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LOAD-FREQUENCY CONTROL FOR HEAVY-DUTY GAS TURBINE

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ABSTRACT

The load speed control of the heavy duty has turbine is the focus of this research work. The method employed in work is load-frequency control of the heavy-duty gas turbine using PID control, fuzzy logic control and the Fuzzy-PD+I control techniques through their designed controllers. This study investigates the performance of different controllers for load control of heavy-duty gas turbines. A simulation study was conducted using MATLAB/Simulink to evaluate the performance of the controllers. The gas turbine model used was a single-shaft heavy-duty gas turbine with a rated power of 33.4 MW. The controllers were designed and tuned to achieve optimal performance. The controllers used are PID, Fuzzy, and Fuzzy-PD+I. The results show that the FUZZY PD+I controller performs better than the PID and Fuzzy controllers in terms of load management and stability. The study concludes that the Fuzzy-PD+I controller is a suitable choice for load control of heavy-duty gas turbines due to its improved load control and stability. with rise time of 2.6660s, settling time of 5.5726s and overshoot of 2.3082% at 50% load torque, and rise time of 2.6517s, settling time of 7.3710s and overshoot of 3.1252% at 100% load torque. It recommended the use of Fuzzy-PD+I controllers for load control of heavy-duty gas turbines, it also optimize the controller gains using simulation studies, and implement the controller in a real-time environment.

KEYWORDS: Load Control, Gas Turbine, PID, Fuzzy, Fuzzy-PD+I

INTRODUCTION

Electricity is life to nations and to the industries. The growth of a nation is a function of electricity after education. The availability of electricity is the beginning of any nation's industrialization. Electrical power is the first key to any nation industrialization. Engineering and technological activities revolve around electrical power availability. University and research activities are carried out with the help of electricity availability. Electricity is part and parcel of our modern life and the most vastly used form of energy.

Adequate power supply is an unavoidable prerequisite to any nation's development and electricity generation, transmission and distribution are capital-intensive requiring huge resources of both funds and capacity (Sambo et al, 2020).

Electricity is one of the greatest innovations of mankind. It has now become a part of our daily life and one cannot think of the world without electricity (National Council of Education Research and Training, 2022).

The availability of electricity helps countries to ascent to faster economic growth trajectories (Rohan and Paul, 2018). There is a strong correlation between socio-economic development and the availability of electricity, and this is widely acceptable. Electricity plays a very important part in the socio-economic and technological development of every nation (Ibrahim and Salisu, 2011). The availability of electricity is the beginning of the journey to nation's development and economic growth. The life-wire of technology and innovation is availability of electricity.

Every sector of the economic and nation affairs depends on the electricity directly or indirectly. Electricity enhance day to day programmes and activities of a community or nation. Homes in the rural or in the urban area depends on electricity for smooth operations and wellbeing of the home. Electricity is the most remarkable invention because it has transformed every aspect of our daily lives. Electricity has revolutionized the world, from powering our homes and workplace to improving communication and transportation (Valerie, 2023).

Electricity is a form of energy that results from the flow of electric charge (Valerie Forgeard, 2023). The power stations even need electricity to start the turbines and machines for the electrical power generation.

Gas turbine is a high-tech machine used for electric power generation, to drive mechanical loads and for aerodynamics. Heavy Duty Gas Turbine (HDGT) are capable of burning a variety of fuels, ranging from natural gas to heavy gas liquid residuals. Heavy duty gas turbines are large sized, industrial gas turbines which are generally used in power generation in big power plants.

METHODOLOGY

The method employed for this work is load-speed control using PID, Fuzzy and Fuzzy PD+I – control techniques. A simulation study was conducted using MATLAB/SIMULINK to evaluate the performance of the controllers. The gas turbine model used was a single-shaft heavy-duty gas turbine with a rated power of 33.4 MW. The controllers were designed and tuned to achieve optimal performance. It is important to note also that in course of the research work many controllers were employed but this article is limited to PID, fuzzy and fuzzy PD+I.

Development of the Dynamic Models for the Load Control

The different dynamic models employed for the load control are considered in this section; fuel system dynamics, compressor-turbine dynamics, turbine and compressor efficiency.

Dynamic Model of Gas Turbine

The control loop mainly active when the gas turbine is operating at nominal conditions is the load-frequency control (LFC). The input to this control is the speed/load set-point. The control section is restricted by the minimum fuel limit. The speed control or LFC is also the most important aspect during stability study. The dynamics of the parts of the gas turbine that are essentials in analyzing the transient characteristics of a load-frequency control are considered as follows.

