Off-design performance of solar Centaur-40 gas turbine engine using Simulink

M.H. Gobran *

Department of Mech. Power Engineering, Faculty of Engineering, Zagazig Univ., Egypt

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Abstract In the present study, a Simulink model based on Matlab software is used to calculate the off-design running point for single shaft Centaur 40 power generation gas turbine engine. The off-design calculations comprise two models, the first is the operation during engine starting (from 65% to 100% speed, no load) while the other is the engine operation during the loading (constant speed of 100%). For starting model the baseline parameter is the engine speed while the net power is the baseline parameter in the case of loading operation. Herein, the component characteristics maps, the air and air/fuel mixture properties as functions of temperature and the engine design point parameters are introduced to the calculating program. Because of the lack of real component characteristics, scaling law is followed to adapt these characteristics. The loading operation results are then compared with the field results to check the validity of Simulink model. Also the effects of the ambient temperature on the engine performance parameters at the design condition are investigated.

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1. Introduction

During the past decades, computer simulation has become a powerful tool for analysis of the processes in gas turbine engines. Nowadays computer models are indispensable for research and development in the gas turbine world. Computer models are also useful for the gas turbine engine user/operator or aircraft company installing engines in their aircraft. For optimization of installation, usage, diagnostics, health monitoring and maintenance, extensive analysis of engine performance is necessary. In these cases a computer model is very useful. For some analysis types a computer model is essential; for other types it may replace expensive test programs on a real engine. At NLR, gas turbine engine modeling is performed for both off-line and real-time simulation. Off-line gas turbine engine simulation is applied for:

- Performance analysis: needed for diagnostics, solving handling and performance problems experienced by engine/aircraft operators. A new application is performance sensitivity analysis for defining condition monitoring systems which use gas path analysis techniques.
Performance prediction: providing data for aircraft performance calculations, other (real-time) simulation models and emission calculations.

The latter application receives rapidly increasing attention. Real-time gas turbine engine models are developed for use in flight simulator models. Flight simulator models at NLR are generally used for research purposes, like development of flight control systems. Both off-line and real-time models exist in a variety of types, with different levels of complexity, fidelity, accuracy and computer performance requirements.

Szuch [1] designed a hybrid computer program (HYDES) capable of simulating one spool turbojet, two spool turbojet or two spool turbofan engine dynamics. The program is also capable of simulating two- or three spool turbofans with or without mixing of the exhaust streams. The method used the overall performance maps for the compressor, fan and turbine to represent the steady state performance of components basic gas dynamics, one dimensional, unsteady equation (continuity, momentum and energy) were written for the several of pressures, temperatures, flow rates and rotor speeds.

Shapiro and Caddy [2] and Wittenberg [3], use graphical simplified method to predict the engine off design performance using only the gas dynamic relationships (without components maps). This method depends upon many assumptions, so it leads to relatively rough estimation. In addition, it is not convenient for computer programming. Sellers and Daniels [4] used DYNGEN program for jet- and turbofan engine dynamic simulation. However, DYNGEN appeared to have many problems with numerical stability and had a poor user interface. Yang [5] and Zhu and Sarvanamutto [6], introduce the so called “hot end method” for predicting the steady state performance of LM-600 gas turbine engine. Simplified matching equations are used under the assumption that the low pressure turbine is choked (fixed high pressure turbine operating point). The solution begins from hot end to find the compressor operating point.

In Aerospace Department of the Delft Technical University; Broomhead and Visser [7] developed software called GSP (Gas turbine Simulation Program) which is capable of simulating almost all types of gas turbines (turbo shaft, turbo-prop, turbofan, single and multi-shafts, etc.); GSP is an “off-design” model. A predefined design point is calculated first from a set of user specified design point data. The deviation from the design point is calculated by solving a set of nonlinear differential equations. The equations are determined by mass balance, heat balance, equation for conservation of momentum and power (energy) balance for all components. In case of a transient simulation, the differential equations include time-derivatives. Then, in each time step, dynamic effects are calculated and the solution represents a quasi-steady state operating point. Besides being a performance prediction tool, GSP is especially suitable for parameter sensitivity analysis such as: ambient (flight) condition effects analysis, installation (losses) effects analysis, analysis of effects of certain engine malfunctioning (including control system malfunctioning) and component deterioration effects analysis.

