turbine Thermodynmic Modeling for Power Generation Application at Design Point and Off-Design Condition

Rashidzadeh-Hiva[1], Hosseinalipour-Seyed Mostafa[2]* and Mohammadzadeh-Alireza[1]

[1]Department of Mechanical and Aerospace Engineering, Tehran Science and Research Branch, Islamic Azad University, Tehran, Iran
[2] Department of Mechanical Engineering, Iran University of Science and Technology, Tehran, Iran

Abstract

Background/Objectives: The performance of twin-shaft gas turbine for power generation applications has been studied under different environmental conditions. This paper presents a twin-shaft cooled gas turbine design and off-design model. **Methods/Statistical Analysis:** A complete analysis was conducted for components integration and matching of the studies gas turbine at base and part loads conditions. The required characteristic curves of each component were produced in a separate study using full three-dimensional numerical simulations validated with available experimental data. A computer code in visual FORTRAN language has been developed which was capable of calculating the thermodynamic properties at all sections of the gas turbine engine. **Findings:** The proposed model was used to simulate the steady state behavior of power generation application of the gas turbine to study the effects of different parameters such as ambient conditions and loads. The code is completely capable of predicting the engine performance both in design and off-design conditions. The results of simulation are presented at the part load and different ambient conditions and are validated with test measurement data. The results show good agreement between the experimental data and analytic model. **Application/Improvements:** The model can be adapted for any single and twin-shaft industrial gas turbine using appropriate characteristic curves for each separate engine.

Keywords: Design Point, Off Design Condition, Twin-Shaft Industrial Gas Turbine, Zero Dimensional Model

Nomenclature

CFD	= Computational Fluid Dynamic	**Subscripts**	
Eff	= Efficiency (%)	0	= Ambient Condition
GG	= Gas Generator	Cc	= Combustion chamber
h	= Height from Sea Level (m)	Comp	= Compressor
HP	= High Pressure	D	= Design

IGV	= Inlet Guide Vane	Dpin	= Air Intake Pressure Loss
K	= Coefficient	Dpout	= Exhaust Pressure Loss
LHV	= Lower Heating Value (kJ/kg)	Ex	= Exhaust
	= Mass Flow Rate (kg/sec)	F	= Fuel
MW	= Mega Watt	Gen	= Generator
N	= Speed (rpm)	Gear	= Gearbox
NGG	= Gas Generator speed (rpm)	In	= Inlet

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*Author for correspondence

NPT	= Power Turbine speed rpm)	Loss	= Difference Between Inlet And Outlet
P0	= Total Pressure (bar)	Meas	= Measurement
PR	= Pressure Ratio	p-f	= Corrected for Fuel Type
PT	= Power Turbine	Th	= Thermal
ΔP	= Pressure Loss (bar)	Turb	= Turbine
T0	= Ambient Temperature (°C(	Ref	= Reference
T	= Temperature (K)	RH	= Relative Humidity
TIT	= Turbine Inlet Temperature (K)	Sh	= Shaft
TET	= Turbine Exit Temperature (K)	sh-c	= Shaft Corrected
η	= Efficiency (%)		
W	= Power (KW)		

1. Introduction

The electrical power plants are benefited from a vast range of different industrial gas turbines in simple or combined cycles. Mathematical models are gained considerable attentions due to their ability for simulating of gas turbine operations under different steady state and transient conditions. This paper presents a part of results produced by a developed steady state model which has been conducted for a twin-shaft gas turbine with power generation applications. The developed model in this paper has the capability of analyzing the gas turbine behavior in both design and off-design conditions for different parts of the system. In this study, detailed and exact results of the effects of the different parameters on the gas turbine performance have been presented. These results have been generated in the steady state condition with considering the control philosophy of the gas turbine operation. The control philosophy of the gas turbine based on the real control logic correlations is considered in this paper.

