Parameter Estimation and Dynamic Simulation Of Gas Turbine Model In Combined Cycle Power Plants Based On Actual Operational Data

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Abstract: Gas turbines are very important nowadays for electric power generation specially that used in the ***Combined Cycle Power Plants (CCPPs***). For this electric power generation, the dynamics of the gas turbine and parameters estimation are very essential. In this article, a simple procedure is used for estimating the parameters of Rowen's model for HDGTs in dynamic studies for analysis purposes. The parameters of Rowen's model for a 265-MW HDGT are derived and several simulated tests using Matlab/Simulink are presented. The way of obtaining the parameters are based on simple physical laws. It explains briefly how to extract the parameters of the model using the operational and performance data. The obtained results via simulations using Matlab/Simulink are highly matched with the involved scientific articles that published in different literature. Furthermore, the obtained results verifies the operational results of the considered HDGT. However, the procedure here is applied on a practical HDGT. The same procedure could be applied for any scale (size) of gas turbines.

[H. Emam Shalan, M. A. Moustafa Hassan, A. B. G. Bahgat. **Parameter Estimation and Dynamic Simulation Of Gas Turbine Model In Combined Cycle Power Plants Based On Actual Operational Data.** Journal of American Science 2011;7(5):303-310]. (ISSN: 1545-1003). http://www.americanscience.org.

Keywords: Combined Cycle Power Plant (CCPP), Dynamic Simulation, Gas Turbine, Mathematical Modeling, Thermodynamic Process.

1. Introduction:

Gas turbines are one of the most important sources for electric power generation in countries with natural gas resources and are installed in several places in the world due to their special distinctiveness. The needs for a mathematical representation of gas turbines in dynamic studies guide to several publications in this area [1]-[6]. One of the most frequently used simplified models was presented early by Rowen [1] taking into account the load-frequency and temperature control as well as the turbine's thermodynamic responses as a linear function and inlet guide vane effects in a following separate work [2]. Several models with different degrees of simplification for the representation of gas turbines in dynamic studies were introduced. Among these models : the Detailed model (which is based on IEEE model) for ***Combined Cycle Power Plants (CCPPs***) which has deeper sight into internal processes [5]. A recent review of these models is given in [8, 9]. Among those articles dealing with dynamic studies and gas turbine performance only some works have been done on the model and its parameter extraction for deeper analysis purposes [3, 4]. However, many of electrical engineers are keen in performing dynamic studies. Rowen [1] does not include parameter estimation and details of each block's physical behavior. Therefore, similar to a previous work [4], the objective in this article is to produce a broad approach for power engineers and students to describe how the parameters of the turbine model can be derived from simple operational and performance data which is available in general. It is also useful to know at least which quantities are required to derive the parameters of the model and which features affect the parameters. As a case study, Rowen's model parameters are approximated here for a 265 MW single shaft Heavy Duty Gas Turbine (HDGT) by using operational and performance data of the gas turbine. In general, the gas turbine model is more complicated than the steam turbine model and thus needs more features to be studied. In this article, a simple procedure which is adopted recently by [4] is used for estimating the parameters of Rowen's model for HDGTs in dynamic studies. *It should be noted that the study in [1] was applied on a gas turbine of rated 60 MW, while the procedure that adopted by [4] was applied on a gas turbine of rated 172 MW. However, the same procedure is applied in this article, but on a practical gas turbine of 265 MW.*

2. Case Study of 265 MW HDGT

A 265 MW simple cycle, single shaft Heavy Duty Gas Turbine (HDGT) and its available operational and performance data are presented and studied for deriving the parameters of the model. These parameters are used in a further simulation studies. Table (1) shows nominal data of the selected HDGT for modeling. For land based engines, performance data are frequently quoted at the single point standard conditions. These standard conditions used by the gas turbine industry are:

- Ambient temperature (T_A) 15 °C / 59 °F.
- Ambient pressure (P_A) 1.013 bar / 14.7 psi.
- Relative humidity (Φ) 60%.

These conditions are established by the International Standards Organization (ISO) and frequently referred to as ISO conditions [1, 3, 4, 10]. For modeling purposes, a typical operating point is selected and illustrated in Figure (1). In advance, it is assumed that the considered model should be designated to represent the HDGT at rated load. It should be taking into account that in the following computations, the pressure loss in the entrance air filters and at the combustor is neglected. Also, IGVs are not modelled based on Rowen [1].