A Fuel System Dynamics

The fuel system consists of the fuel valve and actuator. The amount of fuel injected for a gas turbine is determined by a valve positioner controlled by the turbine speed controller. The valve positioner transfer function can be represented by the following transfer function given by:

$$\frac{V_p(s)}{E_{vp}(s)} = \frac{a}{bs + c} \tag{1}$$

where $V_p(s)$ output of the valve positioner, $E_{vp}(s)$ is the input , a, b, and c are the valve positioner constants.

The fuel system actuator transfer function is:

$$F(s) = \frac{1}{T_{fc}s + 1} \tag{2}$$

where T_{fc}, is the fuel system actuator time constant in seconds. The block diagram of the fuel system is represented in Figure 1.

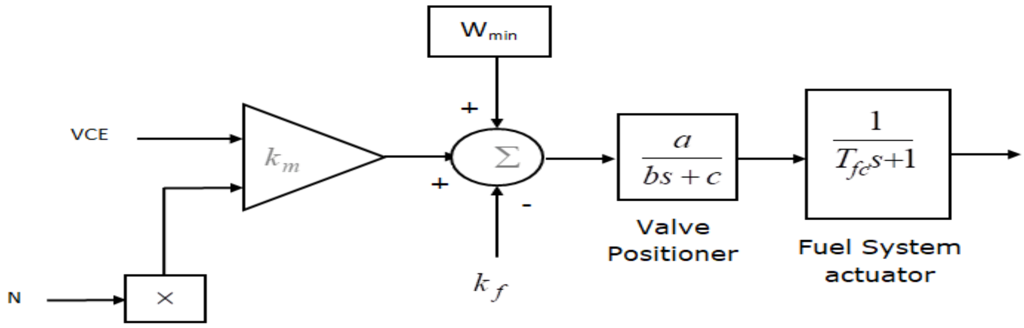

Figure 1: Block diagram of the fuel system Dynamics

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.

B Compressor-Turbine Dynamics

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

$$C_{TD} = e^{-sT_{CR}} \tag{3}$$

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:

$$T(s) = \frac{1}{T_{CD}S + 1} \tag{4}$$

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

The mechanical torque produced which drives the electric generator is given by BankTavakoli et al. (2009):

$$T_m = A + B\dot{m}_f + C(1 - N) \tag{5}$$

where, A and B are the coefficient of output torque which could be obtained by applying the actual data in Table 1. N is the per unit rotor speed. The value for C in the torque equation 5 varies between 0.5 and 0.67 for heavy duty gas turbine (HDGT) (BankTavakoli et al., 2009). In this work, the value of C is 0.5. The block diagram of the compressor-turbine dynamics is show in Figure 2.

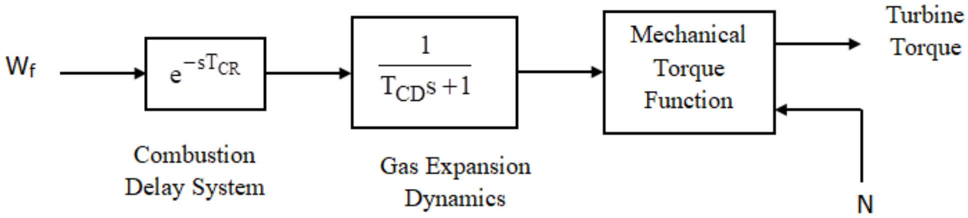

Figure 2: Block diagram of compressor-turbine dynamics

C Mathematical Computation of Turbine Parameter

A 33 MW single shaft heavy duty gas turbine (HDGT) whose specifications are given in Table 1 is considered in this work. It uses natural gas or distillate as its primary operating fuel. In this work, the computations performed did not take into account the pressure loss in the entrance air filters and also at combustor.

The design specifications in Table 1 are provided for nominal operation conditions. Table 2 provides typical operational data selected for computing the turbine and compressor efficiencies. The turbine and compressor efficiencies are inherently assumed to be constant in power output near nominal (Bank Tavakoli et al., 2009).

D Turbine and Compressor Efficiency

Using Table 1 and 2, turbine and compressor efficiencies can be computed using the detailed procedure and equations in (Bank Tavakoli et al., 2009) as follows:

Turbine efficiency (n_t):

$$x_{h(oc)} = \left(PR \times \frac{\dot{m}_{oc}}{\dot{m}_{no}}\right)^{\frac{\gamma_h - 1}{\gamma_h}} = \left(11.8 \times \frac{85.9}{139.1}\right)^{\frac{1.33-1}{1.33}} = 1.64 \tag{6}$$

$$T_{4s(oc)} = \frac{T_{3(oc)}}{x_{h(oc)}} = \frac{520+273}{1.64} = 483.5\text{K} = 165.5\,^{o}\text{C} \tag{7}$$

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

$$\eta_t = \frac{T_{3(oc)} - T_4}{T_{3(oc)} - T_{4s(oc)}} = \frac{520 - 250}{520 - 165.5} = \frac{270}{354.5} = 0.76 \tag{8}$$