The steady state modeling of a gas turbine is focused on the gas turbine operation on different loads as well as the diverse control philosophies. These types of mathematical models enable the designer to calculate the required heat flow, Inlet Guide Vane (IGV) position and the mass flow rates of the bleed valves at different fuel compositions and environmental conditions with different control philosophies. The analysis of the main components conditions provides a good insight for deriving the best control philosophy for a specific condition in order to have an optimized operation for all components. Most of the previous researches were focused only on the design point operation models. There are also some off-design models reported by the researchers. On 1975, NASA developed a model “DYNGEN” for analyzing the operation of turbofan and turbojet engines on both the design and off-design conditions[1]. Waters and his colleagues developed a model “EPRI-GATE” for performance analysis of power cycle[2], which was later improved by Giglio via considering the effect of cooling process in the previous models[3]. El-Masri and his colleagues developed a more detailed model considering the blade cooling[4,5]. El-Masri also developed some models based on the second thermodynamic law and cycle performance analysis with cooling which finally resulted in the GASCAN computer code[6–9]. Other zero dimensional models were developed for thermal analysis by Ismail and Bhinder in 1991[10] as well as Zhu and Saravanamutto in 1992[11]. Jeoung Yoon-Su and his colleagues proposed the system for solar power generation resources in 2015[12].

2. Analytical Modeling

This research is focused on an industrial twin-shaft cooled gas turbine whose schematic view with power generation application is shown in Figure 1. As shown in this figure, the considered gas turbine is composed of a 10-stage air compressor with Inlet Guide Vane (IGV) at the inlet for high performance at off-design conditions; meanwhile they also provide the required safe margin from the surge line of the compressor at speeds lower than the design speed. The bleed valves which are located after 2nd and 5th stages have an important role for controlling the safe margin of the air compressor. The positions of these valves are usually determined based on the shaft speed. Also the flow coefficients of these valves have a vital role in the simulation. These coefficients have been derived for different positions considering the different pressure on both sides of the valves as well as the risk of the choking in flow. Having these coefficients in hand it is possible to calculate the air mass flow rates for different operational condition

The outlet air from the compressor is introduced to an annular combustion chamber and mixes with the fuel which is directed to the burner from another path.

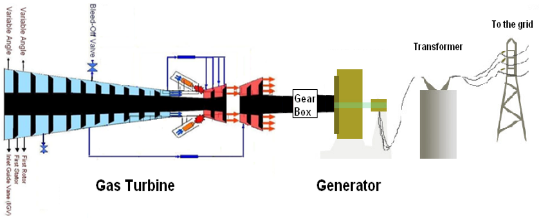


Figure 1. Selected gas turbine schematic for power generation application.

The mixture is ignited and the temperature of the outlet gas is increased. The hot gas mixture is introduced to the first-vanes row of the turbine and then revolves the first-blades row of turbine. These rows are cooled with milder temperature air in order to prevent the blades from the risk of firing because of the high temperature. The considered gas turbine is a twin-shaft which consist a high pressure turbine for driving the air compressor and a low pressure one for driving generator. A fraction of the compressor outlet stream is used to cool the vanes and blades of high pressure turbine. Also the first row vanes of the power turbine are cooled by the air which is extracted from the 5th stage of the air compressor. The rest of the rows do not require to be cooled because of the lower gas temperature in this region.

The agent air and flue gases are considered as ideal gases through all gas turbine components. The correlation used for calculating the thermodynamic properties of the mixtures are based on the Van Wylen[13].

A complete adiabatic combustion chamber process was considered for the combustion of the natural gas inside the engine combustor. The different selected parts and components of the engine were considered as separated control volumes which were related by the thermodynamic properties of the working fluid at their inlets and outlets. The mass and energy conservation equations were applied for all these control volumes and the overall model were constructed by integration of separate parts using matching roles between different components.

