Comparative Controller Design for a Marine Gas Turbine Propulsion System

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Controller design for marine gas turbine systems should consider three measures of performance: transient control, steady-state accuracy, and disturbance rejection. This paper presents and compares two common types of controller design in terms of these measures. The goal of the controllers was shaft speed control. To meet this goal, a classical Proportional-plus-Integral controller was designed and compared to a modern Linear Quadratic Regulator design. The controllers' performances were evaluated with respect to the three measures mentioned above, with disturbances being input as oscillations in shaft torque due to seaway cycling.

Introduction

Controller design is facilitated if a control tradeoff database can be established. Many such tradeoff tools are well understood for linear machinery (Ogata, 1970); they include rise time, overshoot, damping ratio, and stability margin in phase and gain. However, for nonlinear machinery, such as marine propulsion systems, a similar set of well-established general tools for controller design and development does not exist. Further, due to wide variations in nonlinearity classifications, nonlinear designs tend to be ad hoc since there is no unifying body of nonlinear control theory. The control designer is thus forced to proceed with only his two basic principles of proper control design: The controlled system should exhibit good command following (transient control and steady-state accuracy), and good disturbance rejection (resistance to seaway oscillations). His designs are usually developed in the time domain using cut-and-try methods with simulation as his only tool. He has little idea of what to expect when parametric changes are made and almost no idea of what to expect if a new concept is tried. Further confusion is introduced if a new variety of gas turbine is considered. This situation is aggravated by the fact that few comparative studies of control concepts have been published in the literature.

The two types of controllers used in the present comparison were both simple linear regulators: a classical Proportional-Integral (PI) controller and a modern Linear Quadratic Regulator (LQR). The PI regulator was designed after the concept suggested by Rubis and Harper (1982, 1986), which is shown in Fig. 1. Their regulator had two loops, an inner loop to regulate power produced by the gas generator and an outer loop to regulate propeller shaft speed. The inner loop had all the nonlinear safety limiting tests associated with shaft torque, acceleration rate, etc. The present study ignored these nonlinearities. The control of pitch ratio (PR) was also ignored in the present study. The structure of the study PI controller is shown in Fig. 2. Note that we replaced the power lever (PLA) reference signal with a control loop in gas generator speed. The figure shows that three gains needed to be chosen.

LQR controllers for aero gas turbines date from at least 1973 (Michael and Farrar); however, applications to marine systems are not well documented in the literature. One application was found that considered a General Electric LM2500 aboard a US Navy FFG-7 (Kalyn, 1979). The structure of such a controller is shown in Fig. 3. Note that there are now three regular loops, one on each of the system states of gas generator speed, shaft speed, and fuel energy. Again, three gains needed to be chosen.

Approach

The gas turbine system that was modeled is shown in Fig. 4. The figure shows a marine emulation test bed, which is comprised of a 175 hp gas turbine coupled to a water brake dynamometer. The modeling and simulation of this nonlinear machinery has been discussed elsewhere (Smith, 1988; Stam-