The GASTURB program of graphical user interface method developed by Kurzke [8] was updated as version 9.0, and it is widely used as well-known program. The GASTURB simulates most of the common engine types: mixed and unmixed turbofans with or without boosters, turbo shafts with or without heat exchangers, and one shaft or two shaft turboprop engines. Also, the GASTURB can simulate the afterburner and the convergent/divergent nozzles in simulation of the turbojet and the mixed flow turbofan engine.

Since the middle of 1990s, programs of GUI (Graphical User Interface) method have increased using SIMULINK. It is a software package for modeling, simulating and analyzing the dynamic system. It is a provides a GUI for building models as block diagrams for modeling. Its analysis tools include linearization and trimming tools, which can be accessed from the MATLAB application toolboxes. There are some research examples using the SIMULINK.

Bettochi and Spina [9], developed a dynamic nonlinear model for a single shaft industrial gas turbine. This model consisted of modular structure representing engine components, and it was carried out in simplification configuration. Crosa and Pittaluga [10] described a 65 MW heavy duty gas turbine plant model using the SIMULINK®. In this model, a compressor with variable inlet guide vanes was represented in serial arrangement, separated by dynamic blocks with tabular characteristic data set using 2D-look-up table blocks. Kim [11], developed an accurate and reliable dynamic engine model using the SIMULINK®, and he used the dynamic model to simulate normal operating conditions from idle to maximum and severe conditions such as the fuel cut-off condition in flight and the starting condition in wind milling flight.

Kong and Rho [12] proposed a performance simulation model of the PT6A-62 turboprop engine using the SIMULINK to predict transient and steady state behaviors Yarlagadda [13], developed SIMULINK model the turbojet engine that may be used to evaluate the dynamic performance of integrated power system architectures during conceptual and preliminary design. The model can be employed to any turbojet engine with integration of the design data, component performance maps of the respective engine to the model. This model can be used in steady-state and transient analysis of the aero-engine.
Lazzaretto and Toffolo [14] presented a gas turbine design and off-design model in which the difficulties due to lack of knowledge about stage-by-stage performance are overcome by constructing artificial machine maps through appropriate scaling techniques applied to generalized maps taken from the literature and validating them with test measurement data from real plants. In particular, off-design performance is obtained through compressor map modifications according to variable inlet guide vane closure. The set of equations of the developed analytical model is solved by a commercial package, which provides great flexibility in the choice of independent variables of the overall system. The results obtained from this simulator are used for neural network training; problems associated with the construction and use of neural networks is discussed and their capability as a tool for predicting machine performance is analyzed.

In this study Matlab Simulink software was used to develop a model for steady state model of the Solar Centaur 40 single spool industrial gas turbine engine, during start up, load changes and ambient temperature changes. Due to practical difficulties and high cost to study these conditions practically, this software model was developed to simulate the real engine, and results were validated with both practical field data.

2. Case study

The present case study is a single shaft gas turbine coupled to a synchronous electric generator through a (1:10 ratio) reduction gear box. The gas turbine is fitted with 64 air filters to clean atmospheric air before entering an 11-stage axial compressor with two limiting position Inlet Guide Vane (IGV). Then the compressed air enters an annular combustion chamber where fuel is added and fired to supply the hot gases to the three stage axial reaction turbine. The pressure drop across the combustor is assumed to be proportional to the square of its inlet mass flow parameter, Ogbonnaya et al. [15]. Hot gases expand through the three stages axial turbine and deliver the mechanical power needed to drive both the compressor and the electric generator. The enthalpy drop across first and second stage is equal to 60% of the total enthalpy drop across the three stages, Kurz [16], thus,

\[ h_5 = h_3 - 0.6(h_3 - h_6) \]  

(1)

The engine produces a rated electric power of 3515 kW at the ISO condition (15 °C, 1 bar and 60% relative humidity). Gases are then discharged into the atmosphere through an exhaust system. The electric starter motor rotates the engine to self-sustaining speed; 65% of the design operating speed then the engine accelerates under its own power to loading speed (100% speed). After that the load is applied nearly at constant engine speed. Fuel system together with the control system are used to meter the required amount of fuel necessary to obtain engine requirements of speed and load without exceeding engine limitations (over speed, turbine excessive heating, and engine flameout). The electric generator transforms mechanical energy into electrical energy. Table 1 gives the engine design parameters as obtained from the manufacturer; Kurz [16], while Fig. 1 gives a diagrammatic sketch of the studied engine.

