ADAPTIVE REGULATION OF THE BOILER-TURBINE UNIT OF A COMBINED HEAT AND POWER (CHP) SYSTEM

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Abstract:

A simplified nonlinear model of a CHP system is used to investigate the steady state errors, hunting phenomena and high interactions produced in the linear regulators currently employed in industry. A gain-scheduling regulator is designed to overcome these shortcomings.

LIST OF SYMBOLS

A = free surface area of water in drum

D = rotor damping factor

m_e = steam demand

p = drum steam pressure

P_D = mechanical power demand

Q_D = thermal power demand

T_p = pass-out pressure time constant

T_s = fall in mass of water in boiler,

per unit increase in evaporation rate

T_{v_1} = inlet-valve time constant

T_{v_2} = pass-in valve time constant

u_1 = fuel flow

u_2 = inlet valve input

u_3 = feedwater flow

u_4 = pass-in valve input

V_f = specific volume of saturated water

x_1 = inlet valve position

x_2 = pass-in valve position

x_3 = pass-out steam pressure

x_4 = rotor speed

y = drum water level

1:Introduction

Steady state errors, hunting and high interactions have been reported in CHP systems incorporating steam turbines. The regulators used in practice or designed in the literature are usually based on a linearised model of the system using constant gains whatever the operating conditions. To tackle these problems, a more accurate nonlinear mathematical model has been derived in [1], which will be used here to demonstrate the reasons for the phenomena and develop adaptive regulator for the boiler-turbine unit of a typical CHP system.

The feedwater flow, fuel flow, inlet-valve and pass-in valve positions are the inputs which can be used to regulate the rotor speed, mechanical output power, pass-out steam pressure, thermal output power, drum steam pressure and drum water level as outputs of the system. Although the system is multi-input multi-output, it is in practice regulated through separate single-input, single-output loops. Here, the attempt is not to change the concept of separate loops, but to design improved controllers which can adapt with system changes and cope with the system nonlinearities.

In section 2, two PI regulators are designed for the boiler unit to make the turbine input changes possible. In section 3.1 using the open-loop turbine model, the steady state error and hunting phenomena are demonstrated.

To design a pole-shifting regulator, a global linear model is derived in section 3.2 using the special system arrangements. In section 3.3, two pole-shifting regulators are designed for the turbine. As steady state errors and high interactions are observed, the reasons for the steady state errors and hunting and high interactions are discussed in section 3.4. In section 3.5, these shortcomings are overcome by designing a gain-scheduling regulator with adaptive precompensator gains. In section 3.6 some practical issues are outlined and section 4 concludes the paper.

Fig 1 and 2 illustrate the regulator arrangement for the boiler and turbine respectively, where C1, C2, C3 and C4 are the regulators.

2:Boiler Regulation

The model derived in [1] to describe the gross behaviour of a boiler in a CHP system is given by:

$$\frac{dp}{dt} = -\alpha_1 u_2 p + \alpha_2 u_1 - \alpha_3 u_3 \qquad (1)$$

$$\frac{dy}{dt} = \frac{V_f}{A} . (u_3 - m_e + T_s . \frac{dm_e}{dt}) \qquad (2)$$

where $m_e = G . u_2 . p$ and α_i are constants.

A PI regulator has been designed to regulate the water level by governing the feedwater flow, while changing the water flow reference value according to the steam demand. The block-diagram of such a regulator is shown in fig. 3. The feedwater valve is considered to be very rapid in operation so that its time constant is negligible. Hence:

$$u_3 = m_e + K_1 . (y_{ref} - y) + K_2 . \int(y_{ref} - y) \qquad (3)$$

A PI regulator has also been designed to regulate the drum steam pressure by governing the fuel flow . The block-diagram of such a regulator is shown in fig. 4. The fuel valve is considered to be very rapid in operation so that its time constant is negligible. Hence:

$$u_1 = u_{1_o} + K_3 . (p_{ref} - p_t) + K_4 . \int(p_{ref} - p_t) \qquad (4)$$

The designed water level regulator and drum steam pressure regulator were applied to the simulated boiler model. Simulation results on the drum boiler model showed that fairly perfect regulation can be achieved, if a high gain proportinal feedback is used. However because this high gain regulator would require sharp changes in the fuel flow and this is not practically feasible, a lower feedback gain was used for the drum steam pressure regulation.

Fig. 5 shows the response of the boiler to a step change in the inlet valve position. The drum pressure response converges to the original value but in a relatively long period of time. Fig. 6 shows the response of the boiler to a step change to the drum steam pressure reference value.

3: Back-pressure Turbine Regulation

The equations describing the back-pressure turbine, derived in [1],are given by:

$$T_{v1}\dot{x}_1 = u_1 - x_1 \tag{5}$$

$$T_{v2}\dot{x}_2 = u_2 - x_2 \tag{6}$$

$$T_p\, \dot{x}_3 = K_c\,.\,(\, m_1 - m_2 - Q_D\,) - \beta_t.x_3 \tag{7}$$

$$M\,.\,\dot{x}_4 = h_1 m_1 + h_2 m_2 - P_D - D\,.\,x_4 \tag{8}$$

$$m_1 = \beta_1.p.x_1 \tag{9}$$

$$m_2 = \beta_2.x_2.x_3 \tag{10}$$

3.1: Regulation Problems in CHP Systems

Simulation of the open-loop model is first used to demonstrate the regulation problems associated with the turbine unit of CHP systems.

Fig. 7 shows the system response following a -0.1 step change to the rotor speed reference value. It shows steady state errors for both the pass-out steam pressure and rotor speed.

Now suppose a PI regulator of the pass-in valve position is designed to overcome the problem of the pass-out steam pressure steady state error. Fig. 8 shows the system response following the same step change to the rotor speed with the PI regulator on the pass-out steam pressure. The simulation result shows no steady state error for the pass-out steam pressure and the rotor speed.

Fig. 9 shows the open-loop system response to a 5% step change to the pass-out steam pressure reference value. Similar to fig. 7, both the rotor speed and the pass-out steam pressure performance show some steady state errors. The PI regulator designed for the pass-out steam pressure is applied in the simulation of fig. 10. It shows that although the pass-out steam pressure steady state error has been diminished, the rotor speed steady state error still remains.

To overcome the problem of the steady state error of the rotor speed, suppose a PI regulator is also designed for the rotor speed to be regulated by the inlet valve. Simulation results of fig. 11

show that even with a small proportional feedback, the result is *hunting*. (the reason is discussed in section 3.4.2).

In practice, the regulators for such a system are designed based on the linearized model of the turbine around a normal operating point. The rotor speed and mechanical output power are regulated by a constant gain P or PI regulator through the inlet valve, and the pass-out steam pressure is regulated by a constant gain PI regulator through the pass-in valve. To avoid the steady state errors and hunting phenomena, some precompenator gains are used [1]. The regulator arrangement used in practice in shown in fig. 12.

Although by using the precompensator gains, which allow each valve to follow the other valve variations, the steady state errors and hunting problems are partially cured, the regulator designer is bound to compromise between some steady state error and high interactions. This is because a stronger integrator results in less steady state error but higher interactions and vice versa.

The above simulation results showed that the system is very interactive. Two different approaches can be used here to tackle this problem. One approach is to regulate the system with a multivariable regulator. The other approach, which is adopted here, is simply to analyse the system responses and determine the reasons for the steady state errors and hunting and try to overcome these problems by some other means (namely by using adaptive regulators).

3.2: A Global Linear Model For The System

The turbine model, described by equations 5,6,7 and 8 is a multivariable nonlinear model. To design a linear regulator, this should be approximated by a linear model. A linear model of the system can be obtained by expanding the nonlinear terms in a Taylor series around an operating condition and neglecting the second and higher terms of the series. A regulator design based on this linear model would then involve the replacement of Δp by a combination of the feedwater flow, the fuel flow and the inlet valve position deviations using the boiler model. This would result in a complicated and highly interactive multivariable system model, requiring a complicated regulator.