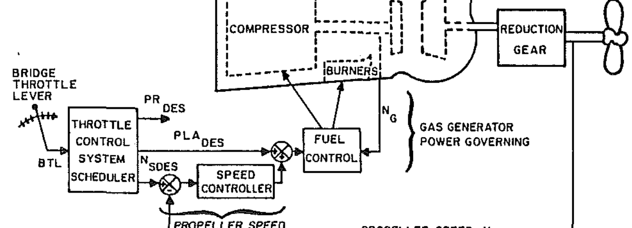

Fig. 1 Closed-loop propulsion control

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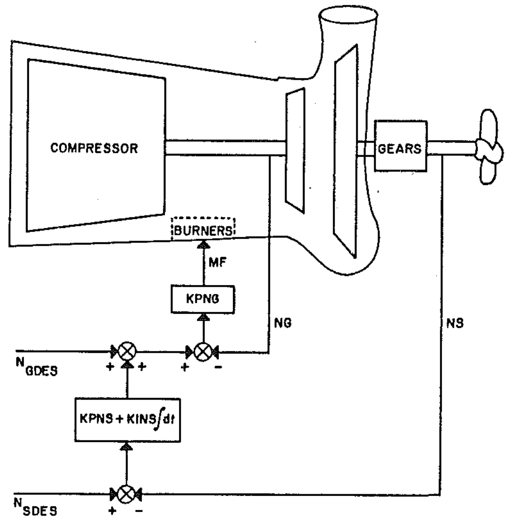

Fig. 2 Study PI controller

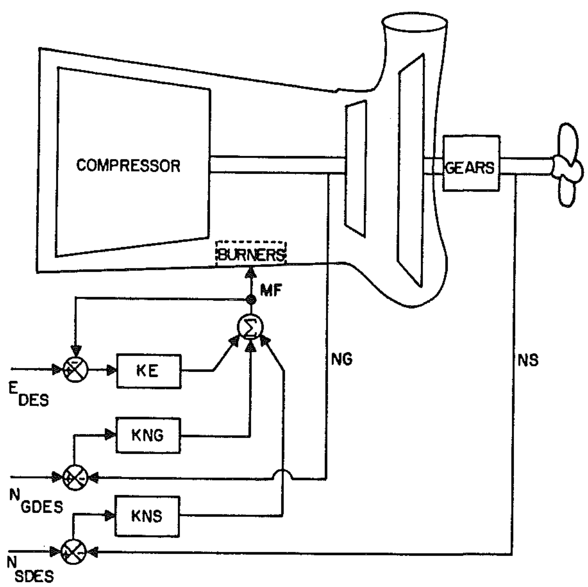

Fig. 3 Study LQR controller structure

metti, 1988) and will not be discussed further here. The situation we were attempting to model and control was that of a fixed pitch propeller system under shaft speed control.

The cause-and-effect plant model is shown in Fig. 5. The dynamometer was loaded at constant water volume V, and operated very much like a propeller loaded at constant pitch.

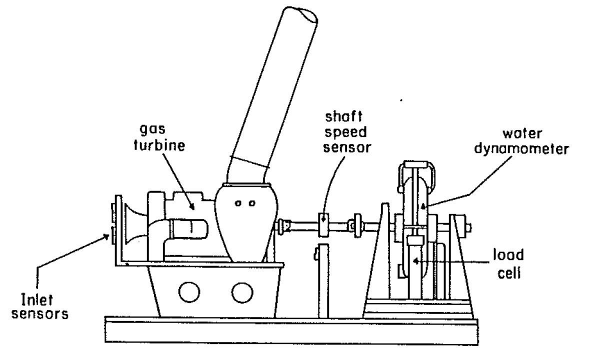

Fig. 4 Test bed installation

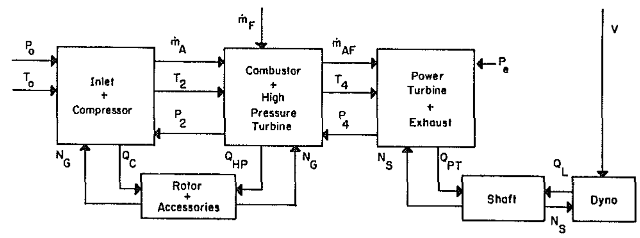

Fig. 5 Plant cause and effect model

The single input to the plant was the fuel flow rate. Given this plant model, and a computer simulation developed earlier, we first did a cut-and-try design for the PI regulator gains shown in Fig. 2. Our goal was to pick the three gains so that there was good command following. That is, we wanted no oscillation and quick response times in all transients, with small or zero steady-state error in desired shaft speed. We next designed the LQR using the same design criteria. The evaluation of disturbance rejection was accomplished by superimposing a small-amplitude torque oscillation on a steady-state condition and observing the responses of the two controlled systems.