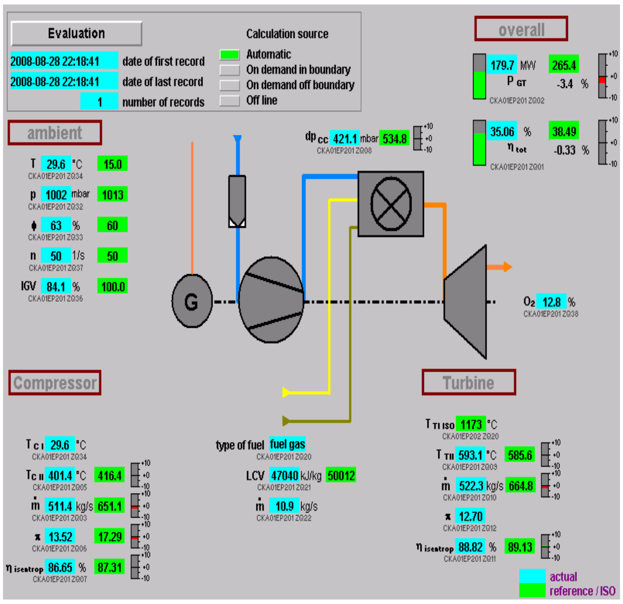

Figure (1): Plant Display Graphic of 265 MW HDGT[10]

2.1 Compressor and Turbine Efficiencies:

Using Table (1) and Figure (1), compressor and turbine efficiencies (η_C, η_T) parameters can be finally computed based on [3, 4] as follows:

$T_{2is(oc)}$ = 635.46 K = 362.46 °C
η_C = 0.895
$T_{3(oc)}$ = 1241.5 °C;
$T_{4is(oc)}$ = 792.9 K= 519.9 °C
η_T = 0.889

Where
(oc) Index stands for operating conditions of Figure (1).
(nc) Index stands for nominal conditions of Table (1).

Due to space limitations, only the final results are mentioned in this article, but for the detailed equations, one could refer to [3, 4]. It is inherently assumed constant compressor and turbine efficiencies in power output near nominal. It should be noted that the estimated efficiencies for the compressor and turbine are close to the actual results as illustrated in Figure (1).

2.2 Turbine Output Mechanical Power:

From now on, the turbine parameters are computed for nominal operation conditions (***nc***) according to Table (1). To extract the parameters of turbine output mechanical power block in [1], Figure (2) is depicted.

Table (1): Design Specifications of an Actual HDGT [10]

Parameter	***Symbol***	***Unit***	***Value***
Electrical power	P_{GT}	MW	265.4
Nominal frequency	F	Hz	50
Turbine speed	N	rpm	3000
Cycle efficiency (simple cycle)	η	%	38.49
Cycle efficiency (combined cycle)	$\eta_{combined}$	%	~ 58
Compressor type	17-stages axial		
Compressor pressure ratio	CPR	bar	17.29
Compressor inlet air mass flow	W_A	kg/sec	651.1
Combustor type	Annular		
Primary operating fuel	Natural gas		
Fuel mass flow	W_F	kg/sec	13.9
Lower heating value of fuel	LHV	kJ/kg	50012
Turbine type	4-stages axial		
Turbine diffuser exhaust mass flow	W_X	kg/sec	664.8
Turbine diffuser exhaust temperature	TET	°C	585.6

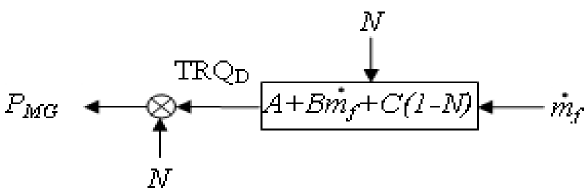

Figure (2): GT Output Mechanical Power Block of Rowen

Assume that, the considered HDGT is at nominal speed. In this point, the per unit (pu) output torque (***TRQ_D***) and mechanical power (***P_{MG}***) would be the same. Based on simple mathematical calculations in [4], ***A*** and ***B*** could be obtained via applying the actual data from Table (1) and finally calculated as follows:

$$A = -0.117 \ ; \ B = 1.11699 \ ; \ m_{f(nc)} = 14.157 \ kg/sec$$

Where

A, B Coefficients of developed (output) torque in Figure (2).

