Performance Improvement of Gas Turbine Speed Using Model Predictive Control

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Abstract- The flexibility and low specific cost investment of gas turbines have made them to be popular as power generating system over the last decades. One of the essential requirements in the study of gas turbine is accurate modeling which is required in the study of power system stability. This paper has presented performance improvement of gas turbine speed using Model Predictive Control (MPC). In order to study the speed control of a gas turbine of 25 MW capacity, the nominal and operational data were obtained and dynamic modeling and design were carried out. The values obtained were used for simulation for nominal operating condition in MATLAB/Simulink environment. The resulting speed response with respect to unit constant input before the introduction of the MPC controller was not able to track the desired speed level at full load assuming no load torque. An MPC controller was designed and added to the speed control loop of gas turbine. Simulation was conducted and the result obtained indicated that the MPC controller enables the desired speed to be optimally tracked with improved transient response performance from settling time of 53.7 seconds to 10.33 seconds (80.8% improvement) while achieving a peak percentage overshoot of 0.82% peak overshoot and steady state error of 0.007 p.u.

Indexed Terms- Gas turbine, MPC, Power stability, Speed control

I. INTRODUCTION

Gas turbines are usually employed as power generating engines because they are flexible and have low investment cost. They are one of the most essential sources of power generations for nations with natural gas resources [1]. With the function of gas turbines as a fast response unit that is suitable for improving transient response of power system being lost to a certain extent as a result of relatively high limitations in increasing and decreasing the power output during the nominal operation, accurate dynamic modeling of power units is required for power system stability [2]. This problem persists and in order to take care of it, it must be fully reflected in the modeling and simulation of a gas turbine.

Speed/load frequency control (LFC) offers essential functions in gas turbine control system. These functions include: a) as the control loop that is mainly involved during nominal operating conditions, b) as the most essential control loop for stability study of gas turbine, and c) providing better conditions for power exchange and supplying in trading electricity.

A simplified model presented by Rowen [3] has been used frequently in the study of gas turbine and has been adopted for the modeling carried out in this paper. This is because the nonlinear characteristics of gas turbine dynamics can be examined using the control system tools of the MATLAB/Simulink and local response can be imposed to a simplified linear model that is easily implementable in conventional power system analysis tools for the intention of carrying out LFC characteristics of a gas turbine.

This paper is a contribution to the systematic analysis of speed (or load-frequency) control concepts for heavy-duty single shaft gas turbines (HDSSGT). It is concerned with the improvement of load-frequency or speed performance of a gas turbine using a Model Predictive Control (MPC) strategy.

II. LITERATURE REVIEW

A. Single Shaft Gas Turbine Control

A typical gas turbine is shown in Fig. 1 and its main components are compressor, combustion chamber (combustor), and turbine. Compressor and turbine are connected by the central shaft and rotate together. From section 1, fresh air enters the combustion chamber at section 2. In the combustor, fuel is mixed with air and is ignited. The hot gases which are product of combustion are forced into the turbine at section 3 and rotate it. Turbine drives the compressor and the gas generator (GG) mechanical output, which can be an electricity alternator in a power plant station, or a large plant, or a large compressor.

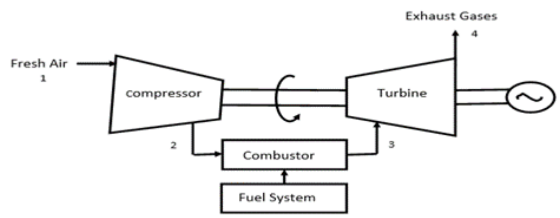

Fig. 1 Schematic diagram of a typical single-shaft gas turbine

Gas is mainly equipped with a complex control system which contains different loops such as governor, temperature controller, acceleration controller, and so on.