2.1 Design Point Modeling

An environment with 15°C temperature, 1atm absolute pressure and 60% relative humidity considered as the design point condition. The isentropic efficiency of the air compressor groups and turbine stages, the pressure loss of the combustion chamber and intake and exhaust, turbine inlet temperature, combustion efficiency, gas generator and power turbine rotational speeds were considered as known input data for design point calculations. On the other hand, the coolant mass flow rates, power required for compressor and power extracted from the high and low pressure turbines, fuel and air mass flow rates and air compressor pressure ratio are the main unknowns which can be calculated using the thermodynamic model. The cooling model is based on the El-Masri which is proposed for design point cooling of the turbine blades[9].

A computer code in visual FORTRAN language has been developed which was capable of calculating the thermodynamic properties at all sections of the gas turbine engine. The proposed model was used for studying the effects of any change in input parameters on the gas turbine performance. The power and efficiency of overall engine were determined from the following equations:

$$W_{net-cycle} = W_{turb} - W_{comp} \tag{1}$$

$$\eta_{th} = \frac{W_{net-cycle}}{\dot{m}_f \times LHV \times \eta_{cc}} \tag{2}$$

The different parameters such as mass flow rate, pressure ratio, polytropic efficiency were calculated from design point results and used for producing dimensionless characteristic curves of the air compressor, turbines and calculating pressure loss of air intake and exhaust and also as first initial guesses at off-design conditions.

2.2 Off-Design Modeling

2.2.1 Air Intake and Exhaust

The pressure loss of working fluid during passing the air intake and exhaust ducts are calculated using following expression[14].

$$\left(\frac{\Delta P}{P_0}\right) = \left(\frac{\Delta P}{P_0}\right)_D \times \left[\frac{\left(\dot{m}\frac{\sqrt{T_0}}{P_0}\right)}{\left(\dot{m}\frac{\sqrt{T_0}}{P_0}\right)_D}\right]^2 \tag{3}$$

2.2.2 Air Compressor

In the off-design model, the compressor was divided into four different groups as indicated in Figure 2. The performance maps of these groups were produced using full Three-Dimensional (3-D) numerical simulations and the necessary figures for different conditions were selected from these curves and were fed into the off-design model of the compressor. 3th group maps are shown at Figure 2.

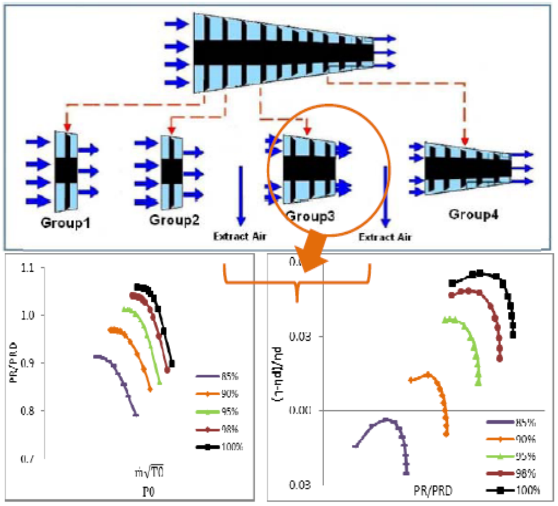


Figure 2. Air compressor groups and compressor group 3 maps.

2.2.3 Combustion Chamber

The simple and robust Dry Low Emission (DLE) combustor is based on an aero-derivative film-cooled concept. The selective combustor chamber is an annular type with 18 burners. Effective parameters on combustion chamber performance are pressure drop and efficiency. Combustion efficiency is defined as the ratio of actual fuel-air ratio for given temperature difference between combustor inlet and exit to theoretical fuel-air ratio[14]. The combustion efficiency acts as an input which is usually calculated from a 3-D model of combustor. This amount has been considered equal to 0.99 in our study and the necessary amounts in the off-design conditions came from combustor characteristic curve which is produced by a full 3-D numerical simulation. Figure 3 presents the combustion efficiency as a function of loading parameter which is defined by the following expression:

$$Loading = \frac{P^{1.75}\times 0.222\times \frac{T}{e^{300}}}{\dot{m}} \tag{4}$$