It should be noted that the estimated nominal fuel flow ($\dot{m}_{f(nc)}$) is very close to the actual one given in Table (1). The value of the speed sensitivity coefficient ***C*** in Figure (2) varies between 0.5 and 0.67 [4]. Here, ***C*** value assumed to be 0.5 (i.e., **C = 0.5**).

2.3 Turbine Exhaust Temperature:

Figure (3) represents the exhaust temperature (T_X) block in [1]. It should be noted that, nominal flows are needed to determine the parameters of this block (i.e., ***D*** and ***E***). At nominal speed, the exhaust temperature parameter ***D*** could be finally computed as [3]: $D = 492.6$ ℃
Based on discussion in [4] regarding temperature changes versus speed (N), the speed sensitivity coefficient ***E*** will vary in the range of 0.55 to 0.65 of rated exhaust temperature. Therefore, ***E*** is chosen to be $0.6T_R$. Hence:
$E = 0.6T_R = 351.36$ ℃

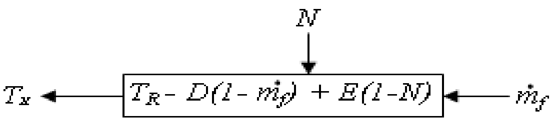

Figure (3): GT Exhaust Temperature Block of Rowen

However, temperature control in HDGTs requires measurement of the exhaust temperatures which may be composed of thermocouple and radiation shield [1]. Here, this study is only interested in the exhaust gas temperature out of the turbine (a convective source) to control the temperature and avoid excessive heating. Nevertheless, the radiation source, i.e., the turbine itself, will cause errors in the temperature measurement. The radiation shield is therefore used to overcome this problem, as illustrated in [3]. Temperature measurement device is the thermocouple which has a typical lag with a time constant based on its type and design. Time constant of thermocouple can be easily extracted from its time response documents. Figure (4) represents the exhaust temperature measurement block of Rowen [1]. To extract the parameters of the exhaust temperature measurement equipments, the same procedure and results of [4] have been adopted. The pertaining results are summarized in Table (2). More details are presented in [3, 4].

2.4 Fuel System:

Gas turbine fuel system is designed to provide energy input to the gas turbine in proportional to the product of command signal (V_{CE}) times the unit speed (N) as shown in Figure (5). Assuming linear response actuators and valves, the fuel flow will change directly with the output signal of the valve positioner. However, there is a lag associated with gas/oil flow in the pipes and fuel system manifold (T_{FS}). According to [11], this lag can be simply approximated.

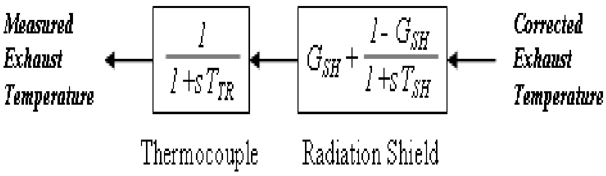

Figure (4): GT Exhaust Temperature Measurement Block of Rowen [4]

Table (2): Exhaust Temperature Measurement Parameters of 265 MW HDGT

Parameter	***Symbol***	***Unit***	***Value***
Radiation shield parameter	G_{SH}	--	0.85
Radiation shield time	T_{SH}	sec	12.2
Thermocouple time	T_{TR}	sec	1.7

Using Tables (1) and (3), the fuel system time constant (T_{FS}) could be easily computed to be as [3]: $T_{FS} = 0.31$ ***sec.*** It should be noted that the approximation is inevitable when using Tables and graphs of thermodynamic properties [12], because not all the operating points are provided there.

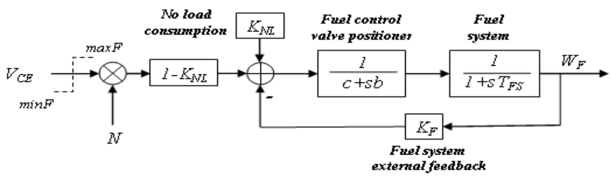

Figure (5): Fuel System of Rowen's Model [1]

Table (3): Operational Data for NG System [10]

Parameter	***Unit***	***Value***
Fuel	Natural Gas	
Fuel pressure	atm	26
Average temperature	K	293
NG piping approximate volume from the NG skid to the GT nozzles	m^3	0.25 ~ (equivalent cylinder of length about 8 m and radius of 10 cm)