Many studies have been carried out to study the speed and overall performance of HDSSGT. The speed controller controls the speed of a gas turbine at operating speed when the turbine is not synchronized. A simplified mathematical model of heavy-duty single shaft gas turbines with full range of 18MW to 106MW was provided by Rowen [3] to investigate power system stability. The stability of a single shaft gas turbine and its control system against overheat as well as variations in frequency and load was examined by Mantzaris and Vournas [4]. A study on the design of self-tuning for a proportional integral and derivative (PID) controller on speed of gas turbines was done by Ismail [5]. Jamshidzadeh and Jamali [6] examined the role of gas turbine power plants in frequency correction within the power networks. Parameter estimation and dynamic simulation of gas turbine model based on actual operating data was done by Shalan et al [1] with simulation study of the gas turbine carried out in MATLAB/Simulink. Conversion of nonlinear dynamic model of industrial heavy duty gas turbine to linearized transfer function model was carried out by Sarumathi et al. [7]. A hybrid controller that combines Particle Swam Optimization (PSO) and Fuzzy-PID controller to give an optimized control technique called POS-Fuzzy-PID controller, for the control of a gas turbine speed and maintain the exhaust temperature in a desired interval during start-up and operating conditions was proposed by Mansourabadand Beheshti [8]. Balochian and Vosoughi [9] examined the design and simulation of turbine speed control system based on adaptive Fuzzy-PID controller. Gas turbine speed supervision based on Rowen model using PID and Fuzzy was presented by Ammar et al [10]. Bank Tavakoli et al. [11] approximated parameters of heavy duty gas turbines using simple thermodynamic postulations, and subsequently carried out step response simulation of the model. Lebele-alawa and Le-ol [12 examined the design of a 25MW gas turbine power plant at Omoku in Niger Delta area of Nigeria.

B. Basic Concept of Model Predictive Control

Model predictive control (MPC) techniques are commonly used over process and oil/petrochemical industries as a result of its capability to solve input and output constraints in an optimal form. This way system can be safely operated by restricting it to be conducted in a limited region of operation such as minimum and maximum liquid level within a distillation column or a maximum opening degree in degree in a valve (Stefano, 2014). General, determining a sequence of control moves in the manipulated variable is the main task of MPC controller. Thus the system can be optimally tracked to its setpoint.

The basic concept of MPC controller is depicted in Fig. 2. It shows the main idea behind MPC technique. Optimization problem is solved using MPC at each time step k through an objective function that is based on the predictions of the output over a prediction horizon of p time steps. This objective function is usually a quadratic one. It is minimized by selection of manipulated variables and moves over s control horizon of M control moves. Though at each time step a group of M moves is calculated, only the first move u_k is executed. Then after the measurement at next time instant y_{k+1} determined, followed by a correction as a result of model error, and after that a new optimization problem is addressed again. These actions are carried for each time step k .

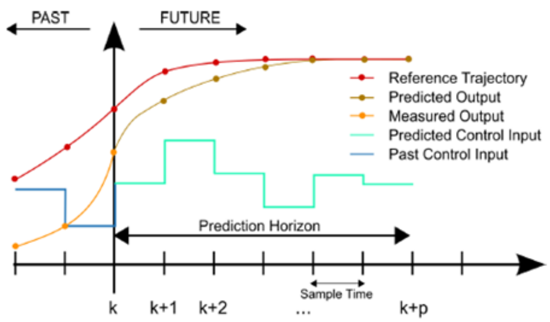

Fig. 2 Basic concept of MPC [13]

The model coefficients $(S_1, S_2, S_3, ..., S_N)$, are the output values at each time step k.

The predicted output of the model for an instant time k is calculated through the following equation [13]:

$$\hat{y}_k = \sum_{i=1}^{N-1} S_i \Delta u_{k-i} + S_N u_{k-N} \qquad (1)$$

Since the predicted output of the model is barely equal to the actual measured output at a certain time step k, this difference is represented by:

$$d_k = y_k - \hat{y} \qquad (2)$$

The corrected prediction is given by:

$$\hat{y}_k^c = \hat{y}_k + d_k \qquad (3)$$

Combining Eq. (1), (2) and (3), the corrected prediction for the jth step into the future can expressed given by:

$$\hat{y}_{k+j}^c = \sum_{i=1}^{j} S_i \Delta u_{k-i+j} + \sum_{i=1}^{N-1} S_i \Delta u_{k-i+j} + S_N u_{k-N+j} + d_{k+j} \quad (4)$$

Equation (4) looks like a "collection" of the effect of the future and past control moves as well as a correction term d_{k+j}. The future control moves are the first term to the right while the past control moves are the second and third terms. In addition, the difference between setpoint trajectory and future predictions in step, with $j \le p$, is given by:

$$r_{k+j} - \hat{y}_{k+j}^c = r_{k+j} - \left[\sum_{i=1}^{N-1} S_i \Delta u_{k-i+j} + S_N u_{k-N+j} + d_{k+j}\right] - \sum_{i=1}^{j} S_i \Delta u_{k-i+j}$$

(5)

Eq. (5) is crucial for the optimization problem and can be used in a quadratic objective function for a predictive horizon p and a control horizon of M moves:

$$f_{obj} = \sum_{i=1}^{p} \left(r_{k+i} - \hat{y}_{k+1}^c\right)^2 + w \sum_{i=0}^{M-1} \Delta u_{k+1}^2 \qquad (6)$$

III. METHOD

The components of the speed control design process are presented in this section. Figure 3 presents a flow diagram taken to realize and implement the control system in MATLAB/Simulink environment. The performance of a gas turbine can be effectively represented by capturing the main dynamics of the heavy duty gas turbine. In this section, mathematical representations or models of the dynamics of a heavy duty single shaft gas turbine are obtained.

