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TIME DOMAIN ANALYSIS OF A MARINE BLED STEAM TURBINE WITH SIMULTANEOUSLY-INTERFERINC CONTROLLERS

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Abstract

A study is carried out to analyse the closed loop control system of a marine bled steam turbine for both turbine's speed and extracted steam pressure. Control action on steam admittance valves is achieved through simultaneous interference of the used controllers. The controlled plant parameters were chosen so as to simulate rather difficult control conditions due to significant time delays inherent to the plant and a small rotor time constant. Furthermore, the disturbances are considered to be sudden changes represented by unit step functions.

1. Introduction

Ships, and consequently their propulsion plants, are normally subjected to considerable load disturbances during their operation.

Due to ship motion among waves, the head on the propeller and the pressure distribution on to blades vary continuously. Waves also result in an increase in ship's resistance thus leading to a reduction in ship's speed. Consequently, the angle of attack of the blades is also affected. All these factors may lead, according to propeller circulation theory, to considerable disturbances represented by changes in propeller lift and drag and consequent changes in propeller thrust and torque.

Variations in propeller torque, when transferred to the steam turbine, through reduction gears, represent external load disturbance on the plant.

The steam cycle considered here is a regenerative steam cycle with single steam extraction.

The fluctuation of the load on a marine steam power plant, while affecting the steam rate, changes in turn the condensate outlet temperature from feed water heaters. Moreover, this condensate outlet temperature may be slightly affected by regional ambient temperature variations along ship's route. Both variations, in the amount of condensate and outlet temperature from feed water heaters, will urge the temperature regulation system on feed heaters to change the amount of bled steam resulting in a change in the steam pressure in the extraction room. This would represent a second source of disturbance on the steam turbine plant.

Sudden changes, e.g. in load on a power plant represent severe disturbance [1]. Although disturbances on a marine power plant are basically of stochastic nature, unit step disturbance is normally adopted, since it represents a more severe test function.

Apart from external disturbances, even a well designed propulsion system, may possess poor dynamic behavior from the point of view of its control. This may be attributed to considerable time delays in accumulators such as steam manifolds. Moreover, turbine rotors tend to be designed rather light and flexible, especially for small driving powers, to avoid whirling of the rotor. This leads to relatively small rotor time constant, which is again inconvenient from the control point of view.

Further, to evaluate the control behavior under adverse conditions, the hydraulic servo-motors are supposed to be slow acting, i.e. to have large time delays. Such increased time delays may be due to deterioration of components of the hydraulic servo-motor affecting the response of the system [2].

Control systems are to be designed such as to maintain deviation tolerances with respect to plant dynamics and the nature of expected disturbance [3]. Few published works exist in the field of treatment of the control of steam turbines. The majority of the published work either tend to adopt simplified mathematical models [4,5] or to present systematic structuring of control systems [6]. However, in [7] a more sophisticated mathematical model was presented, though without numerical evaluation.

In [8], when studying the similarity of dynamic behavior of different heat engines' portions, speed control loops with single input and single output were computationally evaluated using proportional or proportional-derivative controllers with and without a derivative auxiliary controller. Integral control actions were excluded from the analysis.

In [9] a scheme for compound regulation of bled steam pressure and turbine's speed with independent control actions, Fig. (1), was analysed, such independent control actions have the drawback of slow transient response. This is due to the fact that the speed controller interferes only on the main steam governing valve. Moreover, a disturbance introduced from the change in the amount of extracted steam will affect the turbine's power, and consequently the turbine's speed, thus resulting in an additional disturbance on the plant.

Simultaneously-interfering controllers eliminate such drawbacks and hence give a better dynamic response than control systems with independent control actions. Therefore, simultaneously-interfering controllers will be considered here to regulate both turbine's speed and extracted steam pressure in a marine bled steam turbine.