PI Design

In our design cases, we were mostly interested in regulation of large-amplitude transients around the operating region. Consequently, we expected large changes in the plant dynamics to occur. As mentioned earlier, this meant that we were forced to do the PI design with a cut-and-try method, using simulation to evaluate our success. We used a traditional method of opening both loops, first designing and fixing the inner closed loop, and then designing the outer closed loop. The inner loop design is shown in Fig. 6. Note that deceleration (bottom curves) was less stable than acceleration (top curves) since the same gain values produced more oscillations in the first case. In keeping with this result, we subsequently used the deceleration case for our designs. Further, based on our desire to have no oscillations, we set the inner loop gain at KPNG = 0.001. Even though the chosen value of gain left

Nomenclature

$\mathbf{A}$ = state coefficient matrix
$\mathbf{B}$ = input coefficient matrix
$\mathbf{e}$ = error vector
$\mathbf{f}$ = vector of state functions
J = LQR performance measure
$\mathbf{K}$ = LQR gain matrix
$\dot{m}$ = mass flow rate, lb_m/hr
N = rotational speed, rpm
P = pressure, lb_f/in^2
Q = torque, ft-lb_f
$\mathbf{Q}$ = matrix of error weights
R = control weight
T = temperature, °F
u = gas turbine input
x = system state

Subscripts

A = airflow
AF = airflow plus fuel
C = compressor
e = exhaust
F = fuel
G = gas generator
HP = high-pressure turbine
i = linearization point
L = load
o = ambient
PT = power turbine
s = power shaft
1 = compressor inlet
2 = compressor discharge
4 = power turbine inlet

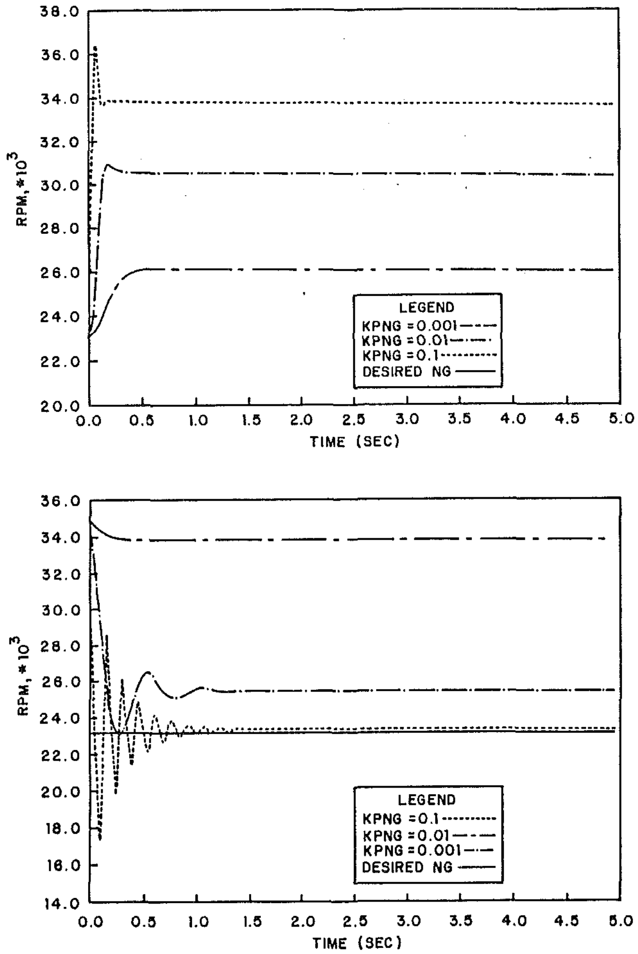

Fig. 6 PI inner loop design: acceleration (top) and deceleration (bottom)