![Diagrammatic sketch of solar Centaur-40 gas turbine engine.](image)

### Table 1: Design parameters for Centaur 40 gas turbine engine.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Compressor pressure ratio</td>
<td>10</td>
</tr>
<tr>
<td>Air mass flow rate</td>
<td>18.34 kg/s</td>
</tr>
<tr>
<td>Gas generator shaft speed</td>
<td>15,000 RPM</td>
</tr>
<tr>
<td>Generator shaft speed</td>
<td>1500 RPM</td>
</tr>
<tr>
<td>Electric power</td>
<td>3515 kW</td>
</tr>
<tr>
<td>Maximum cycle temp.</td>
<td>1177.6 K</td>
</tr>
<tr>
<td>Compressor efficiency</td>
<td>85.8%</td>
</tr>
<tr>
<td>Turbine efficiency</td>
<td>91.5%</td>
</tr>
<tr>
<td>Generator efficiency</td>
<td>99.8%</td>
</tr>
<tr>
<td>Combustor pressure drop</td>
<td>5%</td>
</tr>
</tbody>
</table>

3. Air and gas thermodynamics properties

It is necessary to determine the thermodynamic properties of the working media (air or air–fuel mixture) at any point through the path of gases, these properties are enthalpy \((h)\), entropy \((s)\) and gas constant \((R)\) and specific heat at constant pressure \((C_v)\). These thermodynamic properties for air or fuel constituents \((\text{C}_4\text{H}_{10}, \text{C}_2\text{H}_6, \text{C}_3\text{H}_8, \ldots\text{etc.})\) are given as functions of temperature using the relations adapted from NASA, Bonnie et al. [17]. For the air fuel mixture any thermodynamic property, \(X\) is given by

\[ X = \left(\frac{X_a + f X_f}{1 + f}\right) \]  

where \(f\) is fuel to air ratio and subscripts \(a\) and \(f\) refer to air and fuel respectively. Any fuel thermodynamic property, \(X_f\) can be expressed as

\[ X_f = \sum G_i X_i \]  

where \(G_i\) and \(X_i\) are the mass fraction and property of the \(i\)th constituent. Now if one of these thermodynamic properties is given or calculated and it is required to calculate the corresponding temperature, these equations are inversely solved.

4. Thermodynamics process calculation

Hereafter a demonstration for basic thermodynamic processes is described:
4.1. Compression process

It is required to determine the air properties at compressor discharge (\(P_2, T_2, h_2, \ldots\)) from the known properties at compressor inlet (\(P_1, T_1, h_1\)). Firstly, the compression process is considered to be isentropic so \(S_2 - S_1 = 0\) (\(S = \phi - R \cdot \ln(P)\), where \(\phi\) is the temperature dependent entropy term), then

\[\phi_2 = \phi_1 + R \cdot \ln(\pi_z)\]  \hspace{1cm} (4)

Knowing \(\pi_z\) and \(\phi_1\) so \(\phi_2\) can be calculated and consequently \(T_2\) is inversely calculated by the Bisection method. \(T_2\) is then used to calculate \(h_2\) from the following relation the compressor discharge enthalpy is calculated

\[h_2 = h_1 + \frac{h_{2i} - h_1}{\eta_z}\] \hspace{1cm} (5)

Using Bisection method, the actual compressor discharge temperature can be calculated. Knowing this temperature all the remaining air properties are calculated.