In this paper, the aim is to design two SISO regulators similar to the currently employed regulators described earlier, such that it is as easily implementable and also can cope with the previously mentioned shortcomings by means of adaptive gains. To do this, instead of expanding the nonlinear terms in a Taylor series to obtain a linear model, the special system arrangement (namely large time constants of the inlet and pass-out pressure variations)is exploited to give the system model approximated by:

$$T_{v1}\dot{m}_1 = um_1 - m_1 \tag{11}$$

$$T_{v2}\dot{m}_2 = um_2 - m_2 \tag{12}$$

$$T_p\, \dot{x}_3 = K_c\, .\, (\, m_1 - m_2 - Q_D\,) - \beta_t . x_3 \tag{13}$$

$$M\, .\, \dot{x}_4 = h_1 m_1 + h_2 m_2 - P_D - D\, .\, x_4 \tag{14}$$

where:

$$um_1 = \beta_1 p u_1 \tag{15}$$

$$um_2 = \beta_2 x_3 u_2 \tag{16}$$

This new model is linear in the states and improved compared with the previous linear model since the outputs of the model (i.e. pass-out steam-pressure (x_3) and rotor speed (x_4)) are those of the original model rather than their deviations and are not based on a particular operating condition, and therefore valid for all conditions. To explain the difference between this and the original model, the variable m_1 is first considered. As the regulator will be implemented in a digital form, the discretised version of both models for m_1 are compared.

Using the original model and with some manipulations of equations:

$$m_1(t) = b_1\left(\frac{p(t)}{p(t-1)}\right)um_1(t-1) - a_1\left(\frac{p(t)}{p(t-1)}\right)m_1(t-1) \tag{17}$$

Now using the new model:

$$m_1(t) \;= b_1 um_1(t-1) - a_1 m_1(t-1) \tag{18}$$

Because the time constant of the drum steam pressure variations is very large, (>300 seconds) and the sampling time is chosen to be small (0.01 second), the assumption of $\frac{p(t)}{p(t-1)}=1$ for the model used in the regulator is quite reasonable. By this assumption the two models are equivalent in digital form. Since the time constant of the pass-out steam pressure variations is also relatively large (6.0 seconds), the same assumption can be considered to be valid for the mass-flow to the low-pressure turbine for the model used in the regulator.

The new linear model can be written in Laplace form as:

$$\begin{bmatrix} x_4 \\ x_3 \end{bmatrix} = \begin{bmatrix} G_{11}(s) & G_{12}(s) \\ G_{21}(s) & G_{22}(s) \end{bmatrix} \cdot \begin{bmatrix} um_1 \\ um_2 \end{bmatrix} - \begin{bmatrix} P_D \\ Q_D \end{bmatrix} \tag{19}$$

where:

$$G_{11}(s) = \frac{1}{1 + T_{v_1}s} \cdot \frac{h_1}{D + M \cdot s}$$

$$G_{12}(s) = \frac{1}{1 + T_{v_2}s} \cdot \frac{h_2}{D + M \cdot s}$$

$$G_{21}(s) = \frac{1}{1 + T_{v_1}s} \cdot \frac{K_c}{\beta_t + T_p \cdot s}$$

$$G_{22}(s) = -\frac{1}{1 + T_{v_2}s} \cdot \frac{K_c}{\beta_t + T_p \cdot s}$$

The block-diagram of this model is shown in fig. 13.

3.4: A Digital Pole-shifting Regulator

The first step in designing a digital adaptive regulator was to design a constant gain digital regulator to investigate its behaviour on the system model and to make a comparison of this regulator with the currently employed continuous regulators. Neglecting the effect of the inlet steam pressure and pass-out steam pressure variations on the mass flows (as done in currently employed regulators), the equations 15 and 16 are replaced by:

$$um_1(t) = \beta_1 p_o u_1(t) \tag{20}$$

$$um_2(t) = \beta_2 x_{3_o} u_2(t) \tag{21}$$

The reference values of u_{1_o} and u_{2_o} are related to the reference values of P_D and Q_D by :

$$u_1 = K_{11} \,.\, U_1 + K_{12} \,.\, U_2$$

$$u_2 = K_{12} \,.\, U_1 + K_{22} \,.\, U_2$$

where:

$$U_1 = P_D + D \,.\, x_{4_o}$$

$$U_2 = Q_D + \frac{\beta_t \,.\, x_{3_o}}{K_c}$$

and:

$$K_{11} = \frac{1}{\beta_1 p_o} \qquad K_{12} = \frac{h_2}{\beta_1 p_o}$$

$$K_{21} = \frac{1}{\beta_2 x_{3_o}} \qquad K_{22} = \frac{-h_1}{\beta_2 x_{3_o}}$$

Although the new model is linearised, it is highly interactive and is not suitable for regulation as two SISO systems. Some simulation results (not shown for brevity), showed that any attempt to regulate such a system with two SISO regulators would result in very similar steady state errors or hunting described in section 3.1. The first step to reduce the interactions and avoid steady state error and hunting, was to choose U_1 and U_2 instead of u_1 and u_2 as the input signals to be regulated. This means choosing K_{ij}s as precompensators to reduce the interactions.

The new linear model is:

$$\begin{bmatrix} x_4 \\ x_3 \end{bmatrix} = \begin{bmatrix} G_{11}(s) & G_{11}(s) \\ G_{21}(s) & G_{22}(s) \end{bmatrix} \cdot \begin{bmatrix} U_1 \\ U_2 \end{bmatrix} \tag{22}$$

where:

$$G_{11}(s) = \left(\frac{h_1}{1 + T_{v_1} s} + \frac{h_2}{1 + T_{v_2} s} \right) \cdot \frac{1}{D + M \,.\, s}$$

$$G_{12}(s) = \left(\frac{h_1 \,.\, h_2}{1 + T_{v_1} s} - \frac{h_1 \,.\, h_2}{1 + T_{v_2} s} \right)$$

$$G_{21}(s) = \left(\frac{1}{1 + T_{v_1} s} - \frac{1}{1 + T_{v_2} s} \right)$$

$$G_{22}(s) = \left(\frac{h_2}{1 + T_{v_1} s} + \frac{h_1}{1 + T_{v_2} s} \right) . \frac{K_c}{\beta_t + T_p \, . \, s}$$

The corresponding block-diagram is shown in fig. 14.

The sampling time was chosen to be 10 ms which is adequate for the back-pressure turbine with a smallest time constant of 240 ms. The digital pole-shifting control strategy, described in appendix A, is used to shift the poles of the system towards the origin in the Z-plane.

Fig. 15 shows the response of the back-pressure turbine to a 0.05 p.u. step change to the pass-out steam pressure (15a), a 0.1 p.u. step change to rotor speed (15b), a 0.1 p.u. step change to mechanical power (15c) and 0.23 p.u. step change to thermal power (15d) demand values, with digital control when the pole shifting factors are 0.995. It shows some unacceptable steady state errors for both the pass-out steam pressure and the rotor speed. To reduce the steady state errors, the pole-shifting factors were reduced. Fig. 16 shows the response of the system after the same step changes with pole shifting factors of 0.95. Although the steady state errors have been reduced, it was found that they could not be diminished completly however small the poleshifting factor. Morever, decreasing the pole shifting factors results in :

a: increased oscillations

b: increased interactions

Each of these disadvantages could affect the overall stability of a CHP system. Reducing the pole shifting factors from 0.95 does not improve the system response . Different pole shifting factors for different loops were also investigated and it was found that improving regulation of the first loop results in worsening of the second loop regulation and vice versa.

3.4: Adaptive Regulator

In this section the simulations of system so far are reviewed and the reason for the steady state error and hunting and high interactions are outlined, and then possible solutions are discussed.

3.4.1: Steady State Error

Consider first the simulation of fig. 7. When a step change to the rotor speed is required, the turbine output power should be changed. Since the power loss in the turbine has been neglected in the turbine modelling, to change the turbine output power, the turbine inlet power should be changed by the same amount. The turbine input power or the boiler output power is changed by opening or closing the inlet valve.

Fig. 7 shows that the system with this step change has steady state errors for both the rotor speed and pass-out steam pressure and also the turbine output power and pass-out steam pressure. The reason is that any change in the inlet valve position would result in a change in the pass-out steam flow as well as the turbine output power; Also any change in the pass-out steam flow results in a change in the pass-out steam pressure. The simulation results show that the rotor speed and the turbine output power also have some steady state errors. The reason is that any change in the pass-out steam flow produces a change in the steam flow to the low-pressure turbine and subsequently a change in the turbine output power and the rotor speed.