There is also a fuel demand signal limiter in the HDGT model. ***maxF*** is not reached in normal operation and may act as a backup to temperature control where any increase in turbine exhaust temperature will lead to activation of temperature control and decreasing fuel flow. ***minF*** is a negative value which shows the gas turbine ability of transient power absorption. Its value depends on minimum fuel flow required to maintain the flame in the combustor. A value of 1.5 pu is commonly used for ***maxF*** while ***minF*** value can be determined by operational data for the fuel system. From a dynamic standpoint, gas turbines have many differences from steam turbines. One of the more obvious differences is the need for a significant fraction of rated fuel support self-sustaining at no-load conditions. This fact that HDGTs need relatively high fuel consumption to operate even in no load conditions as reflected in Rowen's model. It is reflected by dividing the fuel demand signal into a constant part (K_{NL}) and the reducing gain multiplied by the demand signal ($1-K_{NL}$). For this reason, one of the economic driving forces to minimize operating time at full speed, no-load conditions. Data for no-load operation and minimum fuel flow rate to maintain combustor flame are given in Table (4) as well as Figure (6).

2.5 Time Delays and Compressor Discharge Lag:

The gas turbine behavior forces its dynamic model to have small delays and lag time constants, as shown in [3]. Actually, there is a small time delay between the fuel injection and heat release in the combustor which is called combustion reaction delay (E_{CR}).

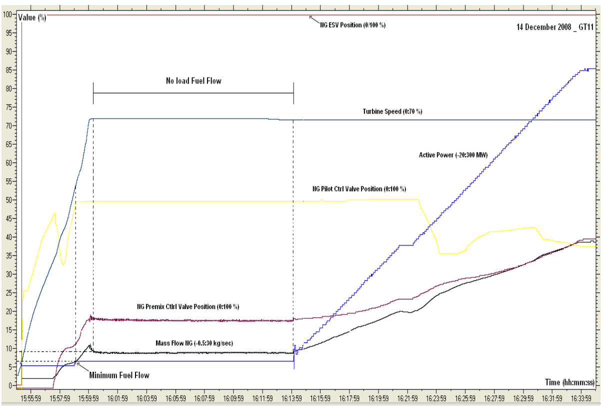

Figure (6): Trend for Minimum Fuel Flow and No-Load Consumption [10]

In modern systems, it is on the order of some milliseconds. Hence, it is assumed that, there is a time of 5 msec between the fuel injection until burning in the combustor (i.e., E_{CR} = 5 msec). There is also a time delay between the fuel combustion and exhaust temperature measuring system (E_{TD}). This delay is caused by exhaust system and turbine to transport the fluid to the measuring point. E_{TD} is in the order of milliseconds and depending mainly on the size of the HDGT and the average fluid speed. A relatively conservative value of 40 msec delay for air and combustion products transfer to the temperature measuring point is assumed (i.e., E_{TD} = 40 m sec). A relatively higher time lag exists in the compressor discharge path to the turbine inlet (T_{CD}). Considering the average temperature = 1100 K for Natural Gas, and using Table (1), and [3] as well as the thermodynamic properties of air [12], T_{CD} is approximated and could be calculated finally as: T_{CD} = 0.142 sec

Table (4): Operational Data for Minimum Fuel Flow and No-Load Consumption [10]

Parameter	***Unit***	***Value***
No load fuel flow	Kg/sec	~2.2
Minimum fuel flow to maintain combustor flame	Kg/sec	~1.4

2.6 Controllers Loops Parameters:

As shown in Figure (7), the values of the frequency-load and temperature control blocks are based on [1], for simulation purposes. All obtained values of the controllers parameters are in the range of the typical values which are usually used for HDGT dynamic models. It should be noted that some modifications are added from GT models in [5, 6] to the model in [4] to obtain the model that shown in Figure (7).

3. Model Simulation and Results of 265 MW Gas Turbine:

The derived and assumed parameters in previous section are used to simulate the behavior of 265MW HDGT. All parameters of the model presented in Figure (7) are reviewed, compared to [1, 4], and listed again in Table (5). The considered model has been simulated and tested under certain conditions as stated below.

3.1 Model performance at rated load:

The considered GT model is simulated under operating in nominal conditions (rated load) against three scenarios as follows:

- 0.1% speed step decrease.
- 0.3% speed step decrease.
- 20% load step increase.