4.2. Expansion process

It is required to determine the gas properties at the turbine exit, knowing properties at its inlet. Firstly the expansion process is considered to be isentropic so \(S_3 - S_0 = 0\), then

\[\phi_3 = \phi_1 + R \cdot \ln(\pi_z)\] \hspace{1cm} (6)

Knowing \(\phi_3\) and \(\pi_z\) so \(\phi_6\) can be calculated consequently \(T_6\) is calculated by Bisection method, \(T_6\) is then used to calculate the enthalpy at turbine exit \(h_{6i}\), then calculate \(h_6\) from relation,

\[h_6 = h_3 + \eta_i(h_3 - h_{6i})\] \hspace{1cm} (7)

Using Bisection method the actual turbine exit temperature \(T_6\) is calculated from \(h_6\), from this temperature we can calculate all other gas properties.

5. Components maps scaling

Compressor and turbine characteristics maps are confidential data and usually difficult to be obtained from gas turbine manufacturer. In order to overcome this problem, component maps for another compressor and turbine are obtained from GSP [7] software and scaled to match the design point of the compressor and turbine for the studied gas turbine engine. The first step in compressor map scaling is to choose map design point; although map design point is arbitrary but in fact there are some criteria that should be taken into account. The best choice for map design point is nearest to the maximum efficiency contour but unfortunately this will make the operating line of the compressor crosses the surge line at low compressor speed.

To overcome this problem the map design point should be moved step by step down the maximum efficiency region until the operating line comes under the surge line. In other words, after choosing the map design point, the steady start model (discussed in details in the following section) should be run and insure that the starting operating line (the inclined part of the operating line) doesn’t intersect with the surge line. If it does, the map design point should be moved down away from the maximum efficiency contour till the operating line becomes completely away from the surge line. In fact it is a trial and error procedure. After choosing suitable map design point, now we have two design points one for the map and one for the model. Scaling the map is simply to generate new map by multiplying each value in the given map by a factor that the design point of the new generated map becomes equal to the design point of the model. As the compressor map is a relation between pressure ratio, mass flow rate and isentropic efficiency, each parameter of the three should be scaled as follows; Szuch [1] and Kurzke [18]

\[\pi_{new, map} = \pi_{design, model} \cdot \left(1 + \frac{\pi_{map} - 1}{1}\right)\] \hspace{1cm} (8)

\[\left(\frac{\dot{m}\sqrt{T_i}\pi_{new, map}}{P_i}\right) = \left(\frac{\dot{m}\sqrt{T_i}\pi_{design, model}}{P_i}\right) \cdot \left(\frac{\dot{m}\sqrt{T_i}\pi_{map}}{P_i}\right)\] \hspace{1cm} (9)

\[\eta_{new, map} = \eta_{design, model} \cdot \eta_{map}\] \hspace{1cm} (10)

6. Design model

The main objective of the design calculations is to calculate all thermodynamics properties (\(P, T, h, \ldots\)) at all engine points, the fuel flow rate, the net power and some properties that used for Maps Scaling for both compressor and turbine, to achieve off design calculations. By using the design data given by Table 1 and the ambient condition \(T_1, P_1\) all the design properties are calculated through the thermodynamic cycle calculations.

(1) From ambient conditions \(T_1, P_1\) and considering air is the working media, the inlet enthalpy and entropy \((h_1, \phi_1, S_1)\) can be calculated using air properties.

(2) from \(\pi_z\) and design speed, the compressor operating point is located and consequently, the compressor exit thermodynamic properties and the air mass flow rate \((m_0)\) are determined.

(3) Now, it is required to calculate the fuel flow rate (fuel to air ratio) and the properties of gases at combustor outlet. Knowing all air properties at inlet and the combustor exit temperature. This is achieved by the following steps.

a. For a first iteration, assume the combustor exit is air, calculate the fuel-to-air ratio \(f\)

\[f = \frac{h_3 - h_2}{\eta_iHV - h_1}\] \hspace{1cm} (11)

b. Using \(f\) and \(T_3\) to calculate \(h_3\), from gas properties section.

c. Recalculate a new value of fuel-to-air ratio, \(f_n\).

d. Calculate the relative error between \(f\) and \(f_n\),

\[E_f = \frac{f_n - f}{f_n} \times 100\]

e. If this error is within tolerance limit, then it is the correct solution.

f. Otherwise, put \(f = f_n\) and repeat from step c

(4) \(T_3\) and \(f\) are then used to determine the remaining combustor outlet properties \((\phi_3, S_3)\).
(5) Knowing the gas properties at turbine inlet and the turbine pressure ratio \( \tau_i = 0.96^*\pi_i \), assuming 4% combustor pressure losses, the turbine outlet properties can be calculated.
(6) The overall gas turbine performance parameters (power, thermal efficiency) are then calculated.