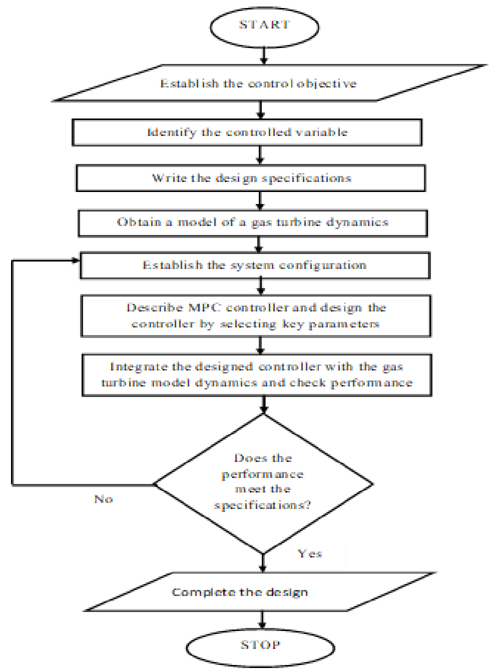

Fig. 3 Design flow diagram

Objective of the Control System: the objective of this study is to develop a speed controller that will enhance speed performance of a 26 MW heavy duty single shaft gas turbine.

A. Design Specifications

This work intends to design a control system that will improve the speed response performance of a single shaft gas turbine in some refineries for optimum stability offered by the speed control loop during normal operation. In order to realize the objective of this work, the control system has to meet certain performance criteria. The following specifications have been selected:

i. Rise time taken between10% to 90% of the final step response value should not be greater than 8.0s.
ii. Percentage overshoot lees than or equal to 5%.

iii. A settling time of less than or equal 15s.

B. *Dynamic Model of Gas Turbine*

Figure 4 shows a simplified diagram of a gas turbine model proposed in [3]. The figure shows that a typical gas turbine model has basically three control loop.

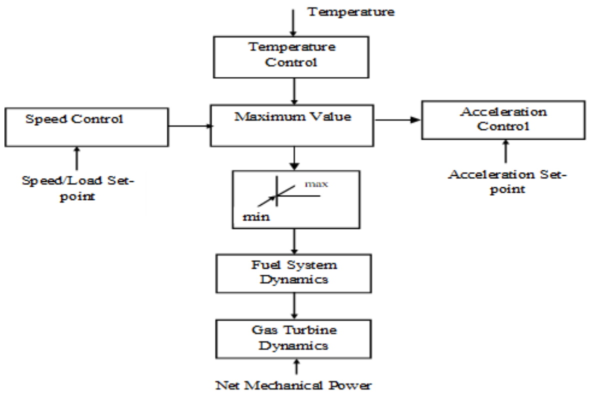

Fig. 4 Representation of gas turbine model

The control loop mainly active when the gas turbine is operating at nominal conditions is the speed control. The input to this control is the speed/load set-point. The control section is restricted by the minimum fuel limit. The speed control is also the most important aspect during stability study. A speed control configuration proposed in this work is shown in Fig. 5. In the figure, R(s) is the reference rotor speed. E(s) is the error, which is the difference between R(s) and C(s). U (s) is the control or manipulated variable. C(s) is the output or response rotor speed.

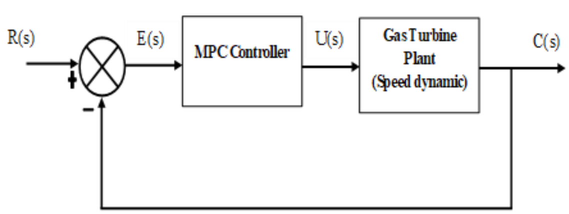

Fig. 5 Closed loop configuration of gas turbine speed control

The dynamics of the parts of the gas turbine that are essentials in analyzing the transient characteristics of a speed control in gas turbine are considered in this work and are presented next.

C. *Data Gathering*

In order to obtain the data for the simulation of in MATLAB/Simulink environment, a heavy duty gas turbine with installed capacity of 26 Mega-watt (MW) and very high rotational speed of 5100 rpm is considered in this work. The output voltage and current are 11 KV and 1267 A at an output frequency of 50 Hz. Power factor and ambient temperature rating are 0.8 and 25-45 degree Celsius. Table 1 shows the nominal data or design specifications and a typical average operational data in Table 2 of heavy duty gas turbine (HDGT).