2. Alternatives of Multi-variable Control Systems

It is intended to investigate the time domain response of the closed loop control system composed of the marine steam turbine, as the controlled plant, with a proportional-integral-derivative (PID) speed controller and a proportional controller for extracted steam pressure. Auxiliary energy hydraulic servo-motors, with P control property are

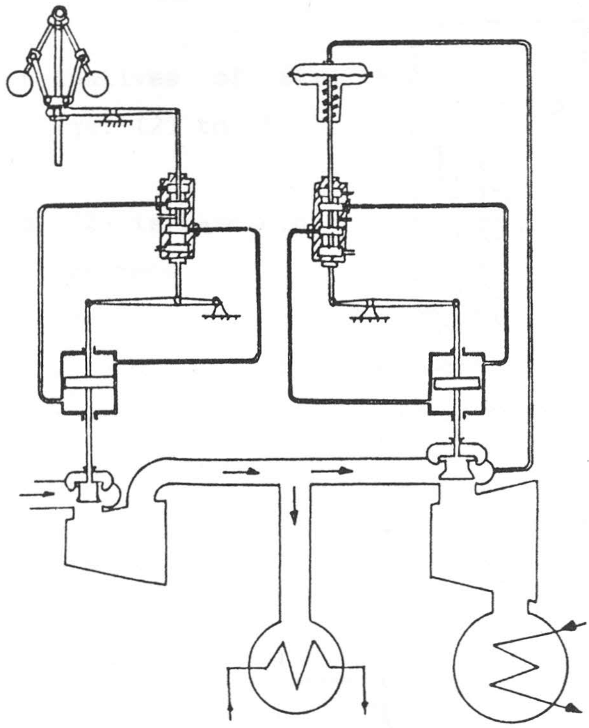


Fig.(1)

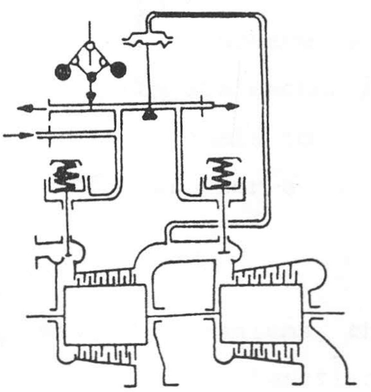


Fig.(2)

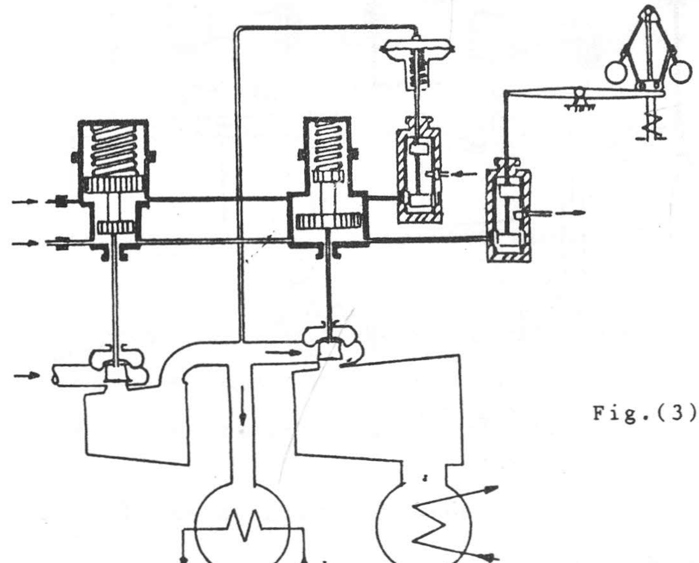


Fig.(3)

Alternatives of multi-variable control systems for marine steam turbines

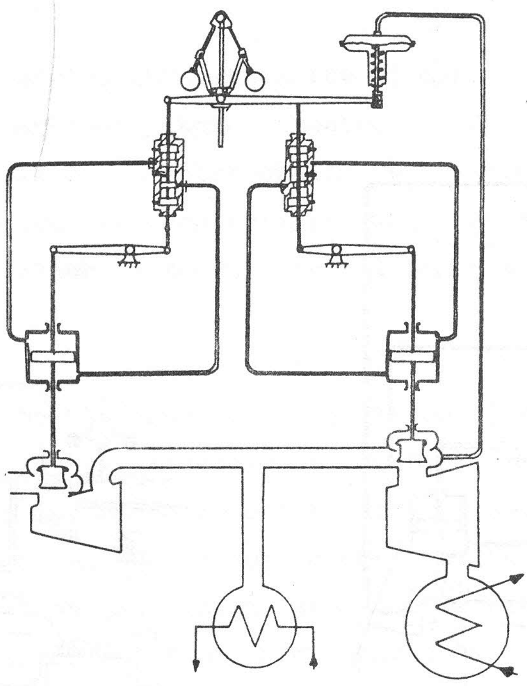


Fig.(4)