7. Off-design operation

Herein, the engine steady state running points during acceleration from the idle speed of 65% to the design speed of 100% without loading and when the load is applied at constant speed (100%) are determined. The compressor corrected speed is used as a base line parameter in the case of steady start and the load is the baseline parameter at steady load. During the engine steady state operation, either without loading or with loading, it must satisfy certain matching constraints. These are:

1. Conservation of mass between compressor and turbine

\[
\frac{m_a^*\sqrt{T_i}}{P_3} = \frac{m_a^*\sqrt{T'_i}}{P_1} \left( \frac{n_{a} + n_{a}'}{m_a} \right) \left( \sqrt{\frac{T_i}{T'_i}} \right)
\]  \hspace{1cm} (12)

2. Power balance

\[
W_t - W_c = W_{net}
\]  \hspace{1cm} (13)

where the net power, \( W_{net} \), equals zero in the case of steady starting and load point \( W_t \) in the steady loading.

3. Rotational speed

\[
N_t = N_i
\]

\[
\frac{N_t}{\sqrt{T_i}} = \frac{N}{\sqrt{T'_i}} \left( \sqrt{\frac{T_i}{T'_i}} \right)
\]  \hspace{1cm} (14)

Two independent variables, locating the compressor and turbine operating points, are assumed. These variables are the compressor pressure ratio and fuel flow rate. There are lower and higher values for each variable and each of them are at the mid-point of the two extreme values (Bisection method). If these variables satisfy the matching constraints, then the operating point is reached. Otherwise, two errors representing the engine mismatch will arise. Then another variable is assumed, and the process is repeated until the mismatch error are within acceptable tolerance limits.

For certain engine speed (from 65% to 100% without loading or 100% for loading condition), the following procedure is carried out to determine the engine operating point:

1. First, assume the compressor operating point is at middle of first and last points in the series of selected speed points

compressor point = \frac{higher value - lower value}{2}

At this point, the compressor pressure ratio, efficiency and corrected mass flow rate (hence mass flow rate) can be obtained from compressor characteristics.

2. From the compression process the compressor discharge air properties are calculated \( T_2, P_2, h_2, s_2, \theta_2 \).

3. Fuel flow rate is assumed to be somewhere between (0 and 1 kg) so fuel lower limit is set to zero, fuel higher limit is set to 1.

\[
n_{f} = \frac{higher value - lower value}{2}
\]

4. The fuel flow rate together with air properties at compressor discharge are used to calculate the enthalpy at combustor exit \( (h_3) \), using the combustor heat balance.

5. Also the gas mass flow rate is calculated \( (\dot{n}_c = \dot{n}_o + \dot{n}_f) \) and \( T_3 \) is determined from \( h_3 \). Turbine inlet pressure \( P_3 \) is then calculated \( (P_3 = P_2 - \Delta P) \).

6. The calculated turbine flow parameter \( \left( \frac{n_t\sqrt{T'_i}}{P_1} \right)_c \) is then calculated from equation. (12).

7. Using turbine characteristics map and knowing turbine corrected speed \( (14) \) and turbine pressure ratio can get another value for turbine flow parameter \( \frac{n_t\sqrt{T'_i}}{P_1} \).

8. Two values of turbine flow parameter obtained from steps 7 and 8 respectively are compared, producing an error1 representing the mismatch in continuity.

\[
error_1 = \left( \frac{n_t\sqrt{T'_i}}{P_1} \right)_c - \left( \frac{n_t\sqrt{T'_i}}{P_1} \right)_c
\]

if the error between them is acceptable (0.5%) then the continuity is constraint is achieved.

9. Otherwise, if error is positive then fuel flow rate higher limit is set to the last value for fuel flow rate and repeat from step 5.