Table 1Nominal data of selected HDGT

Parameters	Symbol	Unit	Value
Electrical power output	P_{GT}	MW	26.3MW
Heat Rate	H_R	kJ/kwh	12650
Exhaust Temperature	T_{OE}	°C	487
Exhaust Mass Flow	$\dot{m}_{no}$	Kg/s	124.1
Pressure Ratio	PR	-	9.87
Normal speed		rpm	5100
Lower Heating Value of Fuel(LHV)		kJ/kg	43309

Table 2 Average operating data for gas turbine [12]

Component	Parameter	Unit	Value
Compressor	Inlet temperature	°C	30.4
	Outlet temperature	°C	367
Combustion chamber	Inlet pressure	bar	1.013
	Outlet pressure	bar	10
	Mass flow rate (Air)	Kg/s	122.9
	Fuel consumption (flow rate)	Kg/s	1.2
Gas turbine	Inlet temperature	°C	959
	Outlet temperature	°C	487

Other data	Exhaust gas flow (flow rate of gas)	Kg/s	124.1
	GT power output	MW	25

D. Gas Turbine Parameters

The approach used in [11] for computing the values of the turbine and compressor efficiencies is adopted in this work. The values of the nominal data and operational data of Table 1 and 2 are used for the analysis performed.

i. Turbine efficiency (n_t):

$$x_{h(oc)} = \left(PR \times \frac{\dot{m}_{oc}}{\dot{m}_{no}}\right)^{\frac{x_h - 1}{x_h}} = \left(9.87 \times \frac{124.1}{124.1}\right)^{\frac{1.33-1}{1.33}} = 1.76 \quad (7)$$

$$T_{4s(oc)} = \frac{T_{3(oc)}}{x_{h(oc)}} = \frac{959+273}{1.76} = 700K = 427^{o}C \ (8)$$

where $x_{h(oc)}$ is the ratio of input-output temperatures for isentropic process. $\dot{m}_{oc}$ is the exhaust mass flow rate at operating condition. $\dot{m}_{no}$ is the exhaust mass flow rate at nominal operating condition. x_h is the specific heat ratio at hot end (combustor, turbine). The index (*oc*) and (*no*) mean operating conditions and nominal conditions. $T_{4s(oc)}$ and $T_{3(oc)}$ are exhaust temperature and turbine inlet temperature at operating conditions.

$$\eta_t = \frac{T_{3(oc)} - T_4}{T_{3(oc)} - T_{4s(oc)}} = \frac{959-487}{959-427} = \frac{472}{532} = 0.89 \ (9)$$

ii. Compressor Efficiency (n_c):

$$x_{c(oc)} = \left(PR \times \frac{\dot{m}_{oc}}{\dot{m}_{no}}\right)^{\frac{x_c - 1}{x_c}} = \left(9.87 \times \frac{124.1}{124.1}\right)^{\frac{1.4-1}{1.4}} = 1.92 \ (10)$$

$$T_{2s(oc)} = T_{1(oC)} \times x_{c(oc)} \quad (11)$$

$$T_{2s(oc)} = (26+273) \times 1.92 = 574.08K = 301.08^{o}C$$

(12)

The compressor outlet air temperature which is given (Table 2) by:

$$T_{2(oc)} = 367^{o}C$$

$$\eta_c = \frac{T_{2s(oc)} - T_1}{T_{2(oc)} - T_1} = \frac{301.08-26}{367-26} = .0.81 \quad (13)$$

where $x_{c(oc)}$ is the ratio of input-output temperatures for isentropic compression, x_c is the specific heat ratio at hot end (combustor, turbine). $T_{2s(oc)}$ and $T_{2(oc)}$ are compressor discharge temperature and compressor outlet temperature at operating conditions. $T_{1(oc)}$ is the ambient temperature at operating condition, and T_1 is the ambient temperature. η_{comb} is the combustion efficiency. A value of 0.99 is assumed for combustion system [3]. Since it is near unity, it has been chosen in this context as unity. C_{ph} is the specific heat of hot end (turbine).