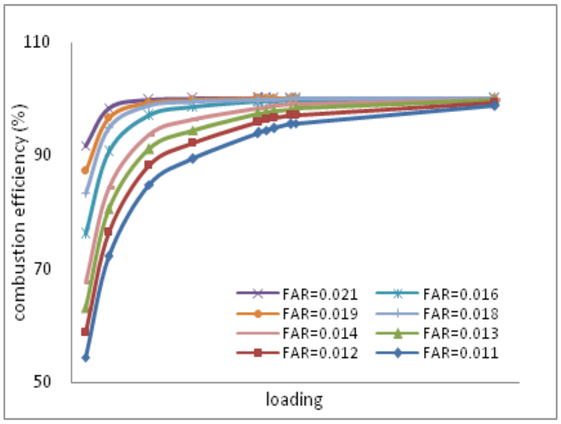


Figure 3. Characteristic curve of combustion efficiency versus loading parameter.

The combustor pressure drop is calculated by the 3-D numerical simulation for on and off-design conditions using:

$$\Delta P_{total} = 0.000404\times\left(\frac{\dot{m}\sqrt{T_0}}{P_0}\right)^2 + 0.00758\times\frac{\dot{m}\sqrt{T_0}}{P_0} \tag{5}$$

2.2.4 High Pressure and Low Pressure Turbine

In order to gain the most appropriate information for the gas path, full 3-D simulations (the same as compressor method) have been conducted and the individual maps for all the stages are derived. Figure 4 presents the stage 1 map for the un-cooled situation.

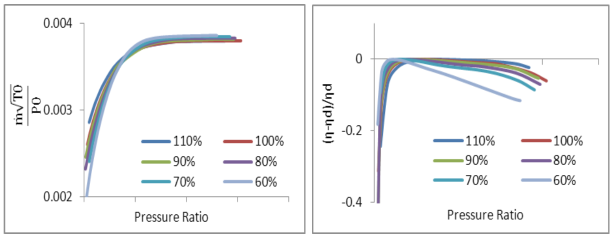


Figure 4. Turbine stage 1 Map.

2.2.5 Cooling Model

In the presented study the blade cooling model of El-Masri was used, in which the blade temperature is considered as an input and the actual stage by stage expansion replaced by a continuous expansion process. The cooling process in each stage is divided into 5 different steps. In the first step, a constant pressure cooling is considered (in Figure 5) which presents the mixing of coolant with main flow inside the stator. The second step is considered as an isothermal pressure loss process which is due to mixing of coolant and main flow (in Figure 5). The third step is an adiabatic expansion process inside the rotor. An isothermal pressure drop and a constant-pressure temperature drop inside the rotor are the two last steps; respectively[9].

2.2.6 Generator

A pseudo infinity power distribution grid was assumed in order to justify a constant frequency behavior in the

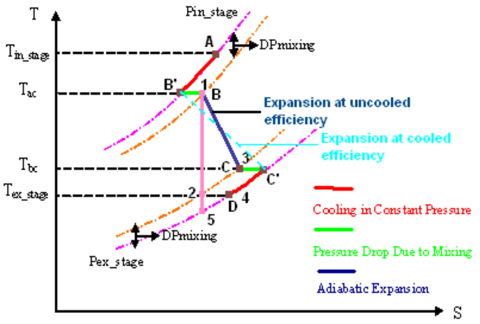


Figure 5. Thermodynamic states of the stage model[9].

model. This allowed the considering constant power turbine speed at 7700 rpm for all inlets environmental and load conditions.

2.2.7 *Control Philosophy*

As the control philosophy has a very crucial effect on the engine performance, therefore an appropriate philosophy would be needed in order to gain the best efficiency for the engine. There are different control philosophies for controlling the off-design engine performance for instance the constant turbine inlet temperature, constant turbine exhaust temperature and etc. The governor which is the main component and controls the engine operation determines the actuator position of the fuel valves, IGV and bleed valves. In this study, the main correlations for IGV, bleed valves positions and base turbine exhaust temperature are derived from turbine logic control.