Compressor Efficiency (η_c):

$$x_{c(oc)} = \left(PR \times \frac{\dot{m}_{oc}}{\dot{m}_{no}}\right)^{\frac{\gamma_c - 1}{\gamma_c}} = \left(11.8 \times \frac{85.9}{139.1}\right)^{\frac{1.4-1}{1.4}} = 1.76 \tag{9}$$

$$T_{2s(oc)} = T_{1(oC)} \times x_{c(oc)} \tag{10}$$

Equation (10) yields:

$$T_{2s(oc)} = (26 + 273) \times 1.76 = 526.24K = 253.24\,^{o}C$$

The compressor outlet air temperature which is computed as:

$$T_{2(oc)} = T_{3(oc)} - \eta_{comb} \frac{\dot{m}_{oc}}{\dot{m}_{no}} \times \frac{LHV}{C_{ph}} \tag{11}$$

$$= 520 - 0.99 \frac{85.9}{139.1} \times \frac{43309}{1.1569} = 517.71\,^{o}C$$

$$\eta_c = \frac{T_{2s(oc)} - T_1}{T_{2(oc)} - T_1} = \frac{253.24 - 26}{517.71 - 26} = .0.47 \tag{12}$$

Where $x_{c(oc)}$, γc are the ratios of input-output temperatures for isentropic compression, exhaust mass flow rate, and specific heat ratio at the hot end (combustor, turbine). $T_{2s(oc)}$ and $T_{2(oc)}$ are compressor discharge temperature and compressor outlet temperature at operating conditions. T_1 is the ambient temperature. η_{comb} is the combustion efficiency. A value of 0.99 is assumed for the combustion system (Rowen, 1983). 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 block in Equation 5 based on nominal value:

$$x_h = (PR)^{\frac{\gamma_h - 1}{\gamma_h}} = (11.8)^{\frac{1.33-1}{1.33}} = 1.84 \quad (13)$$

$$x_c = (PR)^{\frac{\gamma_c - 1}{\gamma_c}} = (11.8)^{\frac{1.4-1}{1.4}} = 2.02 \quad (14)$$

The value of A and B in Equation 5 can be extracted as follows:

$$A = \frac{\dot{m}_n T_1}{P_{Gn}} \left\{ C_{ph} \times \eta_t \left(1 - \frac{1}{x_h}\right) - \frac{x_c - 1}{x_c} \times \left[C_{pc} - C_{ph} \times \eta_t \left(1 - \frac{1}{x_h}\right)\right]\right\} \quad (15)$$

$$A = \frac{139.1 \times (273 + 15)}{33400} \times \left\{ \begin{array}{l} 1.1569 \times 0.76 \left(1 - \frac{1}{1.84}\right) - \frac{2.02 - 1}{0.70} \\ \times \left[1.0047 - 1.1569 \times 0.76 \left(1 - \frac{1}{1.84}\right)\right] \end{array} \right\} = -0.4978$$

$$B = \frac{\eta_{comb} \times \eta_t \times \dot{m}_{fn}}{P_{Gn}} \left[\left(1 - \frac{1}{x_h}\right)\right] \quad (16)$$

$$B = \frac{1 \times 0.76 \times 43309 \times \dot{m}_{fn}}{33400} \left(1 - \frac{1}{1.84}\right) = 0.4504\, \dot{m}_{fn}$$

$$\dot{m}_{fpu} = \frac{P_{Gpu} - A}{B} = \frac{1 + 0.4978}{0.4504 \dot{m}_{fn}} = 1 \Rightarrow \dot{m}_{fn} = 3.33 kg/s \quad (17)$$

$$\Rightarrow B = 1.5002$$

where $\dot{m}_n$ and $\dot{m}_{fn}$ are the air and fuel nominal flow rates, P_{Gn} and P_{Gpu} are the nominal output power and 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} , γ_c, and γ_h are assumed as 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 (Walsh and Fletcher, 2004).

Proportional-Integral-Derivative (PID) Controller

The PID controller is the common controller largely used in industrial control systems for three-term closed loop feedback mechanism and it is shown in Figure 3.