Parameters will be computed for mechanical power based on nominal value:

$$x_h = (PR)^{\frac{x_h - 1}{x_h}} = (9.87)^{\frac{1.33-1}{1.33}} = 1.76 \quad (14)$$

$$x_c = (PR)^{\frac{x_c - 1}{x_c}} = (9.7)^{\frac{1.4-1}{1.4}} = 1.93 \quad (15)$$

The values of the coefficients of the output torque A and B are obtained as follows:

$$A = \frac{\dot{m}_n T_1}{P_{Gn}}\left\{C_{ph} \times \eta_t\left(1 - \frac{1}{x_h}\right) - \frac{x_c - 1}{x_c} \times \left[C_{pc} - C_{ph} \times \eta_t\left(1 - \frac{1}{x_h}\right)\right]\right\} \ (16)$$

$$= \frac{124.1 \times (273+15)}{26300} \times \left\{ \begin{array}{l} 1.1569 \times 0.89\left(1 - \frac{1}{1.76}\right) - \frac{1.93-1}{0.81} \\ \times\left[1.0047 - 1.1569 \times 0.89\left(1 - \frac{1}{1.76}\right)\right] \end{array} \right\}$$

$$A = -0.2685$$

$$B = \frac{\eta_{comb} \times \eta_t \times \dot{m}_{fn}}{P_{Gn}}\left[\left(1 - \frac{1}{x_h}\right)\right] \quad (17)$$

$$B = \frac{1 \times 0.89 \times 43309 \times \dot{m}_{fn}}{26300}\left(1 - \frac{1}{1.76}\right) = 0.6329\dot{m}_{fn}$$

$$\dot{m}_{fnpu} = \frac{P_{Gpu} - A}{B} = \frac{1+0.2685}{0.6329\dot{m}_{fn}} = 1 \Rightarrow \dot{m}_{fn} = 2.004kg/s$$

(18)

$$\Rightarrow B = 1.2683$$

where $\dot{m}_n$ is the air nominal flow rates, $\dot{m}_{fn}$ is the fuel nominal flow rates, P_{Gn} is the nominal output power of gas turbine and it is equal to the electrical power output at nominal condition (see Table 1) , P_{Gpu} is the per unit (p.u.) output power which is equal to the p.u. torque.

It should be noted that the values of C_{pc}, C_{ph} is the specific heat of cold end (turbine), C_{ph} is the specific heat of hot end (turbine) , x_c, and x_h [11]. There values are assumed to be 1.0047kJ/kgK, 1.1569kJ/kgK, 1.4, and 1.33, as a common approach to be employed for the cold end and hot end air properties [14].

In order to determine the turbine exhaust temperature, the exhaust temperature parameters D and E are computed using nominal flow.

$$D = \eta_{comb} \frac{H}{C_{ph}} \times \frac{\dot{m}_{fn}}{\dot{m}_n} \left[1 - \left(1 - \frac{1}{x_h} \right) \eta_t \right] \qquad (19)$$

$$D = 1 \times \frac{43309}{1.1569} \times \frac{2.004}{124.1} \left[1 - \left(1 - \frac{1}{1.76} \right) \times 0.89 \right] = 372.19^o C$$

Table 3 presents the simulation parameters used in this paper and their values obtained from the gas turbine nominal performance specification data and average operational data.

Table 3 Gas turbine simulation parameters

Parameter	Description	Value
Max	Fuel demand upper limit (p.u)	1.5
Min	Fuel demand lower limit (p.u)	-0.1
a	Valve positioner	1
b	Valve positioner	0.05
c	Valve positioner	1
W_{min}	Minimum fuel flow	0.23
T_f	Fuel control time constant (s)	0.4
K_f	Fuel system feedback	0
E_{CR}	Combustion reaction time delay (s)	0.01
E_{TD}	Turbine exhaust delay (s)	0.04
T_{CD}	Compressor discharge volume time constant (s)	0.2
A	Gas turbine torque bock parameter	-0.2685
B	Gas turbine torque bock parameter	1.2683
C	Gas turbine torque bock parameter	0.5

Assumptions made in Table 3:
It should be noted that the value for the output torque coefficient C varies between 0.5 and 0.67 for heavy duty gas turbine (HDGT) [11]. In this work, C is assigned a value of 0.5. A value of 1.5 p.u. is commonly used for maximum fuel demand while minimum fuel demand value can be determined by operational data for the fuel system [1]. There is a small time delay between the fuel injection and when heat is released in the combustor, which is called combustion reaction delay, E_{CR}. In modern gas turbine systems, the order is of milliseconds [1]. It is assumed in this paper as 10 milliseconds (0.01s). Also there is time delay between the fuel combustion and exhaust system, which is called $E_{TD.}$ It 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 millisecond delay for air and combustion products transfer to the temperature measuring point is assumed [11]. In this paper a value of 0.04s is assumed. There is relatively higher time delay existing in the compressor discharge path to the turbine inlet (T_{CD}). A value of 0.2 is assumed considering the thermodynamic properties [15] and the approach used in [11]. The fuel system feedback is assigned a value of zero (0) [1],[11].