10. If error is negative then fuel flow rate lower limit is set to last value for fuel flow rate and repeat from step 5.

11. By using expansion process, the turbine exhaust gas properties can be determined \( (T_6, P_6, h_6, s_6, \theta_6) \) and then turbine power is calculated \( (W_t = \dot{n}_c(h_6 - h_5)) \). The turbine power is then compared with the compressor power \( (W_c = \dot{m}_c(h_i - h_1)) \) in the case of steady start or the turbine-compressor net power \( (W_{net} = W_t - W_c) \) is compared with the base line power, \( (W_t) \). Then, the error representing the power mismatch will arise.

\[
error_2 = \frac{W_t - W_c}{W_t}
\]

\[
error_2 = \frac{W_{netcalculated} - W_t}{W_{netcalculated}}
\]

if the error arising from this comparison is within acceptable limit, so the power matching constraint is achieved.

13. If the error is positive, the higher point will be set to fuel flow, and repeat from step 1.

14. If the error is negative, the lower point will be set to fuel flow, and repeat from step 1 Appendix A shows the steady loading SIMULINK model.
8. Results and discussion

8.1. Starting

Figs. 2 and 3 give the variation of the engine parameters \((P, m_f, m_a, T_2, T_3, T_5, T_6)\) during starting considering the compressor speed as a baseline parameter. It's noticed that all the above parameters increase with increasing the engine speed (relative to design speed), except the engine exhaust temperature \(T_6\), which is nearly constant.

8.2. Loading

Figs. 4–6 show the compressor pressure ratio, the air and fuel flow rates, compressor discharge temperature, turbine inlet, inter-turbine and exhaust temperatures, specific fuel consumption and engine thermal efficiency against the load. It is observed that the pressure ratio and fuel flow rate increase with increasing the load while the air mass flow rate remains constant with load, this is because the steady state loading line moves along choked compressor speed line. Also it is noticed that all engine temperatures \((T_2, T_3, T_6)\) increase proportionally with increasing load, except compressor discharge temperature \(T_2\), which is slightly decrease. This is due to the feature of compressor characteristics at 100% speed as the load increase the effect of improving the compressor efficiency diminishes the effect of increasing the pressure ratio. Fig. 7 shows the operating line of the compressor during starting and loading processes.

8.3. The impact of ambient temperature

When ambient temperature \(T_1\) increases the maximum power extracted from the engine (engine capacity) is decreased due to following reasons.

1. The density of air decreases so air flow rate decrease as the engine volume is constant.
2. The compressor work \((h_2 - h_1)\) increases.
3. Decreasing the maximum fuel flow rate that can be added due to limiting the inter-turbine temperature, \(T_5\) that gives an indication to the maximum cycle temperature, \(T_3\).
Herein, the steady state loading behavior with the ambient temperatures ranging from 258 K to 320 K is studied. In this model $T_5$ (which is function of real maximum temperature $T_3$) is kept constant at 905 K. This model is simply the steady load model with some modifications. The compressor point is guessed by the bisection method and $T_5$ used instead of power balance to close this loop.

Figs. 8–10 illustrate the effect of ambient temperature on various engine parameters (pressure ratio, output power, flow rate (fuel and air), thermal efficiency, specific fuel consumption and various temperatures (compressor discharge temperature $T_2$, turbine inlet temperature $T_3$ and turbine exit temperature $T_6$). From Figs. 8–10, it is clear that the pressure ratio, the maximum engine power, flow rate (fuel and air) and thermal efficiency decrease with increasing ambient temperature. Fig. 9 shows that the specific fuel consumption increases from 0.228 to 0.288 kg/kW h as the temperature changes from 258 to 318 K. Regarding to Fig. 10. Compressor discharge temperature $T_2$ and engine discharge temperature, $T_6$ increase with the ambient temperature (15% and 3.2% from their design values, respectively as the temperature is increased from 258 to 318 K), while turbine inlet temperature $T_3$ decrease by 2.7%.
8.4. Checking the validity of the model

Figs. 11 and 15 illustrate the validity of the present model by comparing some of its results \((T_2, \pi, m_f, T_5, T_6)\) with that obtained from the manufacture actual data, it is noticed that: as the operating point moves toward the design one, the errors between the manufacture data and simulation results decrease. This is may be due to that the real map and the scaled one are identical only at the design point. Table 2 gives the maximum percentage error in the above compared values. From this table, one can say that: the present model results are in a good agreement with the manufactured ones Figs. 12–14.