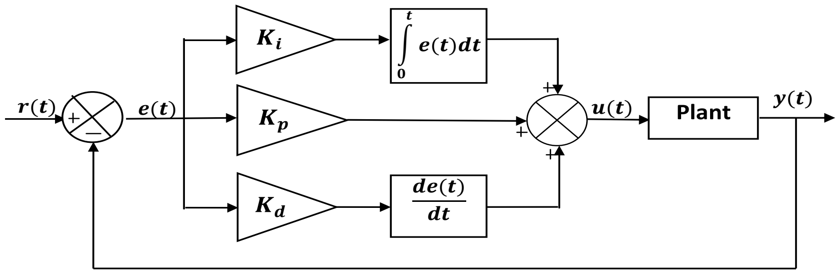

Figure 3: PID control system representation (Zurich In)

The mathematical expression of the PID controller can be determined by analyzing Figure 3. Hence, $r(t), e(t), u(t)$ represents the desired input, error, and the control command. Furthermore, K_p, K_i, K_d are gains of the PID controller for the proportional element, integral element, and derivative components, and y(t) is the output.

$$e(t) = r(t) - y(t) \tag{18}$$

The proportional, integral, and derivative computation carried out on the error as it is fed into the PID controller results in a control action given by equation(19):

$$u(t) = K_p e(t) + K_i \int_0^t e(t)dt + K_d \frac{de(t)}{dt} \tag{19}$$

Equation (20) is an ideal PID controller in the continuous time domain. Thus, the Laplace transform of the PID controller assuming zero initial condition is given by:

$$U(s) = K_p E(s) + K_i \frac{1}{s} E(s) + K_d sE(s) \tag{20}$$

Or in a simplified form as:

$$C(s) = K_p + K_i \frac{1}{s} + K_d s \tag{21}$$

where $C(s) = U(s)/E(s)$ and is the PID controller. Hence, the gains of the designed PID controller obtained by tuning the MATLAB/Simulink PID block are stated as follows, and the controller is given by Equation 22.

$$K_p = 10;$$
$$K_i = 1.26e-5;$$
$$K_d = 1.85.$$

$$C(s) = 10 + \frac{1.26e-5}{s} + 1.85s \tag{22}$$

Fuzzy Logic Control Design

The initial design of a fuzzy logic controller was carried out. The structure of the FLC for the gas turbine is shown in Figure 4. The system consisted of a fuzzy logic algorithm with two input variables: error (E), which is the difference in input-output of the load frequency and change in error (CE) is the rate of error in the load frequency, and an output U, which is the control variable that changes according to the changes in the inputs. The input-output relationship, represented by the linguistic variables that determine the control characteristics of the FLC according to the transformed crisp fuzzy input set by fuzzification. The linguistic variables based on the relationship of inputs to output are listed in Table 3 and are defined as follows: Negative Large (NL), Negative Medium (NM), Negative (NE), Zero (ZE), Positive Large (PL), Positive Medium (PM), and Positive (PO). There are three Membership Functions (MFs) for each input and five MFs for the output. The total number of formulated rules for the designed FLC is nine.

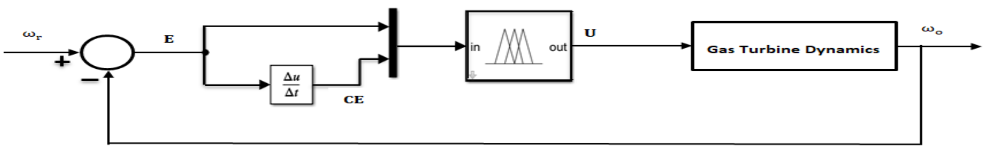

Figure 4: FLC system for gas turbine speed control

The design specifications of the heavy duty HDGT and the operating data and given in Table 1 and Table 2.

Table 1: Design Specifications of HDGT Selected.

Parameters	**Symbol**	**Unit**	**Value**
Electrical Power	P_{GT}	MW	33.4
Heat Rate	H_R	kJ/kwh	11302
Exhaust Temperature	T_{OE}	^oC	538
Exhaust Mass Flow	$\dot{m}_{no}$	Kg/s	139.1
Pressure Ratio	PR	-	11.8
Compressor Discharge Temperature		^oC	343

Source: (Palmer and Erbes, 1994)

Table 2: Typical Operating Data for Computing Turbine and Compressor Efficiencies

Parameter	Unit	Value
Output Power	MW	30.2
Turbine Inlet Temperature	^{o}C	520
Exhaust Gas Temperature	^{o}C	200
Ambient Temperature	^{o}C	26
Fuel Flow	Kg/s	0.99
Exhaust Mass Flow	Kg/s	85.9
Lower Heating Value of Fuel (LHV)	kJ/kg	43309

RESULTS AND DISCUSSION

From figure 5, figure 6, figure 7, figure 8, figure 9, figure 10, figure 11, figure 12 and figure 13 show different simulations result graphs for the PID controller, Fuzzy controller and Fuzzy PD+I controller.