The problems in the simulation of fig. 7 can be solved by applying a PI regulator to regulate the pass-out steam pressure through the pass-in valve. This regulator would change the pass-in valve position such that the pass-out steam flow would not change with the inlet valve changes. Fig. 8 shows that as the pass-out steam flow was kept constant the steady state error of the pass-out steam pressure as well as the turbine output power and rotor speed were diminished.

Fig. 9, similar to fig. 7, shows steady state errors for all the system outputs for a step change to the pass-out steam pressure. The reason is that any change in pass-out steam pressure should be through a change in the pass-out steam flow. Any change in the pass-out steam flow results in a change in the steam flow to the low-pressure turbine and subsequently to the turbine output power and the rotor speed. Fig. 10 shows that although the PI regulator designed to overcome the pass-out steam pressure steady state error cures the problem, the steady state error of the rotor speed and turbine output power remains almost the same. This is because the increase in the steam flow to the low pressure turbine caused by the decrease in the pass-out steam flow should be compensated by a decrease in the inlet steam flow. But the result of doing so is ***hunting***.

3.4.2: The Hunting Phenomena

The hunting phenomena is the result of high interactions between the two loops of the regulation in a CHP system. When in the simulation of fig. 11 the steady state error of the rotor speed is intended to be compensated by closing the inlet valve and subsequently decreasing the steam flow to the high pressure turbine, it also decreases the pass-out steam flow. The pass-out steam pressure regulator closes the pass-in valve position to increase the pass-out steam flow. This results in decreased steam flow to the low pressure turbine and subsequently decreased output power and rotor speed. The rotor speed regulator this time opens the inlet valve to increase the steam flow to the high pressure turbine and the opposite procedures take place and result in increased rotor speed. The result of repeating all these interactive regulations is hunting.

3.4.3: High Interactions

All the simulations and the reasons discuused for the steady state error or hunting phenomena, showed that any change to either of the valve positions should be accompanied by a proper change to the other. This arrangement has been considered in the currently employed regulators and also in the designed digital regulator of section 3.3 by means of precompensator gains. Although in section 3.3, the best possible constant gain precompensators were considered, the result is far from perfect regulation. The reason is that because these gains depend on some steady state values, they are only suitable for the operating conditions for which these precompensators are designed. When the operating conditions are changed, the gains are only partially suitable. And the result is "partial hunting" or "high interactions".

To overcome the problems of the steady state errors and hunting and also high interactions, the only solution is to design some precompensator gains which can adapt with system changes.

3.5: Gain-scheduling Regulator

To convert the constant gain regulators to gain-scheduling regulators, the pass-out and inlet steam pressure variations were used as auxilary variables. That is, the regulator parameters were calculated by changing $K_{ii}s$ adaptively. i.e. using $K_{ii}(t)$ instead of $K_{ii}s$ defined earlier, where:

$$K_{11}(t) = \frac{1}{\beta_1 p(t)}$$

$$K_{12}(t) = \frac{h_2}{\beta_1 p(t)}$$

$$K_{21}(t) = \frac{1}{\beta_2 x_3(t)}$$

$$K_{22}(t) = \frac{-h_1}{\beta_2 x_3(t)}$$

These adaptive gains are in fact the nonlinear transformations which transform the nonlinear system to a linear time-invariant system. This form of gain-scheduling regulation has been described to be the best and saves considerable computing time as the regulator parameters can be calculated off-line and would overcome the previous shortcomings, since the precompensator gains do not depend on some steady state values and change with system operating conditions.

Fig. 17 shows the response of the system with such a gain-scheduling regulator. Although the pole-shifting factor has been selected to be close to unity (0.995) to avoid high interactions, no steady state errors are observed. This gain-scheduling regulator results in diminishing completely the steady state errors while avoiding the disadvantages of the improved constant gain regulators.

As the model of equation 22 suggests, if $K_{ii}(t)s$ are used and $T_{v_1} = T_{v_2}$, then the interactions should disappear.This is physically explainable . If, for example, a step change to the mechanical power demand value is applied, it changes u_1 by $K_{11}\delta P_D$. It also changes u_2 by $K_{12}\delta P_D$ to reverse the effect of new steam flow in the high-pressure turbine on the pass-out steam pressure and thermal output power. To do this perfectly, it should be carried out simultaneously, which is possible if $T_{v_1} = T_{v_2}$. This is also true when u_2 is changed to match the new thermal power or pass-out steam pressure demand and u_1 should change to keep the mechanical output power constant.

Equalising T_{v_1} and T_{v_2} can be done by choosing two similar servomotors for the valves. Fig. 18 shows the response of the turbine to the same step changes as before when the gains $K_{ii}(t)s$ are used and $T_{v_1} = T_{v_2}$. This regulator shows a very good result with very small interactions and oscillations. The fact that the interactions are not eliminated completely is because of the difference between the model employed in the regulator design and the original nonlinear model used in simulation. Choosing two fast valves would significantly reduce system interactions.

3.6: Unmodelled Parameters

In the regulators currently employed in industry, the regulation of the turbine is based on the assumption that steam flow and valve position are proportional. The simulation results showed that this would result in high interactions and inadequate regulation. In this study, however, steam flow was modelled as dependent on the product of valve position and steam pressure. Although this assumption is much closer to reality, it is not true for all operating conditions.

The simulation results with the new model showed that the best regulation is only achieved if the precompensator gains were adaptively changed according to new steam pressure values. However, in practice, where the relation between mass flow and valve position is not affected only by steam pressure (specially in low load conditions), it may still result in high interactions or even hunting.

Considering for example $K_{11}(t)$:

$$K_{11}(t) = \frac{1}{\beta_1 p(t)} = \frac{\text{steam flow into high pressure turbine}}{\text{inlet valve position}}$$

To overcome the problem of unmodelled parameters (e.g. temperature, specific volume), instead of sampling $p(t)$ and changing $K_{11}(t)$ accordingly, the steam flow to the high pressure turbine and the inlet valve position can be sampled and their ratio at time t can be considered as $K_{11}(t)$. Calculating all the precompensator gains by this policy would take into account any other parameters which in practice may affect the system performance.

4: Conclusions

In this paper, the boiler turbine unit model of a typical CHP system has been adopted for regulation. Having designed two PI regulators for the water level and drum steam pressure, the back-pressure turbine model was used to investigate the weaknesses of the constant gain regulators employed in practice. The main disadvantage of such regulators was found to be the high interactions or steady state errors resulting because they are based on a linearised model around a particular operating point.

To avoid these shortcomings, a linear in the states model was derived which was not based on a Taylor series expansion, but based on the assumptions of negligible steam pressure variations during one sampling time. First, a constant gain digital regulator was designed and very similar problems to those of the currently employed regulators were observed. Then the same regulator was applied but the precompensator gains were changed adaptively. The simulation results showed much improvement in system regulation. Later, the time constants of the valves were made to be equal by use of a local feedback on the servomotor of the first valve. This change along with adaptively changing regulator gains improved the regulation considerably.

In summary this research has suggested that to improve the regulation of the boiler-turbine unit of a CHP system:

a: the gains of regulators should be changed adaptively according to steam flow and valve position ratios, and

b: identical and fast acting servomotors should be chosen for the inlet and pass-out valves.