**Table 2** Error in the different parameters.

<table>
<thead>
<tr>
<th>Compared parameter</th>
<th>Max. error%</th>
</tr>
</thead>
<tbody>
<tr>
<td>Compressor discharge temp.</td>
<td>3</td>
</tr>
<tr>
<td>Pressure ratio</td>
<td>1.2</td>
</tr>
<tr>
<td>Fuel flow rate</td>
<td>-11</td>
</tr>
<tr>
<td>Inter-turbine temp.</td>
<td>-2</td>
</tr>
<tr>
<td>Exhaust temp.</td>
<td>-0.8</td>
</tr>
</tbody>
</table>

**Figure 11**  
T2 versus load for both simulation results and test data.

**Figure 12**  
Pressure ratio versus load for both simulation and test data.

**Figure 13**  
Fuel flow rate versus load for both simulation and test data.

**Figure 14**  
Inter-turbine temp. versus load for both simulation and test data.

**Figure 15**  
Exhaust gas temp. versus load for both simulation and test data.
Off-design performance of solar Centaur-40 gas turbine engine using Simulink

Steady state calculations for Generator load from 0 kw to 3500 kw
step 500 kw
While iterator
will repeat the execution
of this module until the condition is true
(so the estimated compressor point leads to
generator power near the estimated power)

do {
  ... cond
} while

this selector will choose mid point
as the lower limit if the calculated power
is less than the estimated power
else it will keep the lower limit as it is

lower value of
the point index (0)

higher value of
the point index (8)

average of
lower and higher values
of point index

multiply by -1

iterations to find the correct compressor map point

absolute value is the difference between the
estimated power and the calculated power
less than 1 watt
this selector will choose mid point as the lower limit if the calculated turbine flow parameter is less than the same parameter obtained from the map else it will keep the lower limit as it is

Lower and initial value for fuel flow rate

Upper and initial value for fuel flow rate

this selector will choose mid point as the upper limit if the calculated turbine flow parameter is higher than the same parameter obtained from the map else it will keep the upper limit as it is

Iterations to find correct fuel flow rate that make the turbine flow error tends to zero

While Iterator will repeat the execution of this module until the condition is true (so the turbine flow parameter mg5p3 error is almost zero)}
Calculate the turbine flow parameter twice once from the estimated fuel flow rate and the other from the map then calculate the difference (the error)
9. Conclusions

Herein conclusions are deduced from plotted graphs as follows:

During starting, all engine parameters increase with increasing the engine relative speed (relative to design speed), except the engine exhaust temperature which is nearly constant.

For loading condition, all engine temperatures increase proportionally with increasing load, except compressor discharge temperature that slightly decreases. This is due to the feature of compressor characteristics at 100% speed as the load increase the effect of improving the compressor efficiency diminishes the effect of increasing the pressure ratio. Both pressure ratio and fuel flow rate increase with increasing the load, while the air mass flow rate is constant. This is because the steady state loading line moves along chocked compressor speed line.

It is clear that maximum engine power, fuel flow rate and thermal efficiency decrease with increasing ambient temperature. Compressor discharge temperature and turbine exit temperature increase with increasing ambient temperature while turbine inlet temperature slightly decreases. Compressor pressure ratio and air flow rate decrease with increasing ambient temperature. For constant $T_5 = 905$ K, as the ambient temperature increases by 10 degrees (typically from 288 K to 298 K), output power was decreased by 11.16%, fuel flow rate was decreased by 7.45%, thermal efficiency was decreased by 4%, engine pressure ratio decreased by 4.2% and air flow rate decreased by 4.13%.

The Simulink model results are in agreement with the field data, so it is convenient to simulate the engine.

Appendix A

Simulink model for steady state loading model:

References