Table 3.3 Rule table of fuzzy logic

E/CE	NE	ZE	PO
NE	NL	NM	ZE
ZE	NM	ZE	PM
PO	ZE	PM	PL

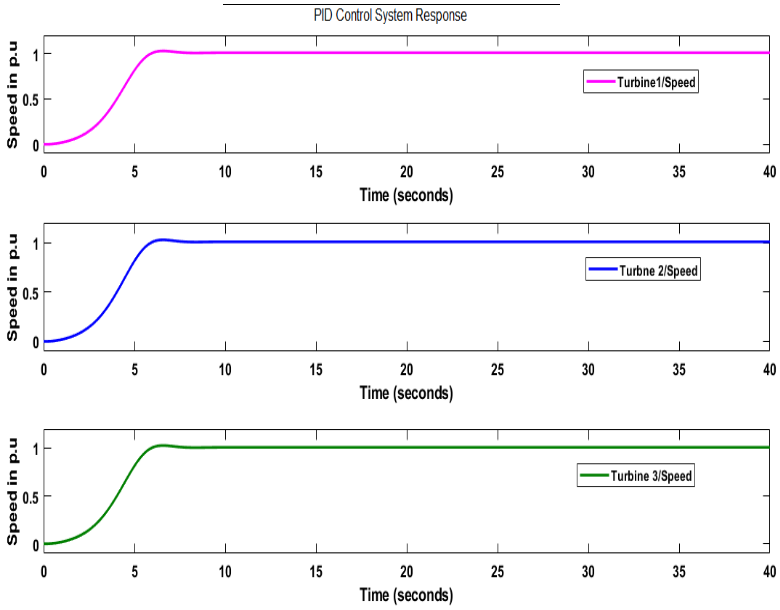

Figure 5: Speed response of PID controller.

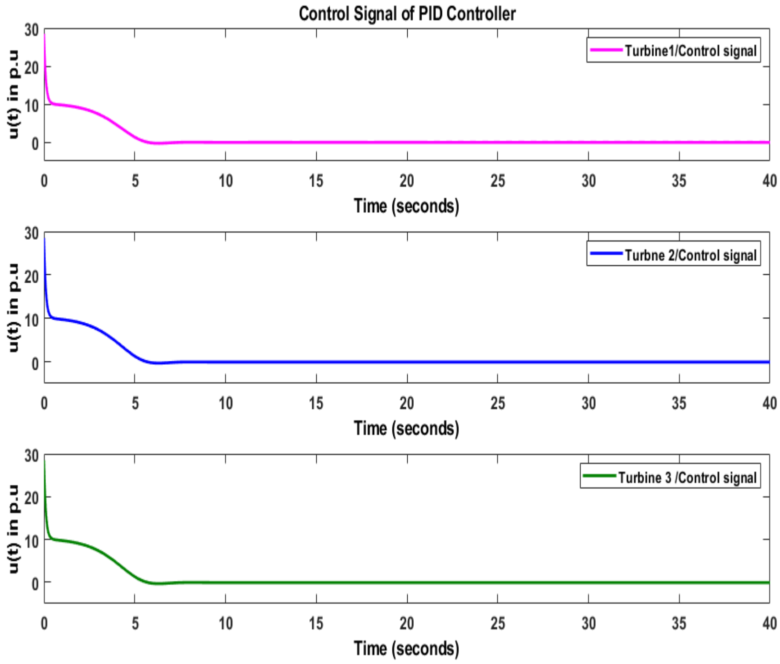

Figure 6: Droop response of PID controller.

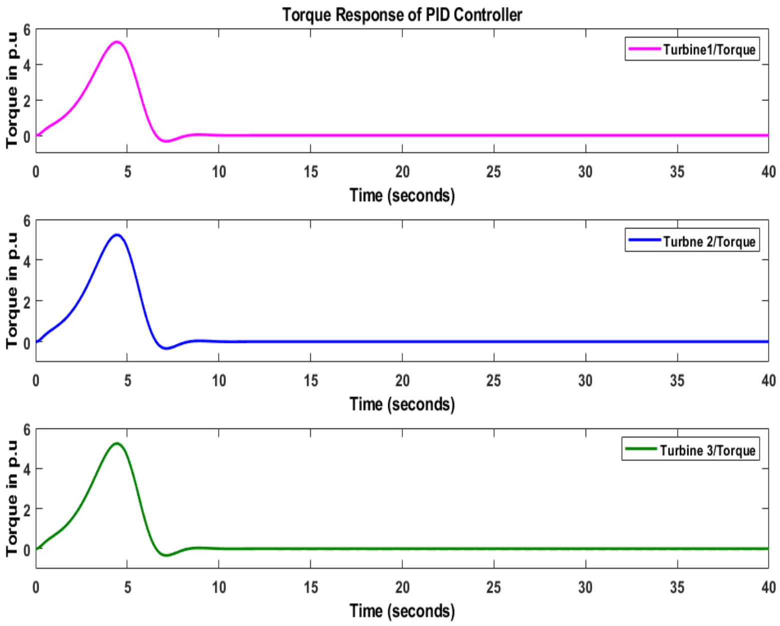

Figure 7: Torque response of PID controller.