APPENDIX A

A SISO discrete model of a system, neglecting noise, can be written as:

$$y_t = \frac{B(z^{-1})}{A(z^{-1})} . u_t \tag{A.1}$$

where y_t is the process output at time t, u_t is process input at time t, and:

$$A(z^{-1}) = 1 + a_1 z^{-1} + ... + a_{n_a} z^{-n_a} \tag{A.2}$$

$$B(z^{-1}) = 1 + b_1 z^{-1} + ... + b_{n_b} z^{-n_b} \tag{A.3}$$

Now assume that a regulator is to be found such that the closed-loop performance is given by:

$$y_t = \frac{B(z^{-1})}{A_m(z^{-1})} . u_t \tag{A.4}$$

where:

$$A_m(z^{-1}) = 1 + a_{m_1} z^{-1} + ... + a_{m_{n_a}} z^{-n_a} = 0 \tag{A.5}$$

is the desired closed-loop characteristic equation. A general control law for such a regulator is given by:

$$R(z^{-1}) u_t = T(z^{-1}) u_c - S(z^{-1}) y_t \tag{A.6}$$

where u_c is the input reference value and:

$$S(z^{-1}) = 1 + s_1 z^{-1} + ... + s_{n-1} z^{-n_s} \tag{A.7}$$

$$R(z^{-1}) = 1 + r_1 z^{-1} + ... + r_n z^{-n_r} \tag{A.8}$$

$$T(z^{-1}) = 1 + t_1 z^{-1} + ... + t_n z^{-n_t} \tag{A.9}$$

With this regulator , the closed loop transfer function is given by:

$$.y_t = \frac{B(z^{-1}).\, T(z^{-1})}{(R(z^{-1}) \,.\, A(z^{-1}) + B(z^{-1}) \,.\, S(z^{-1}))} \,.\, u_c \tag{A.10}$$

The regulator parameters can be calculated by comparing the equations A.10 and A.4. One special form of pole-placement control design is the pole-shifting strategy. Although pole-shifting was originally used to design feedforward open-loop compensators, it is also used for the design of closed-loop regulators in which all the poles of the system are shifted towards the origin by a single factor. Although it might not be the optimum form of pole-placement, it is certainly much easier to implement as only one factor (i.e. the pole-shifting factor) needs to be determined according to the input limits, whatever the system order. In pole-shifting regulation the closed-loop characteristic equation is given by:

$$A_m(z^{-1}) = 1 + a_1.(\alpha - 1).z^{-1} + a_2 \, . \, (\alpha^2 - 1)z^{-2} \cdots = 0 \qquad \text{(A.11)}$$

where α is the pole-shifting factor. Then the regulator parameters are given by:

$$\begin{bmatrix} 1 & 0 & . & 0 & b_1 & 0 & . & 0 \\ a_1 & 1 & . & 0 & . & b_1 & . & 0 \\ . & a_1 & . & 0 & . & . & . & 0 \\ . & . & . & 1 & b_{n_b} & . & . & 0 \\ a_{n_a} & . & . & a_1 & 0 & b_{n_b} & . & b_1 \\ 0 & a_{n_a} & . & . & 0 & 0 & . & . \\ 0 & 0 & . & . & 0 & 0 & . & . \\ 0 & 0 & . & a_{n_a} & 0 & 0 & . & b_{n_b} \end{bmatrix} . \begin{bmatrix} r_1 \\ r_2 \\ . \\ r_{n_r} \\ s_1 \\ s_2 \\ . \\ s_{n_s} \end{bmatrix} = \begin{bmatrix} a_1(\alpha-1) \\ a_2(\alpha^2-1) \\ . \\ . \\ a_{n_a}(\alpha^{n_a}-1) \\ 0 \\ 0 \\ 0 \end{bmatrix} \qquad \text{(A.12)}$$

REFRENCES

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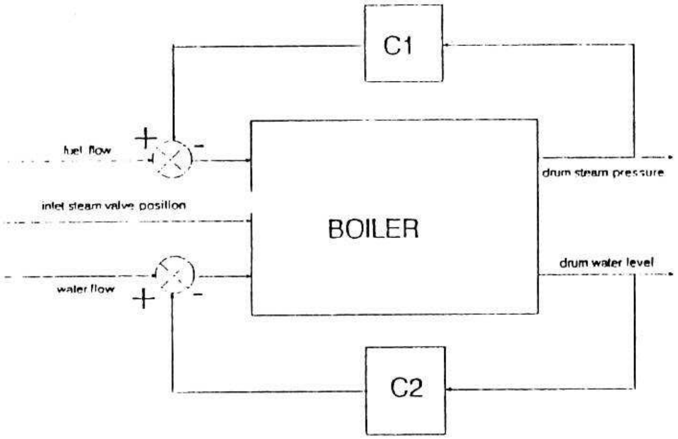