A. *Simulink Model of Gas Turbine components*

The mathematical equations representing the dynamic model of a single shaft gas turbine is transformed into its equivalent Simulink block model. The Simulink model of the components of a gas turbine system considered in this paper are presented in Fig. 6, 7 and 8.

i. The Fuel System:

This unit comprises the fuel valve and actuator. Fuel injection into a gas turbine is determined by the valve positioner whose activity is controlled by the speed controller. A typical valve positioner transfer function is given by [3]:

$$V(s) = \frac{a}{bs + c}$$

(20)

where a, b, and c are the valve positioner constants.
The fuel system actuator transfer function is given by [3]:

$$F(s)=\frac{1}{T_{fc}s+1}$$
(21)

where T_{fc}, is the fuel system actuator time constant in seconds.

The Simulink block diagram of the fuel system is represented in Figure 3.4 below.

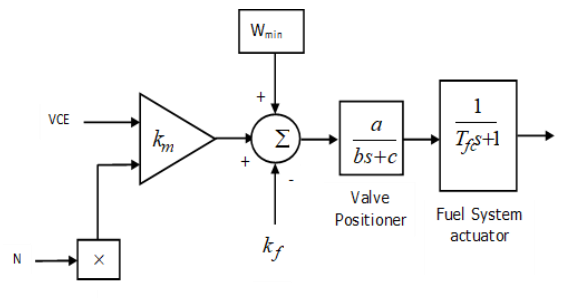

Fig. 6 Simulink block of the fuel system

VCE is the output of the least value gate (LVG) that governs the least amount of fuel needed for a given operating point and also an input to the fuel system. N is the per unit turbine speed which is also an input to the fuel system. W_{min} is the minimum amount of fuel flow. k_m is equal to $1-W_{min}$ and k_f is the fuel system feedback.

i. Compressor-Turbine Dynamics:

The compressor-turbine is often referred to as the heart of the gas turbine. It has a small transport delay associated with the combustion reaction time given in Eq. (22), a time lag given as expression which associated with the compressor discharge volume and transport delay given as expression, and for transport of gas from the combustion system through the turbine.

The burning of the fuel in the combustor is presented by the following function:

$$C_{TD}=e^{-sT_{CR}}$$
(22)

where T_{CR} is the combustion reaction time delay constant in seconds.

The transfer function of the hot computation gas expansion is expressed as follows:

$$T(s)=\frac{1}{T_{CD}s+1}$$
(23)

where T_{CD} is the compressor discharge volume time constant in seconds.

The compressor-turbine transport delay is given by:

$$CT_{td}=e^{-sT_{CR}}$$
(24)

The block diagram of the compressor-turbine is shown in Fig. 7.

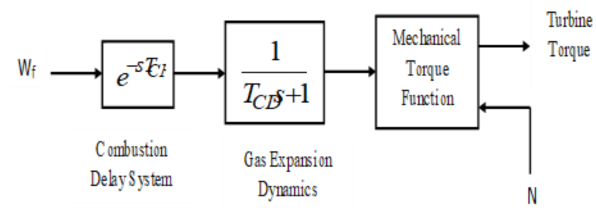

Fig. 7 Block diagram of compressor-turbine dynamics

The mechanical torque in Nm produced which drives the electric generator is presented by the following equation [3].

$$T_m = A+B\dot{m}_f+C(1-N) \qquad (25)$$

The Simulink block diagram of the turbine dynamic is shown in Fig. 8.

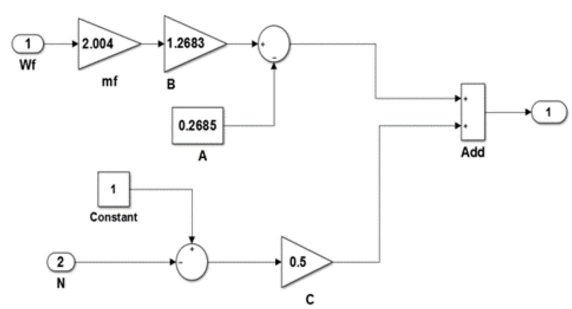

Fig. 8 Turbine dynamic block model

The alphabets A and B are the coefficients of output torque which could be obtained by applying the data in Table 3. N is the per unit rotor speed used for the purpose of simulation in this paper. The value for C in the torque Eq. (25) varies between 0.5 and 0.67 for heavy duty gas turbine (HDGT).