2.2.8 *Gas Turbine Component Matching*

The operation point of any component is determined using off-design performance via component matching procedure. This try and error procedure starts from a couple of initial guesses for 16 parameters including 4 pressure ratios of 4 compressor groups, 4 pressure ratios of 4 turbine stages, 5 cooling mass flow rates, rotational speed of gas generator, fuel mass flow rate and intake pressure loss and ends up to the best operation point determination. The matching equations include mass balance between the components, energy balance between the high pressure turbine and compressor and finally reaching to a constant temperature equal to what the control philosophy dictates.

All gas turbine different components modeled separately and the appropriate equations for these components are solved together in order to develop a complete off-design model. The unknown parameters such as coolant and bleed valves mass flow rates, turbine and compressor pressure ratios, GG rotational speed and fuel mass flow rate are determined using the Newton Raphson solver. The schematic view of the solution algorithm is shown in Figure 6. As shown in the figure, the characteristic curves are applied to the model as known input data. As mentioned previously, these characteristics curves are derived from a CFD 3-D model.

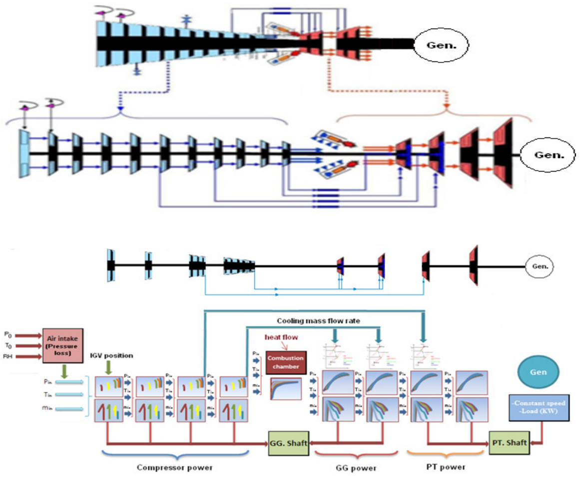


Figure 6. Schematic view of solution algorithm and component matching.

Figure 7 depicts the solution algorithm used to determine the off-design conditions for a twin-shaft gas turbine with power generation application.

3. Simulation Results

Figure 8 depicts the variation of some important gas turbine performance parameters as a function of gas turbine load and also the operating points on compressor map during loading process. In all performed simulations the gas oil with 42330 kJ/kg lower heating value was considered as the fuel. An increase in load causes GG speed up which will cause an increase in compressor air mass flow rate and compressor pressure ratio will occur. The power turbine will need more fuel in order to compensate the higher load demand which will produce higher TIT and TET. As the power change rate will be greater than the heat release rate therefore an increase in heat rate and a decrease in thermal efficiency will happen when load decreases.