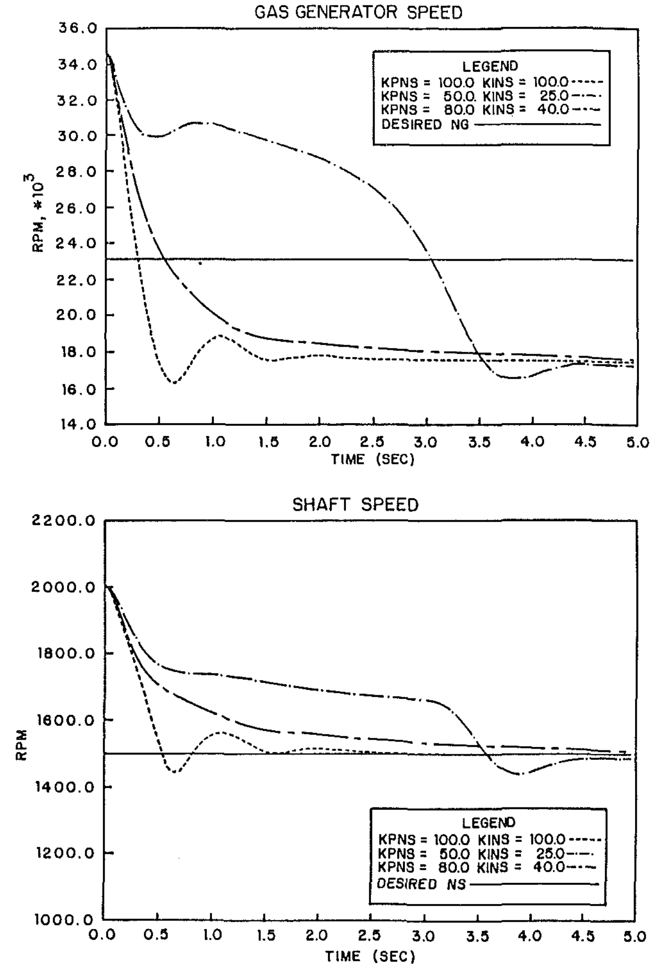

Fig. 7 PI outer loop design

quite a large steady-state error, this was really not a performance concern since we expected the integral control in the outer loop to dominate the system performance.

The outer loop design was more difficult since it involved two design degrees of freedom. The results of this design effort are shown in Fig. 7. Here we see that the outer loop integral action did dominate the inner loop regulator. With the gains chosen to coincide with the middle curve in the figures, we achieved a smooth transition to the desired steady-state shaft speed. The operation of the two loops now became clear; the gas generator would move to the required level to achieve the desired shaft speed regardless of the setting for the desired gas generator speed. This happened because of the summing junction connecting the two loops. In effect, the junction caused the reference setting for the gas generator to be constantly changed until the shaft speed integral regulator could be satisfied.

LQR Design

The LQR design process was a much more mathematical method, but it still required a cut-and-try approach since we used it on a nonlinear machine. The method started with the selection of a linearization point with which to model the propulsion plant. The point we chose was in the center of the operating region

$$N_G = 25{,}000 \text{ rpm}$$

$$N_S = 1500 \text{ rpm}$$

The propulsion dynamics were then linearized about this point into the following form:

$$\dot{\mathbf{x}} = \mathbf{A}\mathbf{x} + \mathbf{B}u \tag{1}$$

We then assumed, *for the purposes of gain determination only*, that this was an adequate general model of the plant. In our work, the input u was the fuel flow rate, and the states x were perturbations of the states discussed in the Approach section

$$x(1) = \delta N_G$$

$$x(2) = \delta N_S$$

$$x(3) = \delta E$$

Given this plant model, we sought to minimize a quadratic performance measure J for the linear regulator in the following form:

$$J = \int_0^\infty (\mathbf{e}^T\mathbf{Q}\mathbf{e} + uRu)dt \tag{2}$$

where

$$e(1) = N_{GDES} - N_G \tag{3}$$

$$e(2) = N_{SDES} - N_S \tag{4}$$

$$e(3) = E_{DES} - E \tag{5}$$

and

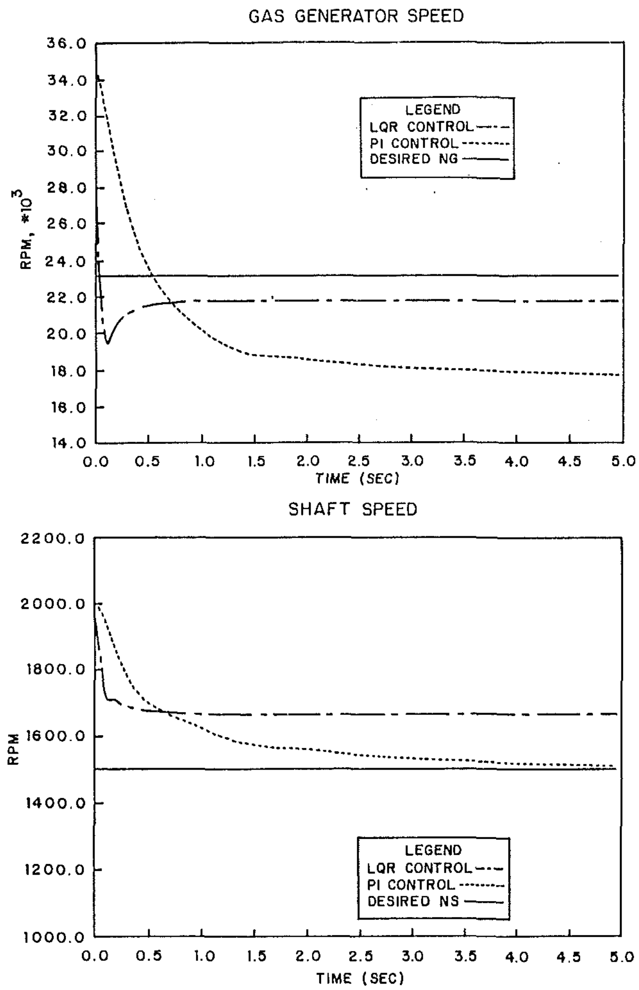

Fig. 8 Controller comparison

$$Q = \begin{bmatrix} Q(1,1) & 0.0 & 0.0 \\ 0.0 & Q(2,2) & 0.0 \\ 0.0 & 0.0 & Q(3,3) \end{bmatrix} \quad (6)$$

The solution to this minimization problem has been well documented in the literature for linear systems (Ogata, 1970). It takes the form

$$u = \mathbf{Ke} \quad (7)$$

where K is dependent on A and B (the plant model), and the designer's choices for Q and R. In this way, the J measure provides a mathematical framework with which to evaluate the performance of the controlled system. That is, the Q and R entries provide weighting for the error terms versus the input term. The use of relatively large Q entries causes the system to respond more quickly in order to minimize the error terms at the expense of input magnitude, while the use of a large R causes the controller to act to minimize the input at the expense of error.

The present LQR design method was thus one of arbitrarily choosing the Q and R entries, solving for the corresponding K entries, and then simulating the response of the nonlinear propulsion system.

Controller Comparison

The two most important features of controlled system performance are command following and disturbance rejection. In the present work, good command following was the criterion for good design and was further subdivided into the two areas of good transient control (no oscillations) and small steady state error.

Command Following. Figure 8 shows the simulated responses of the two final regulator designs for a nonlinear plant model. Clearly, the LQR regulator achieved its final steady state more quickly than the PI regulator. However, while neither regulator showed any oscillation, the LQR regulator did show some steady-state offset in shaft speed. This offset occurred because the system had no inherent integrating effect. The LQR regulator thus acted like a classical proportional controller that had steady-state offset error, as demonstrated in Fig. 8. It was found that the LQR steady-state offset could be reduced at the expense of a lower dip in gas generator speed (at about 0.1 s in the figure). It was finally decided that the curves shown represented the best tradeoff in these responses. (While the curves show about a 10 percent error in LQR steady-state shaft speed for the deceleration maneuver, an acceleration maneuver between the same end states showed only about a 1 percent error—a clear result of the plant nonlinearity.)