fig 1: The arrangement of regulators in the boiler

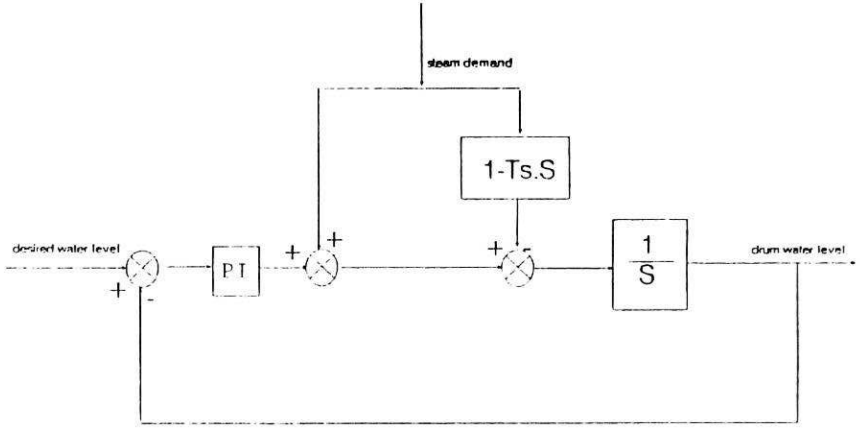


fig. 3 : The block-diagram of the water level regulator

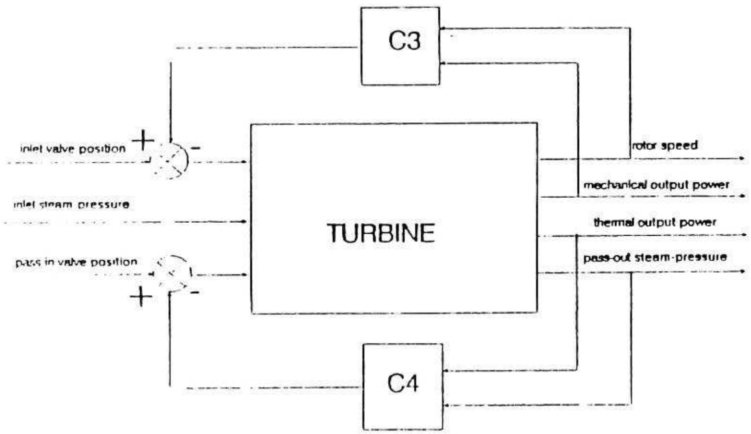


fig. 2: The arrangement of regulators in the turbine

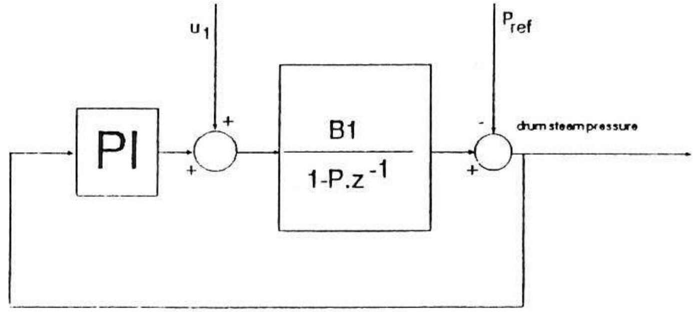


fig. 4: The block-diagram of the drum steam pressure regulator

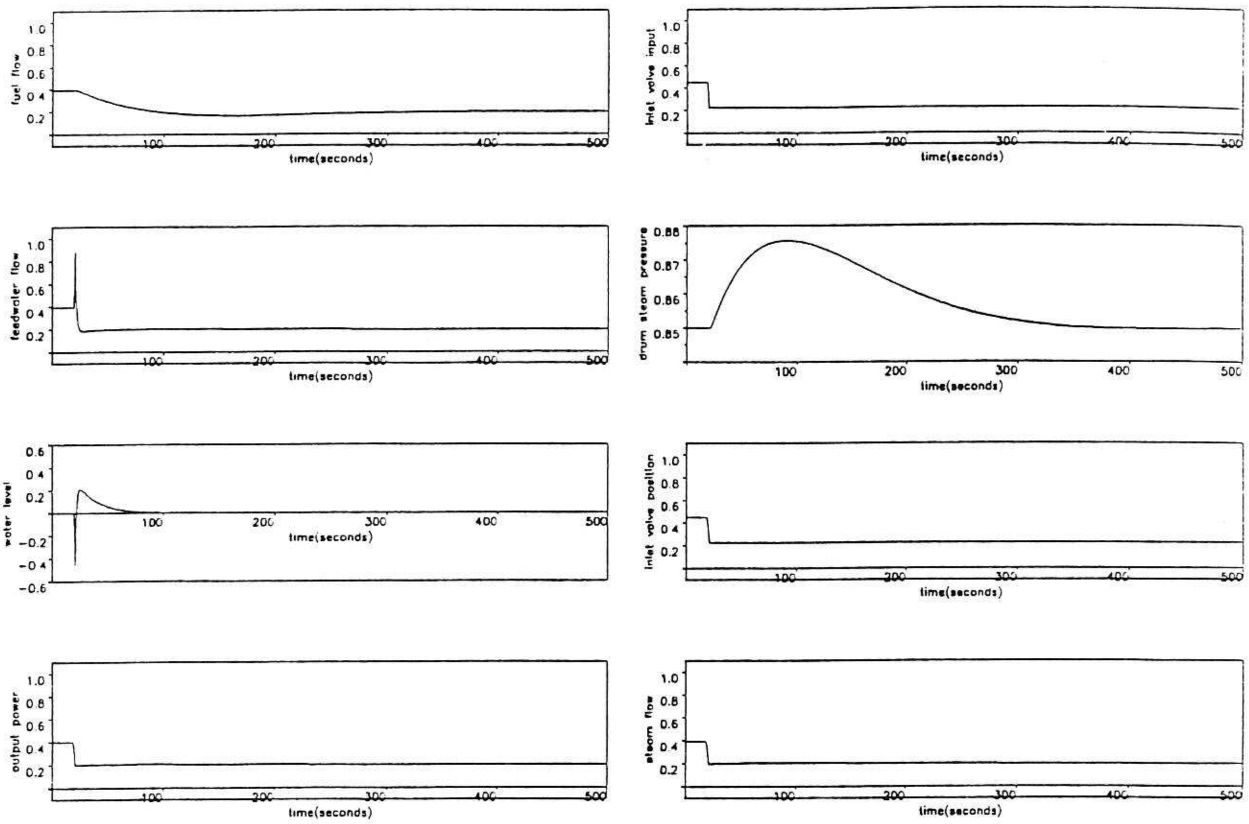


fig. 5: The boiler response to output power demand change with two PI regulator

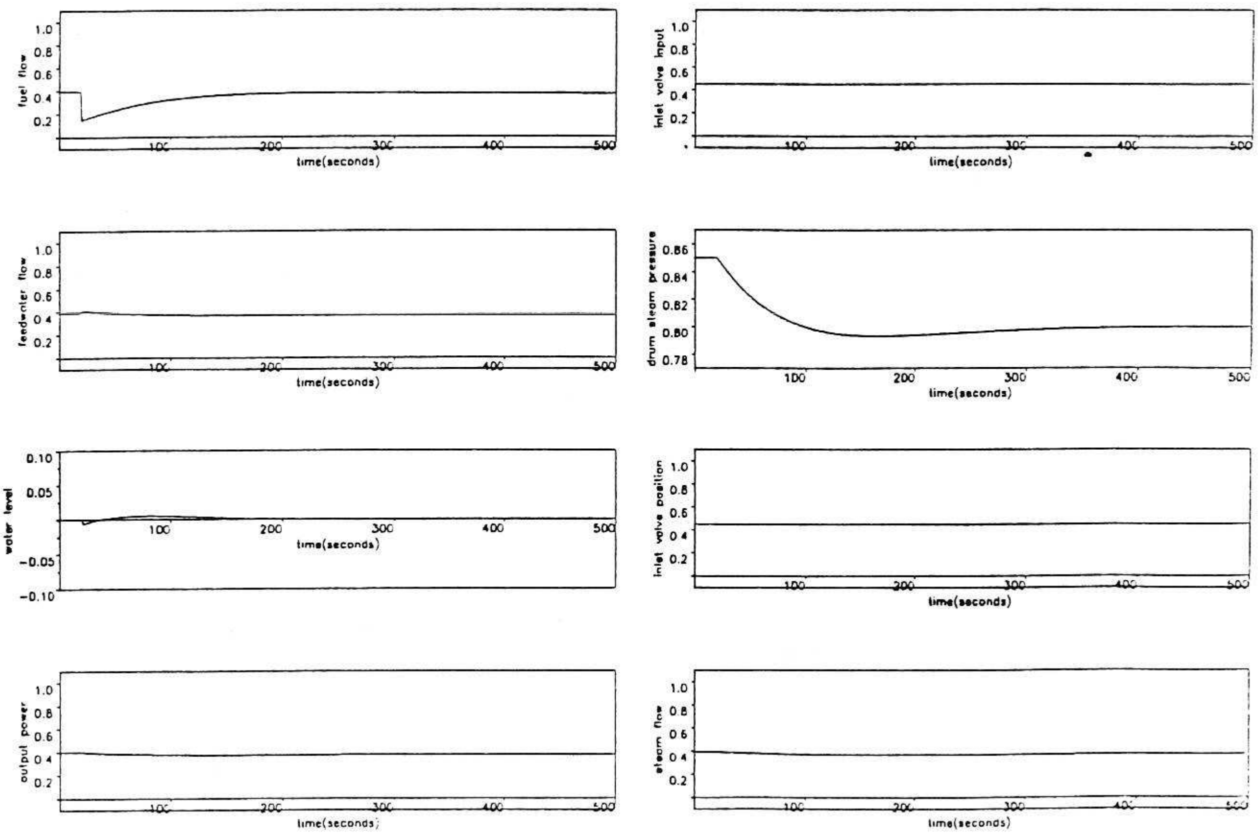


fig. 6: The boiler response to drum steam pressure demand change with two PI regulator