B. Controller Design and System Configuration

i.Controller Design

The design and implementation of Model Predictive Control (MPC) is carried out in MATLAB/Simulink environment. Implementing MPC requires that the following steps be followed:

a) A discrete step response model with length N and sample time Δt is developed.

The developed discrete step response model using the MATLAB/Simulink block of the Control and Estimation Manager Tool (CEMT) is given by:

$$\frac{0.0001281z^3 + 0.0008735z^2 + 0.0005106z + 2.481e-05}{z^4 - 2.521z^3 + 2.181z^2 - 0.7238z + 0.06393}$$
………. (26)

b) Specification of prediction and control horizon is established such that $N \geq P \leq M$; where N is the length or order of the discrete step response model, P is the Prediction horizon, and M is Control horizon or number of control moves.
c) Weighting w on the control action is specified.

The values of $N, \Delta t, P$, M, *and* w are presented in Table 4.

Table 4 Design Parameters

S/N	Parameter	Symbol	value
1	Length or order of the discrete step response model	N	4.0
2	Prediction horizon	P	1.0
3	Control horizon	M	1.0
4	Sample time	Δt	0.1s
5	Weighting on the control action	w	1.0

Also the constraints on manipulated (or control) variables are: Minimum = -inf and Maximum = inf. The designed MPC structure overview in MATLAB/Simulink is shown in Fig. 9.

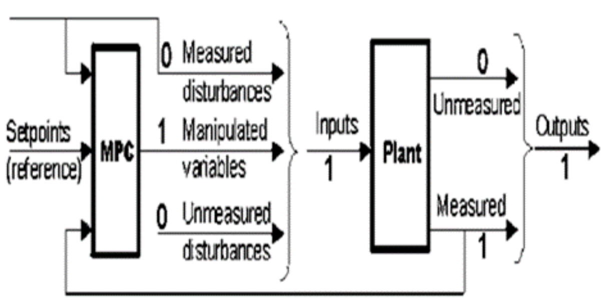

Fig. 9 Structure of the designed MPC in MATLAB/Simulink

ii. System Configuration

The closed loop structure of the Simulink model for speed control of gas turbine studied in this paper is shown in Fig. 10.

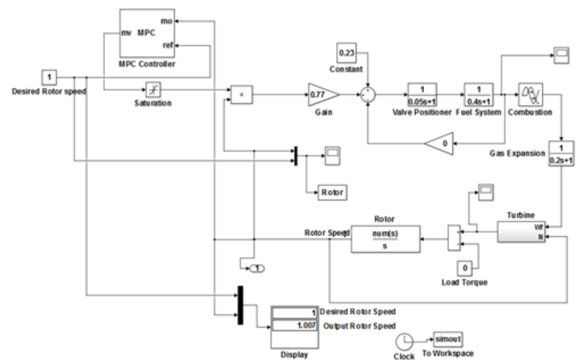

Fig. 10 Simulink model of the system

IV. RESULT ANALYSIS AND DISCUSSION

A. *Result Analysis*

The results from the simulations conducted in MATLAB/Simulink environment are presented in Fig. 11, 12, 13, and 14. Table 5 shows the performance analysis of the designed speed controller.

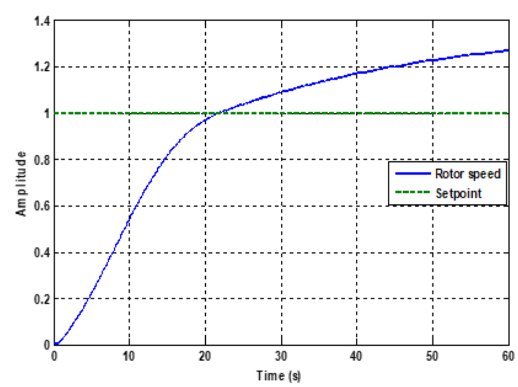

Fig. 11 Step response performance of gas turbine rotor speed (without MPC controller)

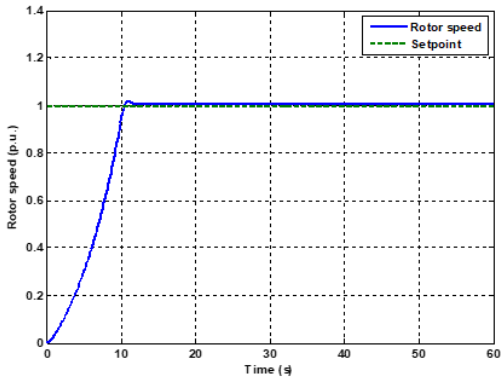

Fig. 12 Step response performance of gas turbine rotor speed (with MPC controller)