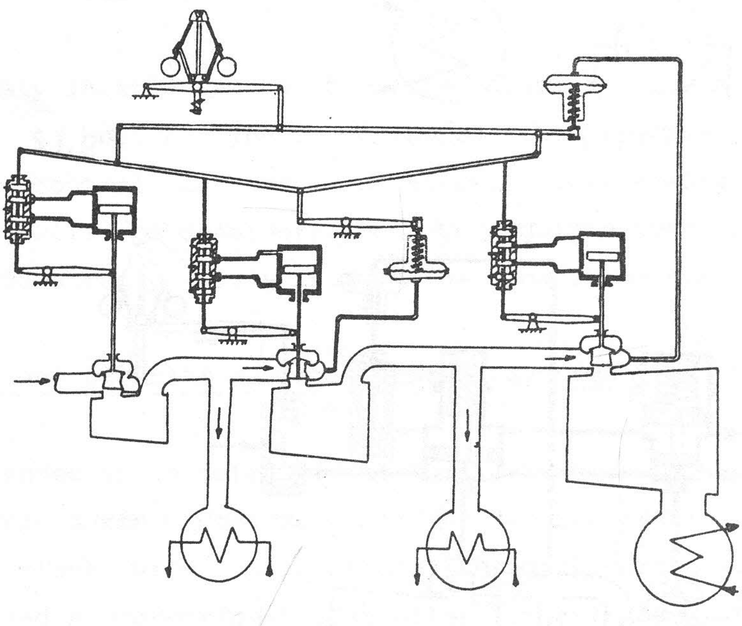


Fig.(5)

Alternatives of multi-variable control systems for marine steam turbines (cont.)

also in use. Each controller affects both steam governing valves for simultaneous control of turbines speed and extracted steam pressure.

Alternatives of simultaneously-interfering control systems are shown in figs. (2) to (5).

Fig. (2) is based on the hydraulic flapper-nozzle principle [10]. Fig. (3) represents a system based on the same principle, however with differential servo-motor [5]. Figs. (4) and (5) incorporate mechanical links and hydraulic servo-motors transfering control signals to steam regulating valves [5]. The system in Fig. (5) is for a double extraction steam turbine.

Despite that these systems posses different control mechanisms, they all reduce to basically similar block diagrams and transfer functions. The system considered here is the one shown in Fig. (4).

3. Mathematical Modelling

The control system of Fig. (4) is mathematically simulated by the block digram shown in Fig. (6), indicating transfer functions, unit step responses and control properties of minor blocks. For determining the transfer functions, reference in made to [11,12,13].

It is worth mentioning that reference signals of speed as well as extracted steam pressure are nullified since we are interested only in the dynamic behavior of the control system under external disturbances. Moreover, all control signals represent non-dimensional values related to the corresponding nominal values.