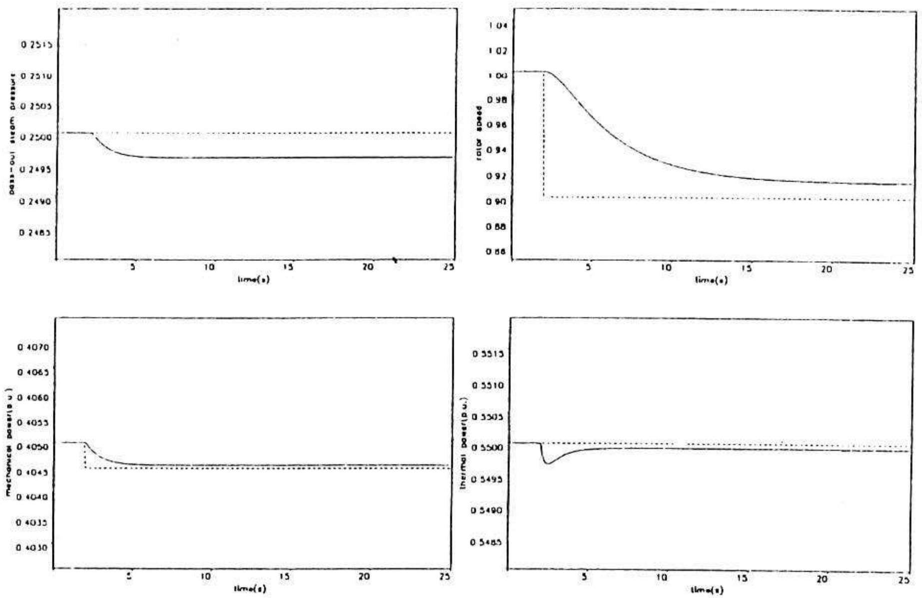


fig. 7 :the open-loop system response to -0.1 p.u. step change to rotor speed

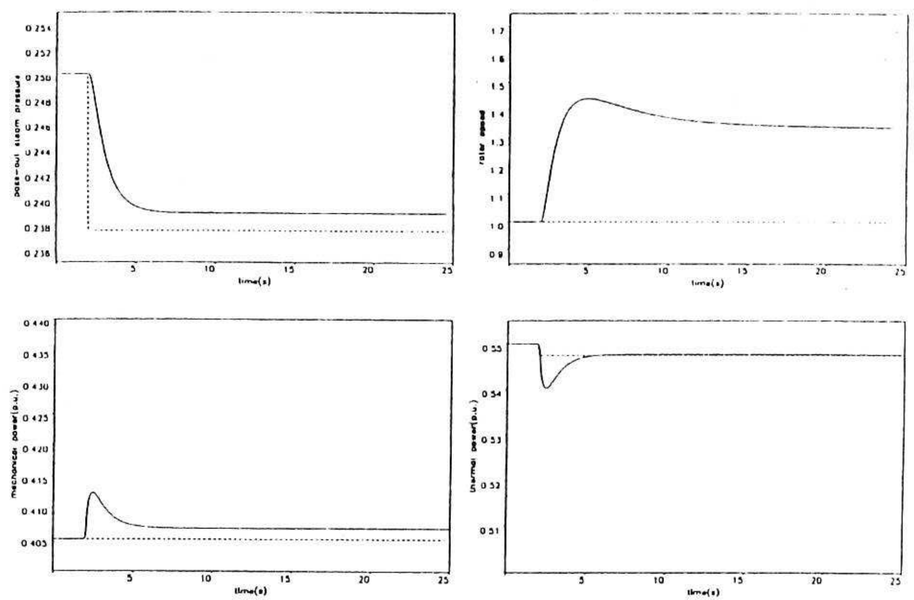


fig. 9 :the open-loop system response to 0.125 p.u. step change to pass-out steam pressure

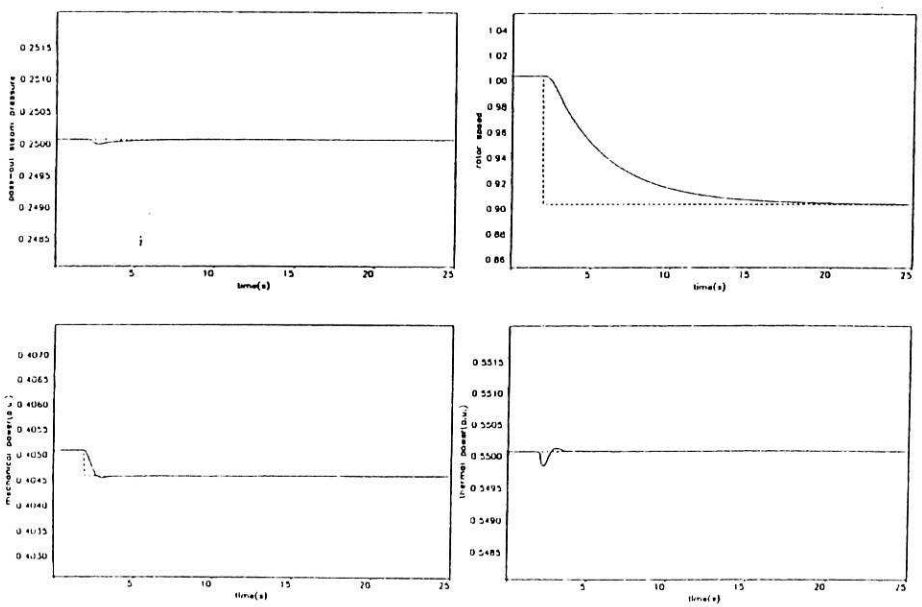


fig. 8 :the system response to -0.1 p.u. step change to rotor speed with the PI regulator

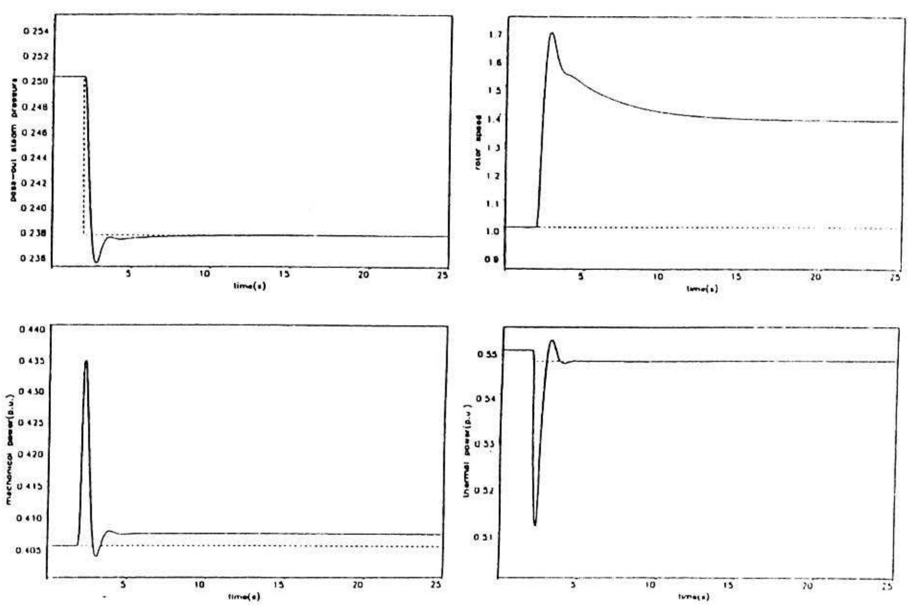


fig. 10 :the system response to 0.125 p.u. step change to pass-out steam pressure with the PI regulators

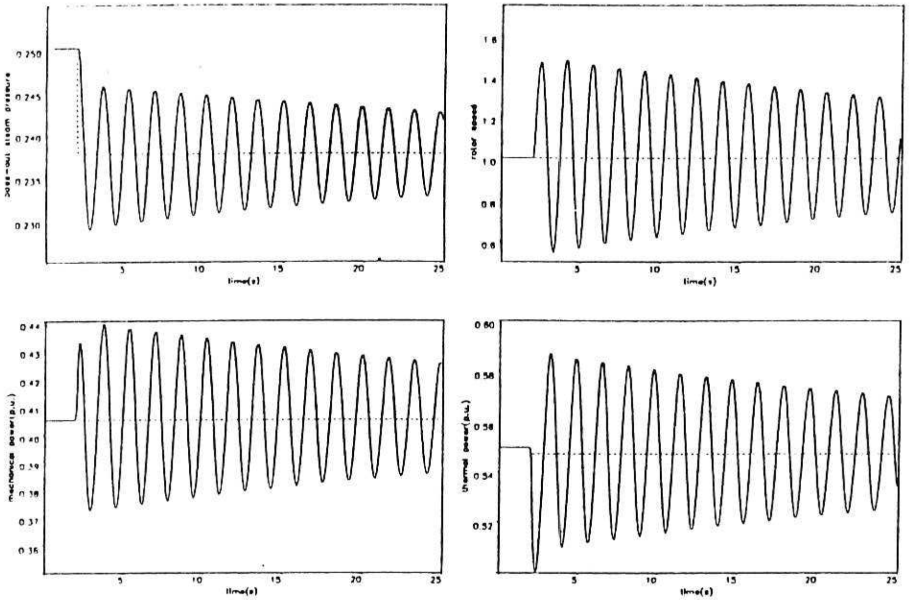


Fig. 11: the system response to 0.125 p.u. step change to pass-out steam pressure with a PI regulator for pressure and a proportional regulator for rotor speed