The speed governor droop of 4% is assumed for the simulation, as illustrated in speed-load control block of Figure (7) and Table (5).The first scenario is presented where the turbine speed downs by step of -0.1% during normal operation under nominal conditions. The behavior of the mechanical output power against -0.1% speed deviation is shown in Figure (8).

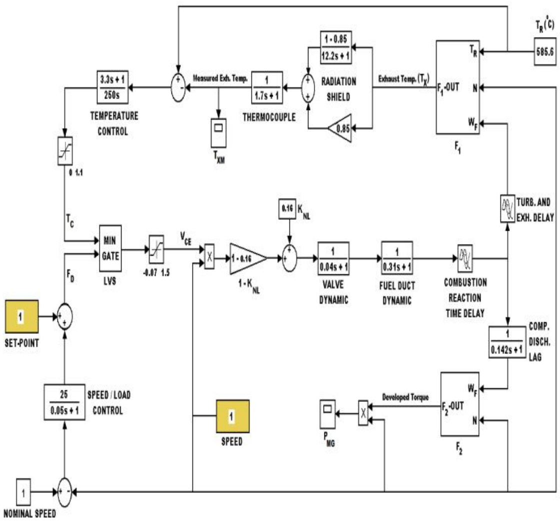

Figure (7): 265MW HDGT Model for Dynamic Studies

In steady state case, for 4% droop, the value of ~ 1.02 pu is observed for the output mechanical power. Figure (9) shows the exhaust temperature of the gas turbine which is measured by the thermocouple. Also, a final steady state value of near 596 °C is observed for the measured exhaust temperature, where the temperature control is not activated yet. Figure (10) ensures this meaning where the output of the frequency-load controller (***F_D***) is all time lower than the output of the temperature controller (***T_C***). Therefore, the frequency-load controller output passes through the minimum gate and controls the fuel flow to the combustion chamber. However, in the second scenario, -0.3% step in turbine speed would cause the temperature control to be activated, as shown in Figure (13). Figure (11) shows the behavior of the mechanical output power against -0.3% turbine speed step, where a final value close to 1.066 pu is noted. Also, in Figure (11), the power output remains constant until temperature control activation in 60 sec roughly. The measured exhaust temperature increases in this period (~60 sec) until it reaches the value of around 616 °C. At this time, temperature control activation have been occurred and forces the exhaust temperature to down to its rated value of 585.6 °C and decreases the output power, as illustrated in Figure (12). The third scenario is presented where the turbine power set point increases by step of 20% during normal operation under nominal conditions.The behaviour of the turbine parameters is illustrated in Figures (14), (15), and (16).

Table (5): Extracted Parameters of HDGT

Parameter	Symbol	Unit	265 MW HDGT values [3]	Rowen Values [1]	172 MW GT Values [4]
Speed governor gain=1/droop	W	MW_{pu} / N_{pu}	25	25	25
Speed governor lag time constant	Y	sec	0.05	0.05	0.05
Fuel demand signal upper limit	*maxF*	pu	1.5	1.5	1.5
Fuel demand signal lower limit	*minF*	pu	-0.07	-0.1	-0.13
No load fuel consumption	K_{NL}	pu	0.16	0.23	0.24
Fuel system external feedback loop gain	K_F	pu	0	0	0
Valve positioner time constant	b	sec	0.04	0.05	0.04
Fuel system transfer function coefficient	c	--	1	1	1
Fuel system time constant	T_{FS}	sec	0.31	0.4	0.26
Combustion reaction time delay	E_{CR}	sec	0.05	0.01	0.05
Turbine and exhaust delay	E_{TD}	sec	0.04	0.04	0.04
Compressor discharge volume time constant	T_{CD}	sec	0.142	0.2	0.16
Turbine rated exhaust temperature	T_R	°C	585.6	510	522
Gas turbine torque block parameter	A	--	-0.117	-0.299	-0.158
Gas turbine torque block parameter	B	--	1.1169	1.3	1.158
Gas turbine torque block parameter	C	--	0.5	0.5	0.5
Gas turbine exhaust block parameter	D	°C	492.6	390	413
Gas turbine exhaust block parameter	E	°C	351.36	306	313
Radiation shield parameter	G_{SH}	--	0.85	0.8	0.85
Radiation shield time constant	T_{SH}	sec	12.2	15	12.2
Thermocouple time constant	T_{TR}	sec	1.7	2.5	1.7
Temperature controller parameter	G_{TC}	--	3.3	3.3	3.3
Temperature controller integration rate	T_T	°C	250	250	250
Compressor efficiency	η_C	pu	0.895	--	0.86
Turbine efficiency	η_T	pu	0.899	--	0.89