Table 4: Dynamic response parameters of PID speed control system

Parameter	**value**
Rise time	3.2164 s
Settling time	5.8189 s
Peak time	6.5580 s
Overshoot	1.9484%
Peak value	1.0277

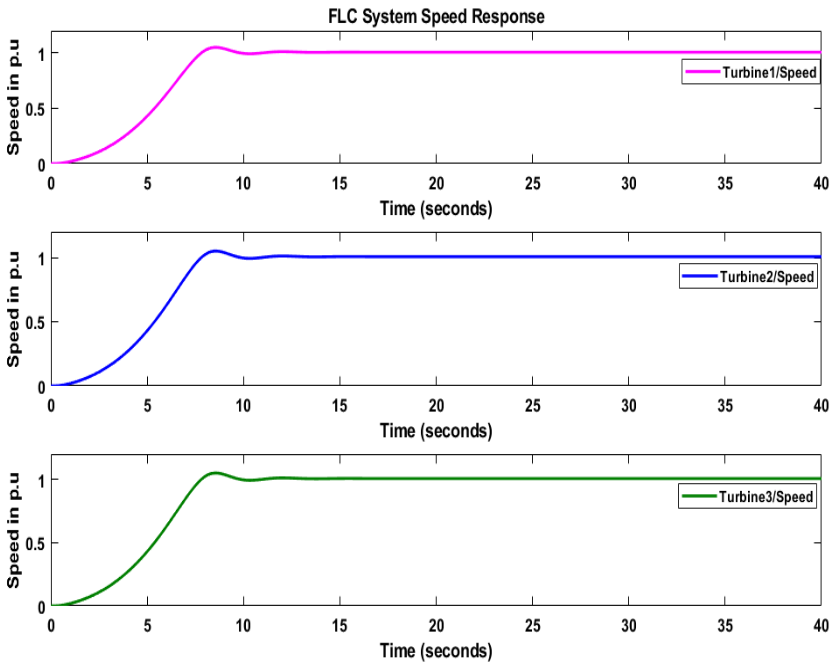

Figure 8: Speed response of FLC based LFC gas turbines.

Table 5: Dynamic response parameters of fuzzy speed control system

Parameter	value
Rise time	4.8663 s
Settling time	9.2326 s
Peak time	8.6025 s
Overshoot	4.3283%
Peak value	1.0502

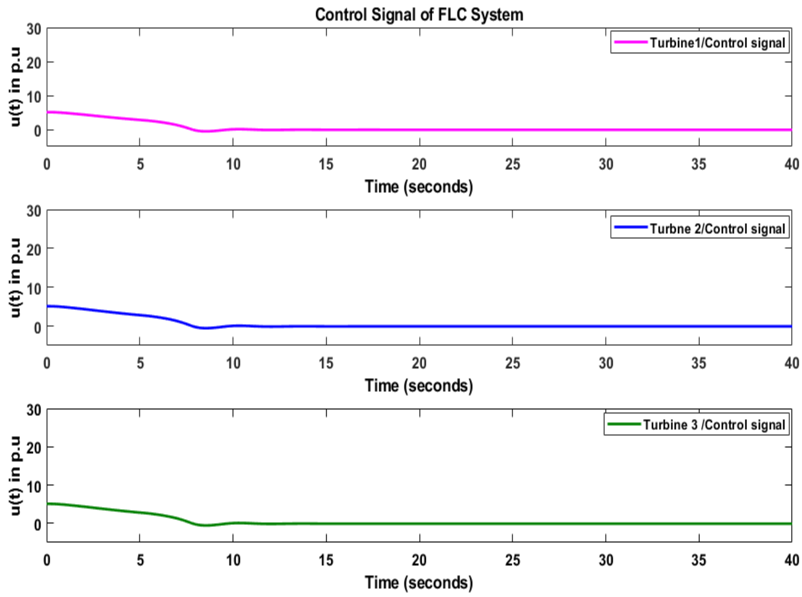

Figure 9: Droop response of FLC in LFC of gas turbines.