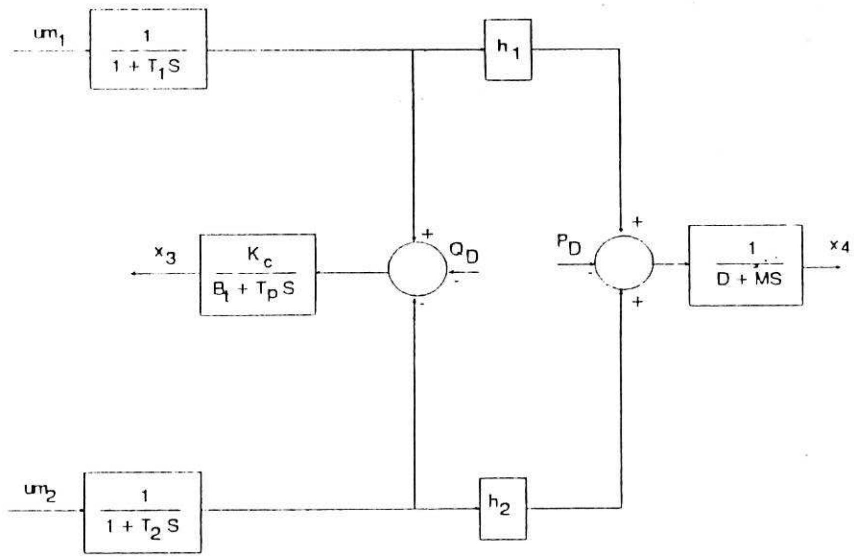


Fig. 13 : The linear model of the turbine

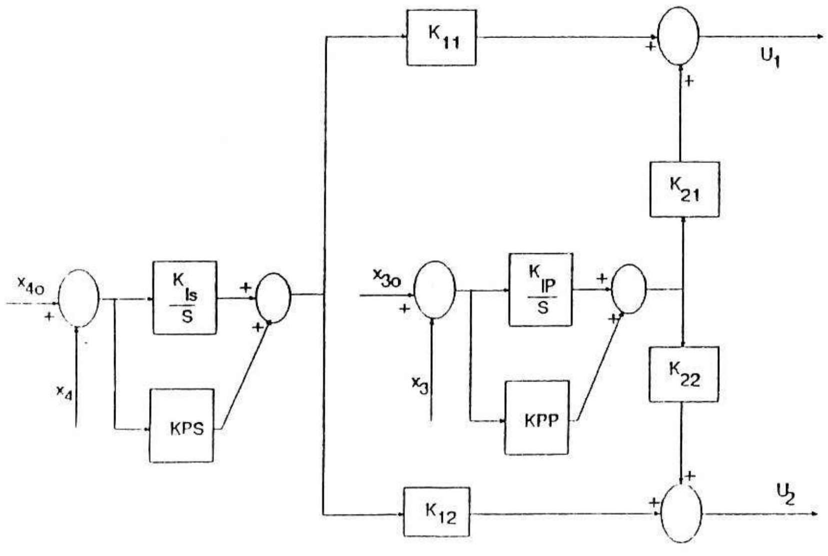


Fig. 12 : The regulator with a PI for each loop

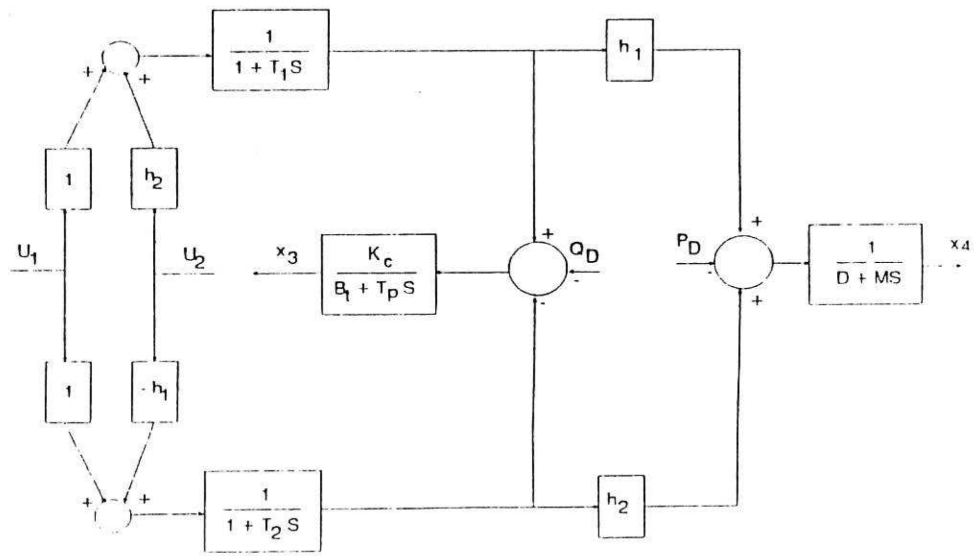
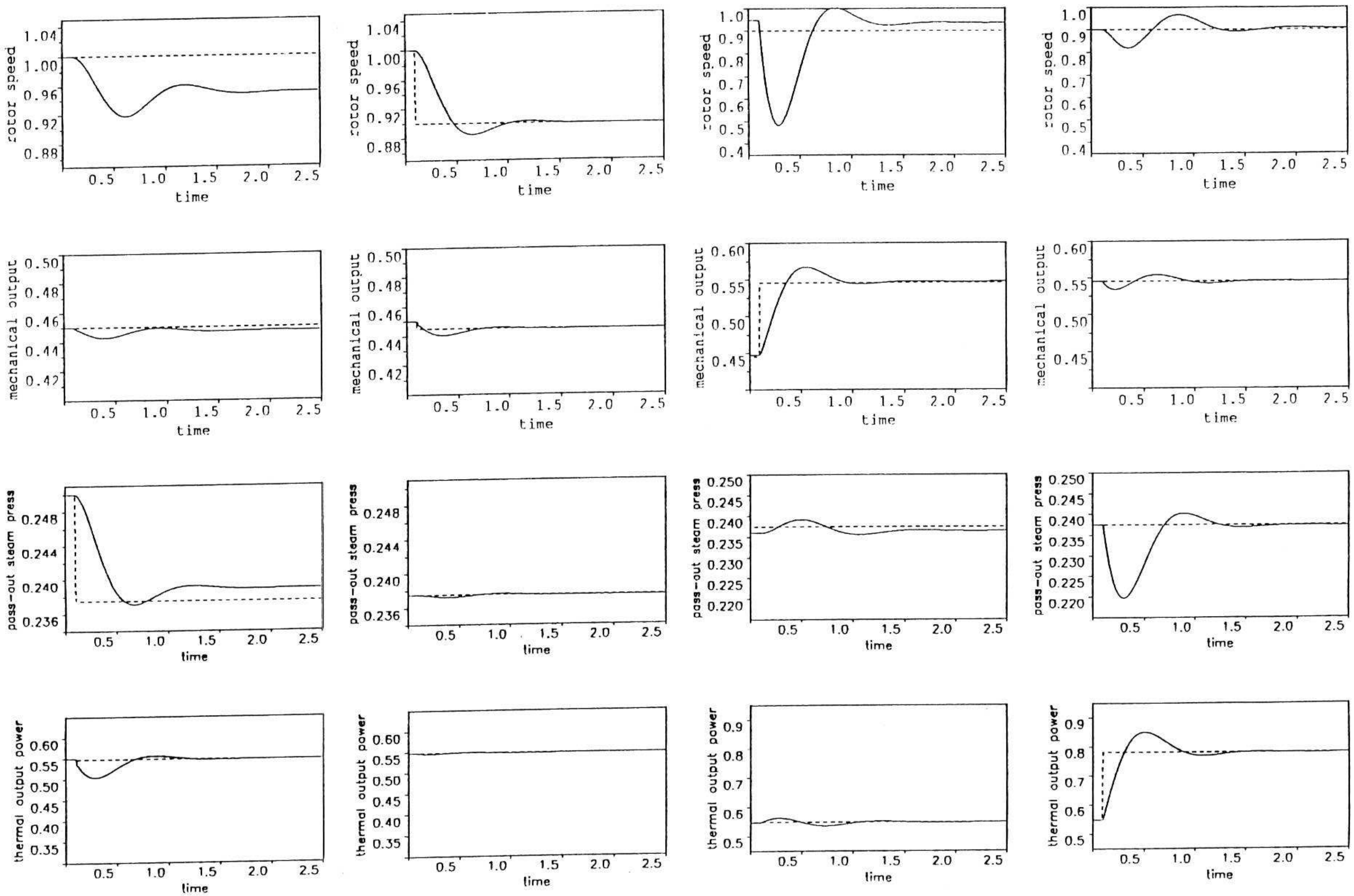