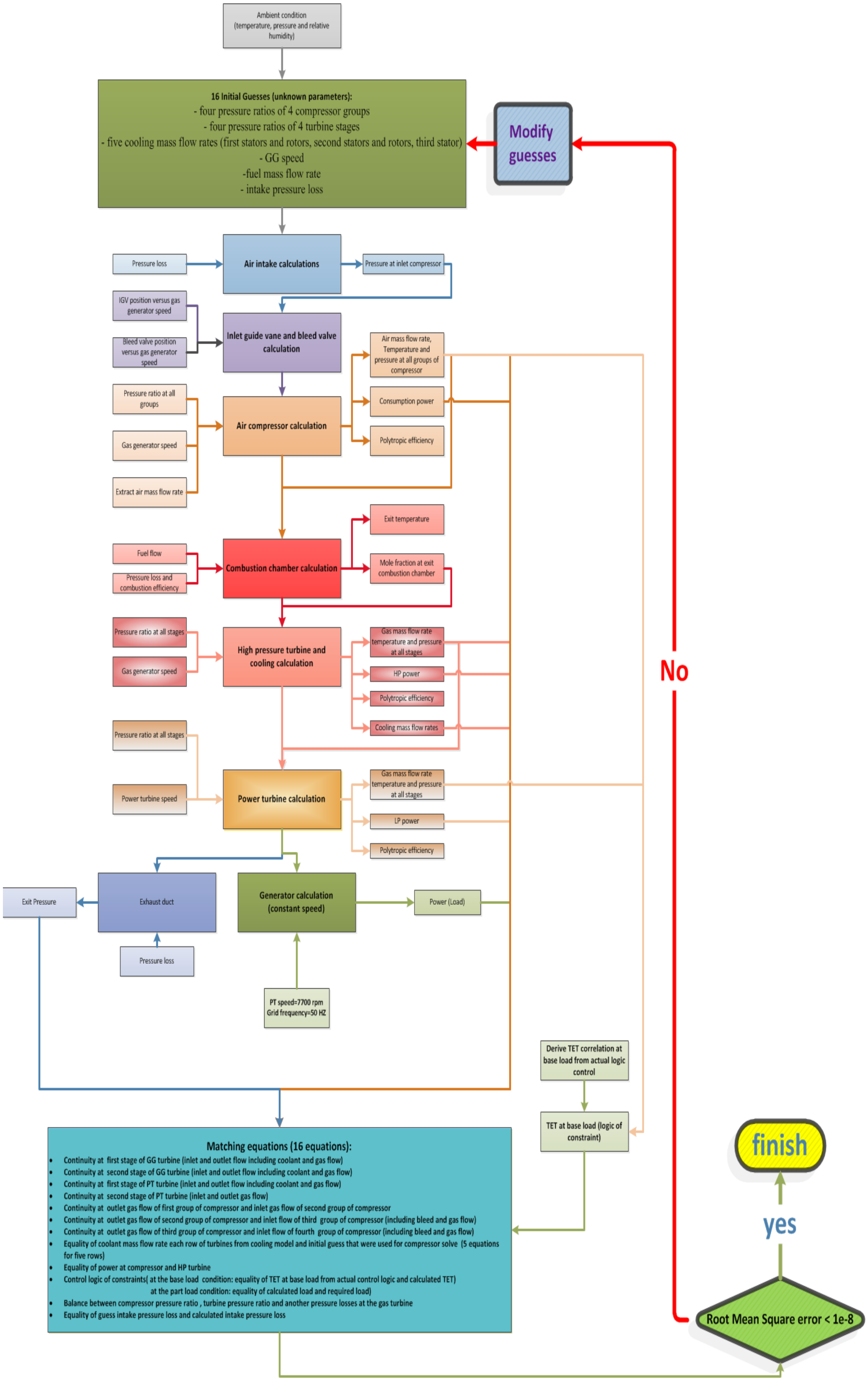


Figure 7. Off-design solution flowchart for the industrial twin-shaft gas turbine with power generation applications.

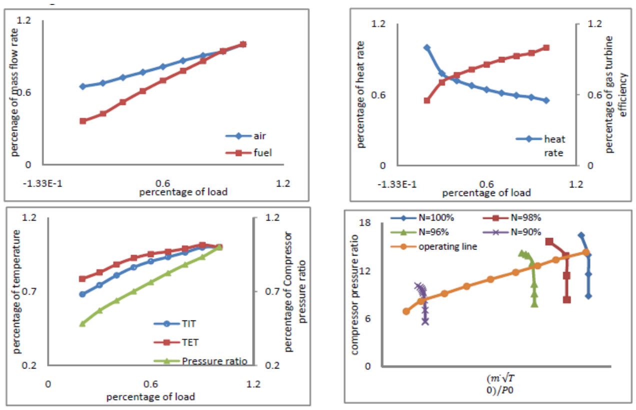
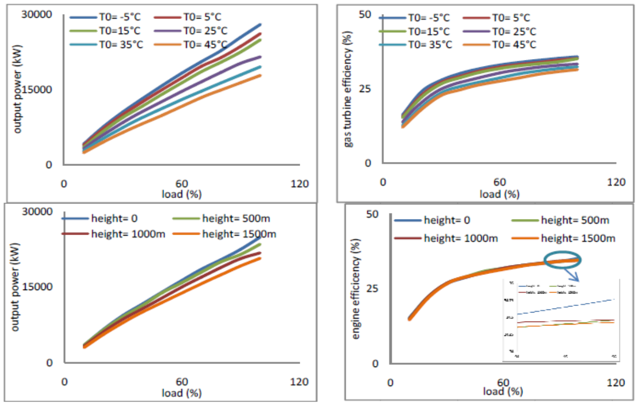