Disturbance Rejection. The ability of the final regulators to reject load oscillations due to seaway cycling was also evaluated. This was done by letting the regulated system come to steady state at

$$N_G = 25{,}000 \text{ rpm}$$

$$N_S = 1500 \text{ rpm}$$

and superimposing a moderate torque oscillation onto the load. A sinusoidal disturbance was used as follows:

$$\delta Q_L = 20 \sin(\pi t/5.0) \quad (8)$$

For the chosen steady-state condition, the total load torque was about 205 ft-lb_f. The amplitude of the disturbance was thus about 10 percent of the steady-state value. When the oscillating load was input to the regulator simulations, a periodic response in all variables was observed. Table 1 summarizes the results.

Table 1 Oscillating load comparison

	δN_G^1	δN_S^1	Fuel^2
LQR	3320.0	203.1	76.25
PI	4233.0	74.5	76.46

^{1}Peak-to peak values, rpm.
^{2}Total fuel consumed, lb_m/hr.

Table 1 shows that the final LQR regulator was less susceptible to seaway oscillations in N_G and fuel. The latter was true since the regulator design explicitly considered fuel weighting through the J equation (equation (2)). From another point of view, the PI regulator sought shaft speed regulation at the expense of the other variables. Hence, the PI gas generator oscillations were greater and more fuel was consumed in oscillation, while the shaft speed oscillations were smaller.

Conclusions and Recommendations

The PI regulator and the LQR regulator offered very similar performance for the simple system studied. The strength of the LQR regulator was in reducing gas generator oscillations and fuel consumed in a seaway. The strength of the PI regulator was in shaft speed regulation. The LQR regulator also got to its final steady state more quickly, thus implying a quicker ship response.

In retrospect, the LQR design methodology was more appealing. The selection of weighting for the LQR design variables was much more satisfying than the direct choosing of PI loop gains, even though both must be approached on a cut-and-try basis. The LQR approach also allowed us explicitly to trade off regulator performance with fuel consumption during the design process. In a more serious large-scale design effort, such as that for a General Electric LM-2500 aboard a variable pitch propeller ship, the more mathematical approach offered by the LQR method may be the only viable approach. In that case, there are simply too many variables and inputs to be effectively managed by the classical design methodology.

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References

Kalyn, R. A., 1979, "An Application of Optimal Control Theory to the FFG-7 Gas Turbine Propulsion System," MSME Thesis, Naval Postgraduate School, Monterey, CA, June.

Michael, G. J., and Farrarr, F. A., 1975, "An Analytical Method for the Synthesis of Nonlinear Multivariable Feedback Control," United Aircraft Research Laboratories Report M941338-2, June.

Ogata, K., 1970, *Modern Control Engineering*, Prentice-Hall, Inc., Englewood Cliffs, NJ.

Rubis, C. J., and Harper, T. R., 1982, "The Naval Gas Turbine Ship Propulsion Dynamics and Control Systems Research and Development Program," *SNAME Transactions*, Vol. 90, pp. 321–338.

Rubis, C. J., and Harper, T. R., 1986, "Governing Ship Propulsion Gas Turbine Engines," *SNAME Transactions*, Vol. 94, pp. 283–308.

Smith, D. L., 1988, "Linear Modeling of a Marine Gas Turbine Power Plant," ASME Paper No. 88-GT-71.

Stammetti, V. A., 1988, "Comparative Controller Design for a Marine Gas Turbine Propulsion Plant," MSME Thesis and report NPS 69-88-010, Naval Postgraduate School, Monterey, CA, Sept.