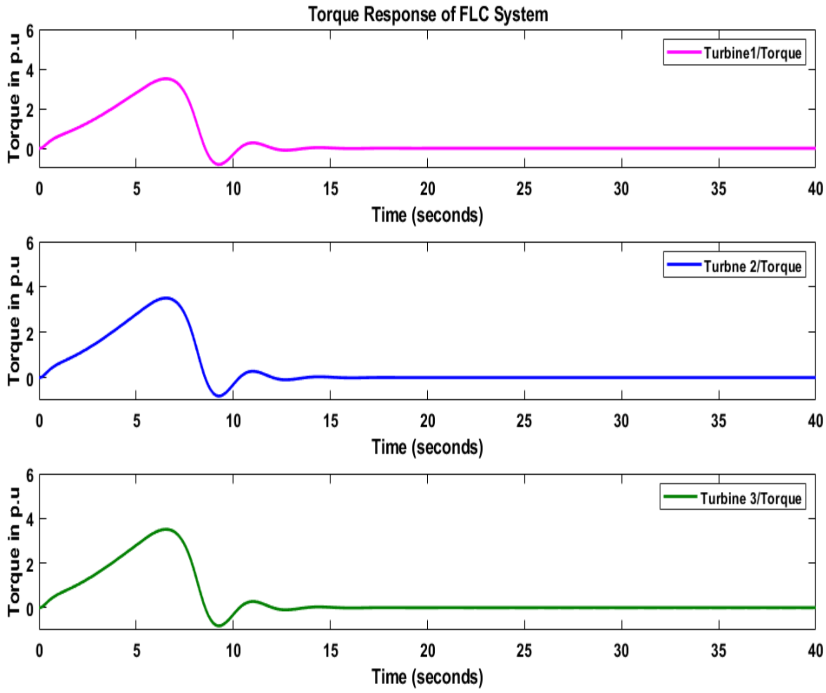

Figure 10: Torque response of FLC in LFC of gas turbines.

The designed FLC uses nine (9) logic rules to achieve speed control for the gas turbines. In Figure 10, which is numerically evaluated in Table 3, it can be deduced that the use of FLC offered significant reduction in overshoot, while the PID outperformed the designed FLC in all the performance indices measured in terms of dynamic response parameters listed in the table.

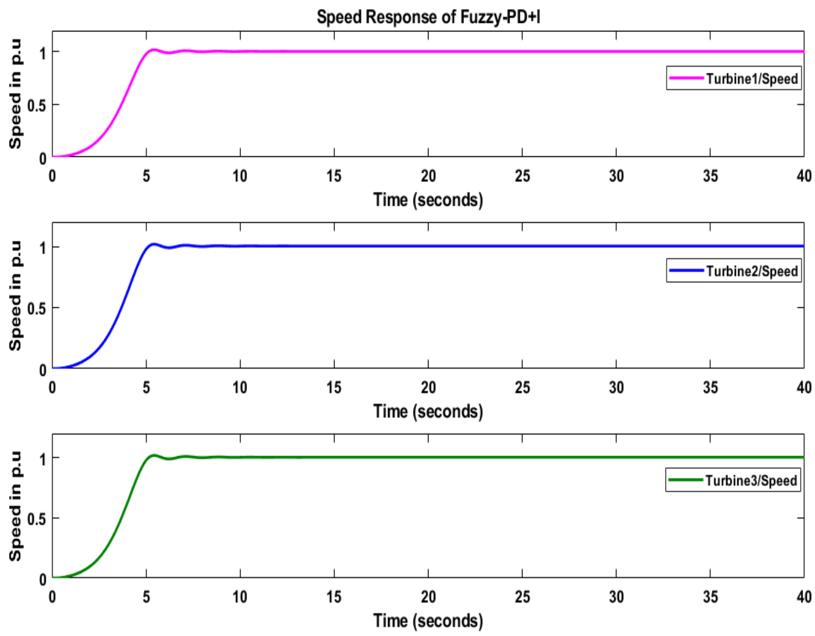

Figure 11: Speed response of Fuzzy-PD+I based LFC gas turbines.

Table 6: Dynamic Response Parameters of Fuzzy-PD+I Speed Control System.

Parameter	value
Rise time	2.6786 s
Settling time	5.0325 s
Peak time	5.4280 s
Overshoot	1.5161%
Peak value	1.0181

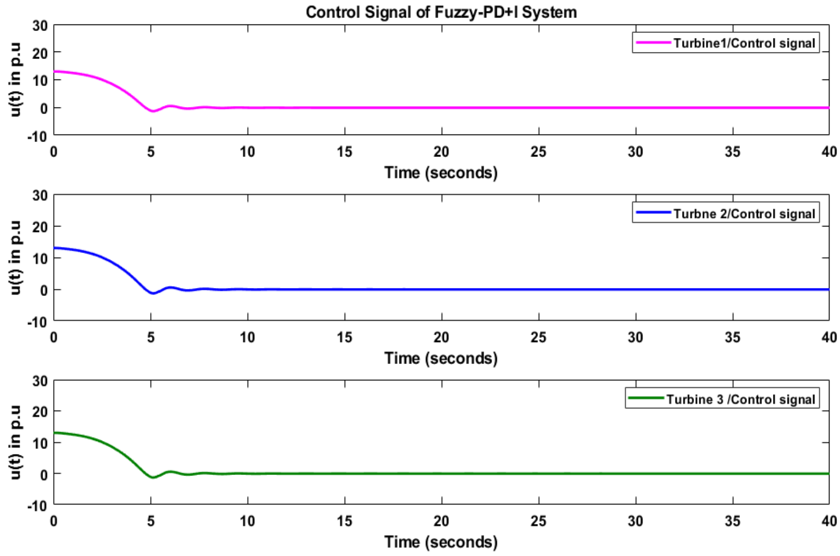

Figure 12: Droop response of Fuzzy-PD+I in LFC of gas turbines.