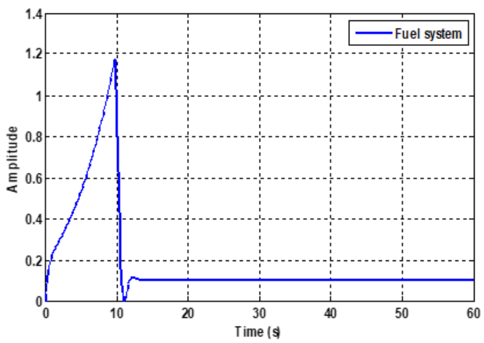

Fig. 13 Fuel demand system characteristics

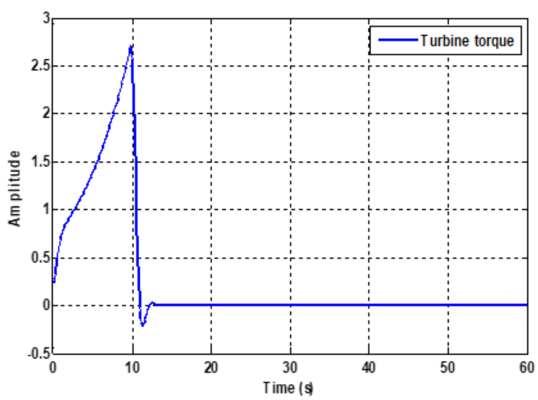

Fig. 14 Characteristics of load torque at nominal load

Table 5 Performance Analysis

Parameter	No-MPC	MPC
Setpoint (or reference) value	1.0 p.u	1.0 p.u.
Final (Response) value	1.270 p.u.	1.007 p.u.
Steady state error	0.270 p.u.	0.007 p.u.
Rise time	33.1 s	7.5035 s
Settling time	53.7 s	10.33 s
Maximum overshoot	0 %	0.82%

p.u. means per unit.

The result of simulations carried out for speed control of a gas turbine using MPC controller has been presented. The simulations are conducted on the bases that the system is operating at nominal condition. It should also be noted that the simulation is performed considering no external load (generator) connection to gas turbine. Figures 11 and 12 are the speed (or load frequency) step response performances of the turbine when it is operating at unit speed (1 p.u.) or 100 % full load without and with MPC controller. Figure 13 shows the fuel demand response characteristics at nominal operating condition. The load torque characteristic is shown in Fig. 14.

The operation of the speed control is based on the speed error formed when the HDGT rotor speed is compared with the referenced speed of one per-unit (1p.u.). The speed error or deviation is fed to the controller which in turns produces the control signal to the fuel system. Since the objective of this paper is to develop a Model Predictive Controller (MPC) that will improve the step response of a HDGT. With the nature of the transient response time shown in Fig. 11, it is obvious the system will sluggishly rise and even beyond the required rated load or full load (set point of 1p.u.). This, will to a large extent, affects the stability of the system at nominal operating condition. In order to address this problem and improve the performance of the system, MPC controller is introduced as part of the control loop. The simulation results showed that the introduction of the MPC controller significantly improved the step response time of the system from 20 s to 12.5 seconds by tracking the reference input and with an overshoot of 5 %. This indicates improved stability performance.

Table 5 shows the steady state error which is the difference between the reference input and the response value. For the speed control, the MPC achieved a steady state error of 0.007p.u. Since the steady state error is almost zero, it indicates that the designed MPC controller improved the transient response performance of the turbine speed and stabilizes the system at full load.

In Fig. 13, it can be seen that the fuel demand rises at the startup. The effectiveness of the MPC controller in this case is that it ensures that the fuel demand system is regulated and brought to minimum fuel level as soon as the system reaches full load. In Figure 14, the load torque increases as the speed (or load) increases. At about 10 seconds, the load torque drops to zero because gas turbine is now running on full load and it is steady.

CONCLUSION

This paper has presented speed control scheme for a 26 MW gas turbine. The main objective of this paper was to design a Model Predictive Controller (MPC) to improve the transient response characteristics performance of a gas turbine. In order to achieve the objective of this paper, parameters and continuous time dynamic equations of a 26 MW heavy duty single shaft gas turbine (HDSSGT) were obtained and analyzed. These equations were transformed into their equivalent Simulink blocks. MPC controller was developed in MATLAB/Simulink for the speed control. The parameters used for the simulation were obtained from calculations done using the nominal values and average operating data of a 26 MW gas turbine. The simulation results obtained showed that at nominal speed (1 p.u.), the load frequency performance was taken care of with settling time of 10.33 seconds and at this point, the system stabilizes and with overshoot of 0.82%.

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