The behaviour of the turbine parameters is illustrated in Figures (14), (15), and (16). These figures show that, the temperature control plays a vital role. Since it is override the speed/load control in case of overload. This is to protect the turbine against overheat. In other words, despite the set point is adjusted to a higher value than the permitted value, the temperature control takes over the responsibility to fix the positions of fuel valves and the turbine power accordingly to the permitted values.

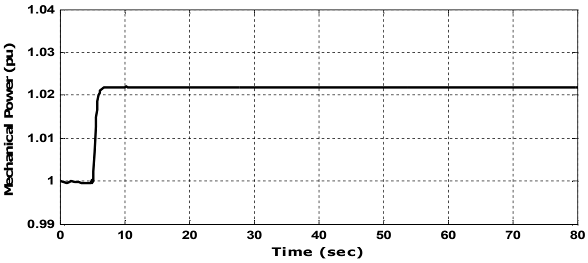

Figure (8): GT Mechanical Output Power after Speed Drop by -0.1%

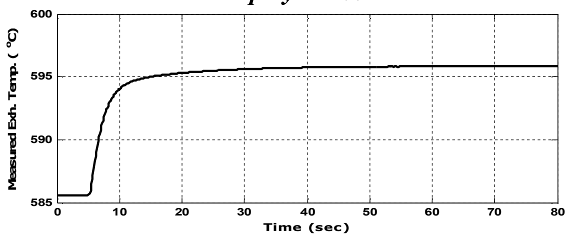

Figure (9): T_{XM} Measurement after Turbine Speed Step of -0.1%

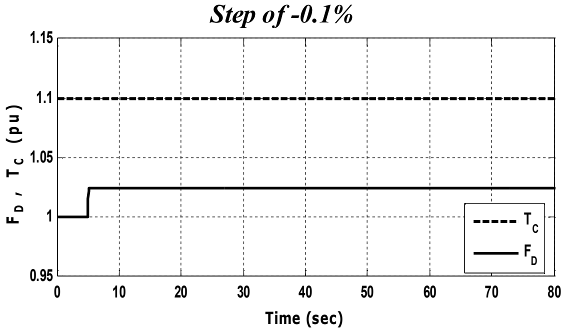

Figure (10): F_D and T_C against -0.1% Turbine Speed Deviation

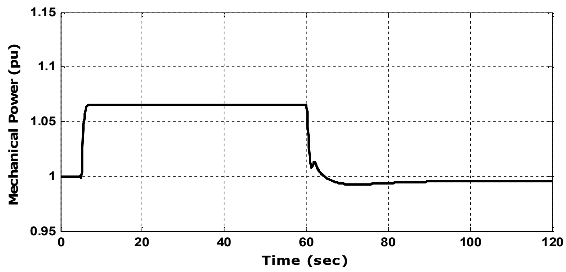

Figure (11): GT Mechanical Output Power after Speed Drop by -0.3%

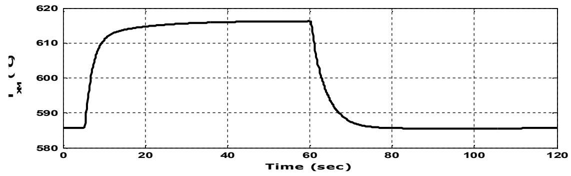

Figure (12): T_{XM} Measurement after Turbine Speed Step of -0.3%

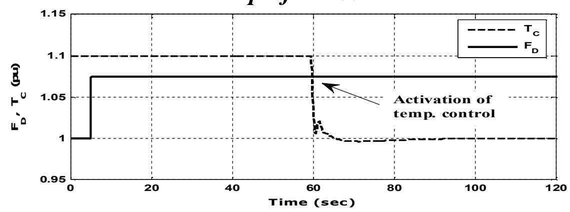

Figure (13): F_D and T_C against -0.3% Turbine Speed

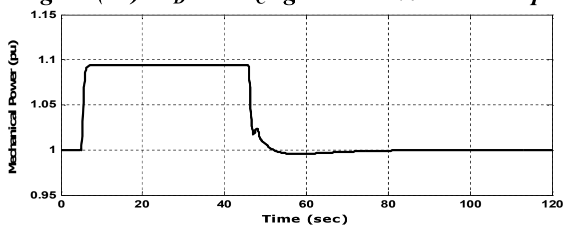

Figure (14): P_{MG} against +20% Load Increase

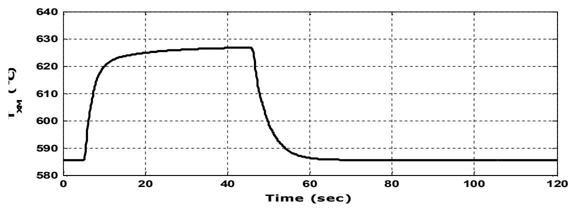

Figure (15): T_{XM} against +20% Load Increase

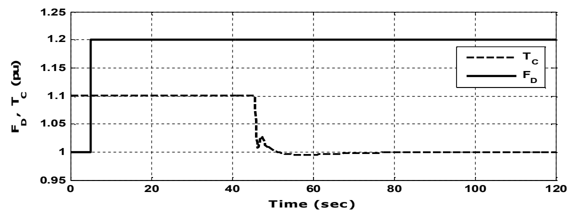

Figure (16): Inputs of LVS against +20% Load Increase

3.2 Model Performance at Partial Load:

Since the considered model is designated to represent the HDGT at nominal point, the load set point is significantly more accurate at base (rated) load (i.e. set point = 1 pu), but less accurate at partial load. However, adjustments for the parameters $\boldsymbol{A}$, and $\boldsymbol{B}$ of mechanical power block that shown in Figure (2) can enhance the accuracy of the set point at partial load. Values of $\boldsymbol{A}$ = -0.3 and $\boldsymbol{B}$ = 1.3 can be selected for this purpose. Behavior of the turbine output power ($\boldsymbol{P_{MG}}$) has been examined against -0.3% turbine speed deviation, when operating at the selected operating point (i.e. the turbine initially load by 0.67 pu roughly). After that, the temperature control loop is disabled and the simulations are repeated again. The results are represented in Figures (17) and (18). It can be seen that there is no difference between the curves with and without temperature control loop. This is expected, as the operating point is far away from the rated exhaust temperature limit as shown in Figure (18), due to the low power output of the gas turbine as shown in Figure (17). These results are highly matched with those given in [13]. Hence, the temperature control loop is not active and makes the presence of the temperature control superfluous. Under such conditions therefore, the temperature control loop can be neglected without any loss in model accuracy, because the output of the gas turbine is effectively determined by the governor only.

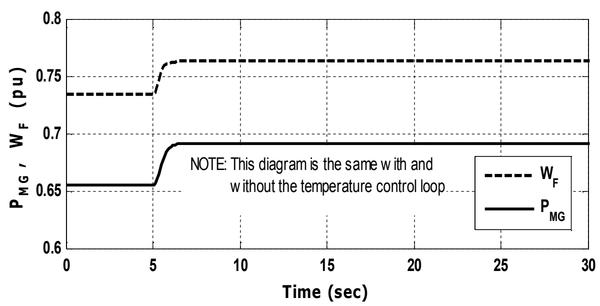

Figure (17): Partial Load of P_{MG} against -0.3% Turbine Speed Deviation

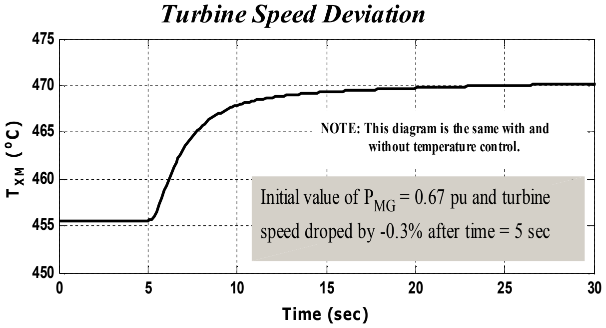

Figure (18): T_{XM} (^oC) against -0.3% Turbine Speed Deviation at Partial Load

This consequence had been proven in [13] as well. Now, it more obvious that, the temperature control loop is implemented in gas turbines to avoid a severe damage in the turbine due to overheat.