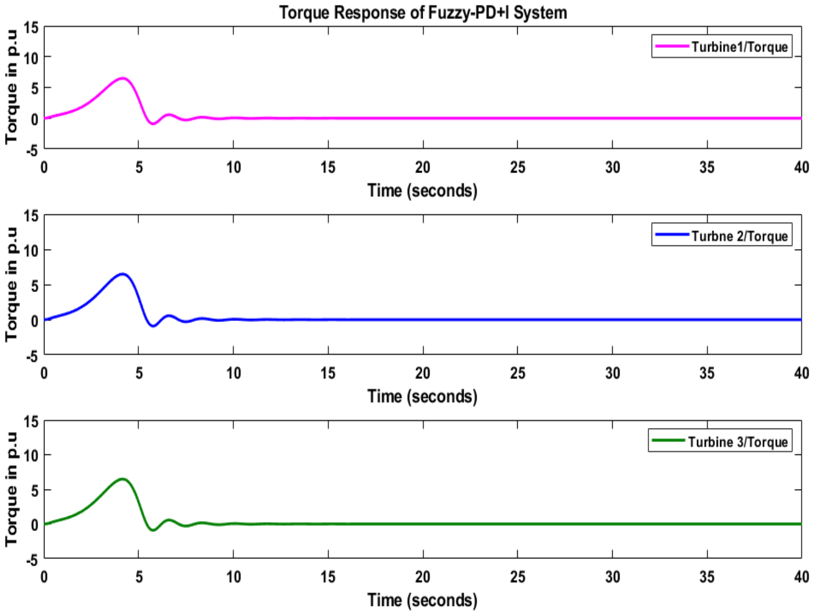

Figure 13: Torque response of Fuzzy-PD+I in LFC of gas turbines.

Table 6: Performance comparison at no load.

Control system	Rise time (s)	Settling time (s)	Peak time (s)	Overshoot (%)	Peak value (p.u.)	Peak value u(t) (p.u.)	Peak value, T_g (p.u.)
PID	3.2164	5.8189	6.5580	1.9484	1.0277	28.5000	5.2326
FLC	4.8663	9.2326	8.6025	4.3283	1.0502	3.5080	5.1000
Fuzzy-PD+I	2.6786	5.0325	5.4280	1.5161	1.0181	12.9755	6.4733

Table 7: Speed dynamic response parameter at 50% loading of gas turbine.

Control system	Rise time (s)	Settling time (s)	Peak time (s)	Overshoot (%)	Peak value (p.u.)
PID	3.1576	10.0556	6.5636	4.1716	1.0279
FLC	4.8115	9.6093	8.5268	6.1912	1.0503
Fuzzy-PD+I	2.6660	5.5726	5.4280	2.3082	1.0183

Table 8: Speed dynamic response parameter at 75% loading of gas turbine

Control system	Rise time (s)	Settling time (s)	Peak time (s)	Overshoot (%)	Peak value (p.u.)
PID	3.1282	10.2753	6.5636	5.3432	1.0279
FLC	4.7820	9.8817	8.5268	7.2318	1.0503
Fuzzy-PD+I	2.6589	5.6583	5.4362	2.7113	1.0183

Table 9:Speed dynamic response parameter at 100% loading of gas turbine

Control system	Rise time (s)	Settling time (s)	Peak time (s)	Overshoot (%)	Peak value (p.u.)
PID	3.0992	10.3975	6.5636	6.5693	1.0279
FLC	4.7510	11.2729	8.5268	8.3761	1.0503
Fuzzy-PD+I	2.6517	7.3710	5.4362	3.1252	1.0183

Table 6 shows the performance of the three controllers at no load speed. Table 7, Table 8 and Table 9 show performance of the three controllers at 50%, 75% and 100% load torque respectively.

The results of this study demonstrate the superiority of the FUZZY PD+I controller over the PID and FUZZY controllers for load control of heavy-duty gas turbines as shown in Table 7, Table 8 and Table 9. The FUZZY PD+I controller's ability to adapt to changing operating conditions and its robustness to disturbances make it a suitable choice for this application.

CONCLUSION

The study concludes that the FUZZY PD+I controller is a suitable choice for load control of heavy-duty gas turbines due to its improved load control and stability with rise time of 2.6660s, settling time of 5.5726s and overshoot of 2.3082% at 50% load torque, and rise time of 2.6517s, settling time of 7.3710 s and overshoot of 3.1252% at 100% load torque.

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