Figure 8. Operational parameters variation against load at ISO[1] condition (non dimensionless).

The effects of ambient air temperature and pressure on the output power and gas turbine efficiency are presented as a function of load in Figure 9. Increases in inlet air temperature or height from sea level causes a lower air density and lower mass flow rate through compressor. This will decrease the gas turbine output power.



a)Output power variation. b) Gas turbine efficiency variation

Figure 9. The variation versus load change at different ambient temperature and pressure.

4. The Validation of Developed Steady State Model

The model validation has been performed for design and some off-design points. The available experimental data for turbo-generator which has been run using gas oil with 42330 kj/kg lower heating value during the test in two steady state operation point were used in the validation procedure.

The model simulation results were validated by some correlations which have been presented by the engine manufacture reported in[15].

The developed model has been run for some off-design points and the results were compared with what have been reported in the available documents provided by the engine manufacturer. The effect of some important operational parameters such as ambient conditions on net output power were studied and compared with calculated figures using the following correlations (Figure 10)[15].

$$P_{sh-c} = (P_{sh-meas} + P_{loss-gear} + P_{loss-Gen}).K_{P-T0}K_{P-RH}K_{P-dpin}K_{P-dpout}K_{P-TIT}K_{P-f}\frac{P_{0ref}}{P_{omeas}} \quad (6)$$

Again a very good agreement exists which shows the high accuracy of the proposed model in this study. The results show a maximum error equal to 1.01% in predicted net output power.

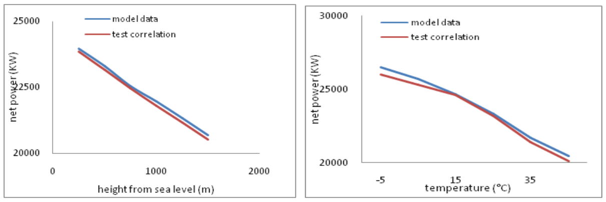


Figure 10. Comparison between output power variation than ambient temperature and pressure variation.

5. Conclusions

A mathematical model based on thermodynamic relations for a twin-shaft industrial gas turbine presented in this paper. The model was used to simulate the steady state behavior of power generation application of the gas turbine to study the effects of different parameters such as ambient conditions and loads. The three-dimensional flow effects were introduced in the model by application of characteristic curves of main subsystems such as air compressor, combustion chamber, HP turbine, LP turbine The flow properties and engine efficiency and output power at different loads were reported. An increase in load causes GG speed up which will cause an increase in compressor air mass flow rate and compressor pressure ratio will occur.

The model can be adapted for any single and twin-shaft industrial gas turbine using appropriate characteristic

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[1] The standardconditions used by the gas turbine industry are15˚C, 1.01325 bar and 60% relativehumidity, which are established by the International Standards Organization (ISO)and frequently referred to as ISO conditions.

curves for each separate engine. One of the main advantages of the proposed model is its ability to consider the effects of geometrical and control parameters on the gas turbine performance. In order to consider the effect of geometry on the performance analysis, the characteristics curves of the compressor, combustion chamber and turbine stages have been produced using 3D CFD simulations.

6. References

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