4. Impact of Speed Droop on Gas Turbine Performance:

To illustrate the impact of the Governor with a speed droop characteristic on the gas turbine performance and hence the power system performance, the following example is provided: Suppose two gas turbine units, as shown in Figure (19). Both units are rated at 265 MW (1 pu) and are initially loaded around the selected operating point 179.7 MW ($I_o \approx 0.66$ pu), as illustrated in Figure (1). Both units have governors with speed regulation; however, Unit 1 is set for 8% speed regulation and Unit 2 is adjusted for 3% speed regulation. It should be known that, the droop setting is adjustable from 2 to 10 percent [1]. To examine units response, -0.4% (ΔF = -0.004 pu) frequency deviation is applied. Unbalance between generation and load will occur as a result of frequency deviation.

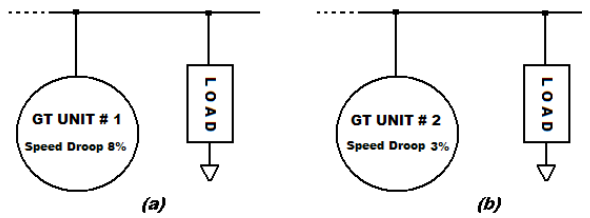

Figure (19): Two GTs with Different Speed Droop
(a) Unit 1 with Droop Setting 8% (b) Unit 2 with Droop Setting 3%

To rebalance generation and load again, an additional power must be produced by the each unit. Depending on the governor droop setting, the unit governor will increase the output power to rematch generation and load. The steady-state frequency can be determined by considering the speed-load characteristic curves for the two units as shown in Figure (20). To calculate the additional power that has to be increased by each unit according to its governor speed droop setting, the following equation can be used [14]:

$$\textit{Percent droop } (R\%) = \frac{\textit{Percent speed or frequency change } (\Delta F\%)}{\textit{Percent power output change } (\Delta P\%)} * 100$$

In case of unit 1 with 8% droop setting, the governor will increase the output power from around 179 MW ($I_o \sim 0.66$ pu) to around 188 MW ($I_1 \sim 0.71$ pu). That is to say, unit 1 produces ~ 13 MW (ΔP_1) roughly as an additional output power due -0.4% (-0.2 Hz) frequency deviation. In case of unit 2 with 3% droop setting, the governor will increase the output power from around 179 MW ($I_o \sim 0.66$ pu) to around 209 MW ($I_2 \sim 0.79$ pu). That is to say, unit 2 produces ~ 35 MW (ΔP_2) as an additional output power due -0.4% (-0.2 Hz) frequency deviation. An important result could be obtained from these calculations; ***a unit with a lower speed droop or regulation setting is more responsive to a change in system frequency.*** This result is highly matching the results that are given in [15]. More details about the droop governor characteristics are given in [14].

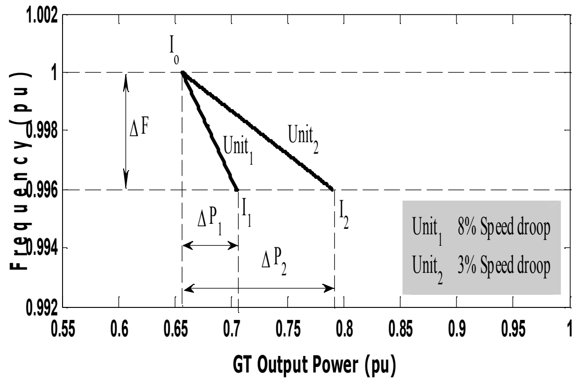

Figure (20): Impact of Speed Droop on Gas Turbine Performance

5. Conclusions:

In this article, a simple procedure is used for estimating the parameters of Rowen's model for practical HDGTs. These parameters could be used in dynamic studies for many purposes. The parameters of Rowen model for a 265-MW HDGT is derived and several tests using simulation are presented. The way of obtaining the parameters are based on simple physical laws and explained to some extents to make it useful for who are involved in dynamic studies of HDGT. All the obtained results via simulations using Matlab/Simulink are highly matched with the involved scientific articles that published in different literatures. The same procedure could be applied for any scale (size) of gas turbines. However, it is noticeable that the obtained results are significantly depends on the selected operating conditions.

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4/3/2011