Fig. 14 : The linear model of the turbine with precompensators



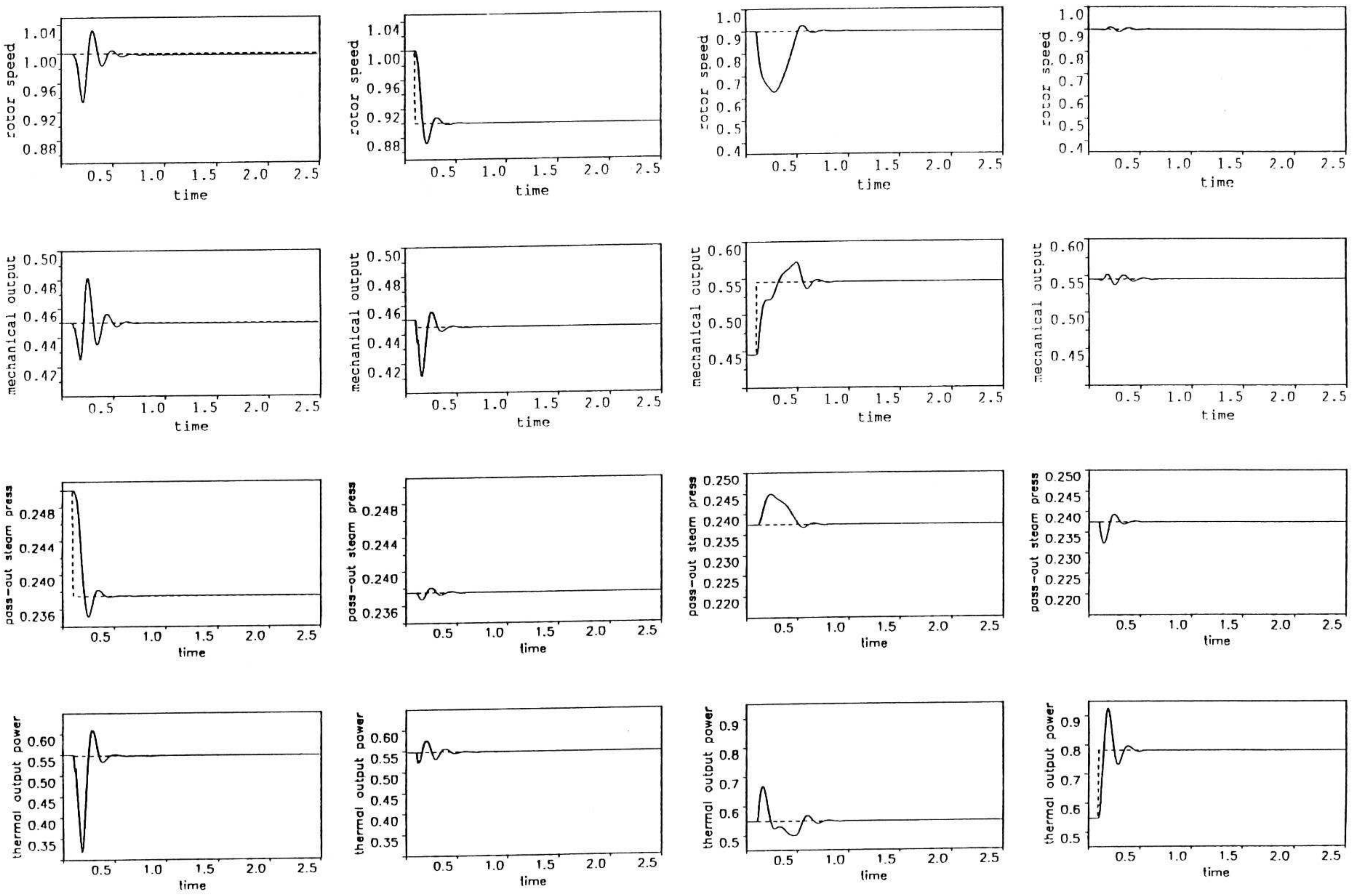
a: the pass-out steam pressure demon

b: the rotor speed demand step change

c: the mechanical power demand step c

d: the thermal power demand step chan

Fig. 15: The turbine response with the digital controller with at pole-shifting factor 0.995 to different step changes



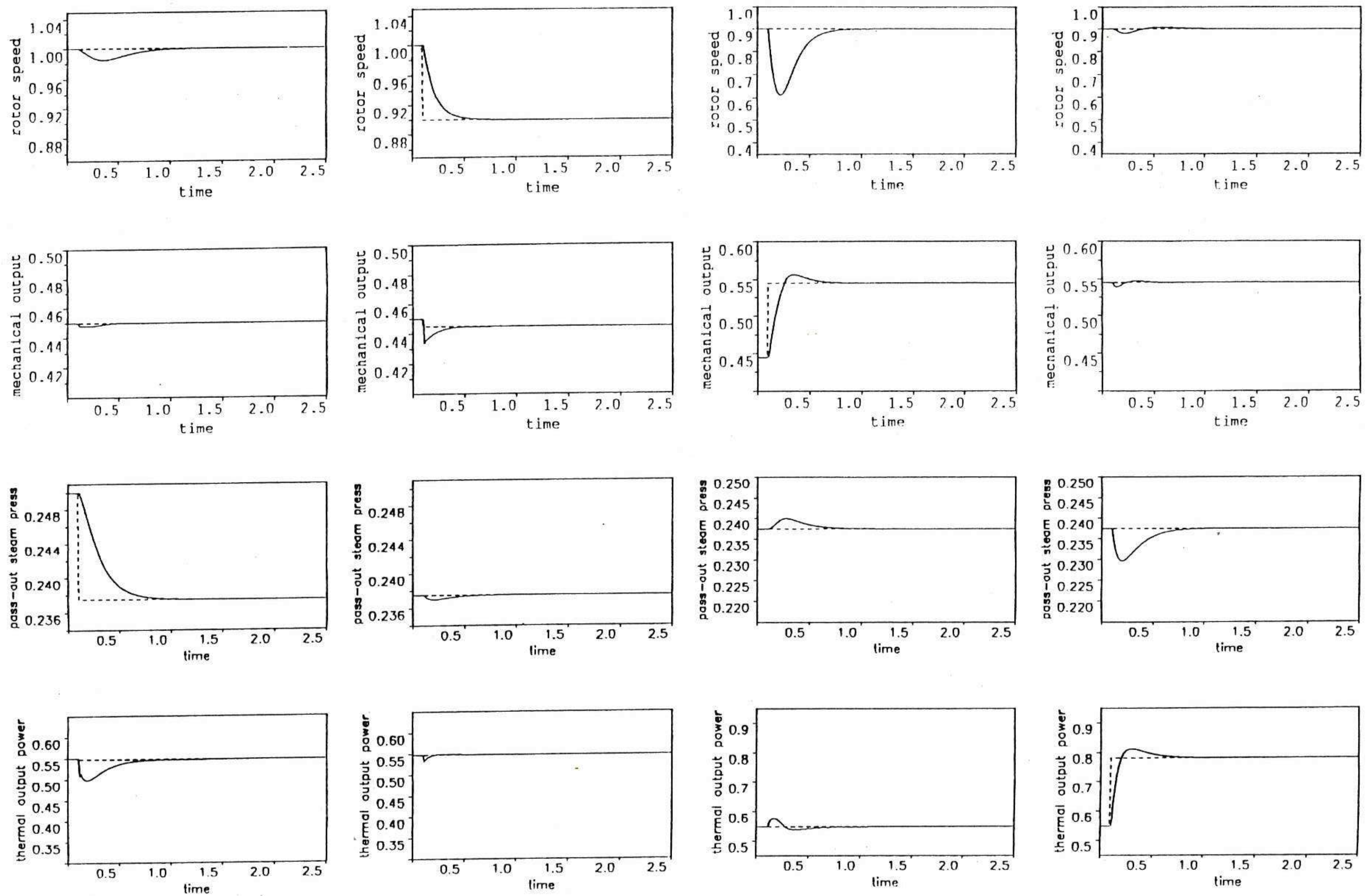
a: the pass-out steam pressure demon

b: the rotor speed demand step change

c: the mechanical power demand step c

d: the thermal power demand step chan

Fig. 16: The turbine response with the digital controller with at pole-shifting factor 0.95 to different step changes



a: the pass-out steam pressure demon

b: the rotor speed demand step change

c: the mechanical power demand step c

d: the thermal power demand step chan

Fig. 18: The turbine response with the adaptive digital controller with equal valve time constants to different step changes