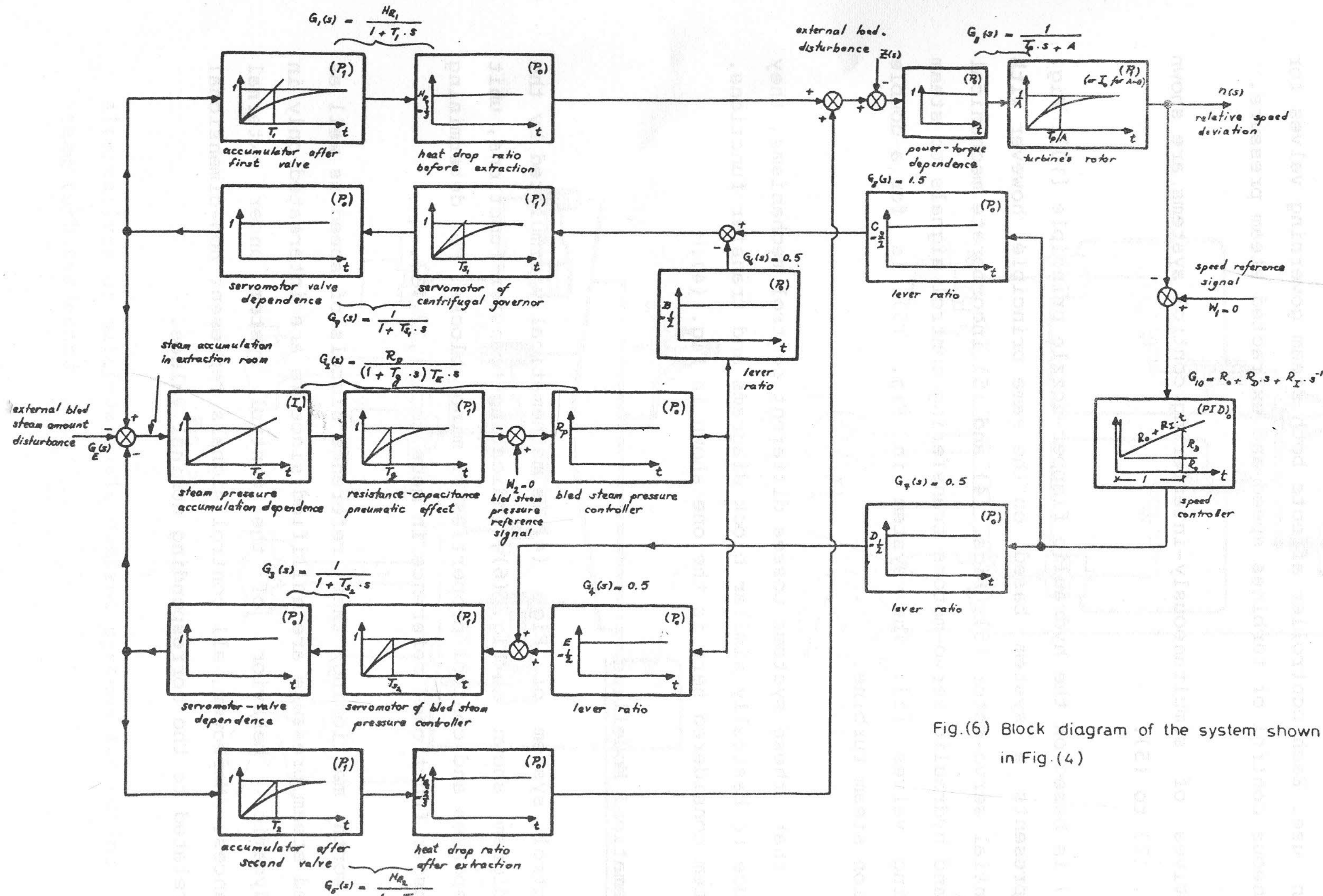


Fig.(6) Block diagram of the system shown in Fig.(4)

The heat drop ratio before steam extraction is, according to common practice, one third of the total heat drop in the turbine. The connection points on the levers connecting the speed and pressure governors to the pilot valves of the hydraulic servomotors are taken equidistant.

By means of the superposition principle, the multivariable block diagram reduction gives the relative speed deviation n(s) in terms of the relative load and steam extraction disturbances, through the transfer matrix. Because of restricted space, the lengthy and tedious manipulations leading to the final form are not reproduced here. The final input-output relation, in matrix form, is given by

$$n(s) = [\, a_{11} \quad a_{12} \,] . \begin{bmatrix} Z(s) \\ G_E(s) \end{bmatrix}$$

where

$Z(s)$ relative external load disturbance

$G_E(s)$ relative bled steam amount

S Laplace operator

$$a_{11} = - \frac{G_{11}(s)\, F_2(s)}{F_1(s)F_2(s) - F_3(s)F_4(s)}$$

$$a_{12} = - \frac{F_3(s)}{F_1(s)\, F_2(s) - F_3(s)\, F_4(s)}$$

$$F_1(s) = 1 + 0.5\, G_{10}(s)\, G_{11}(s)\, [3\, G_1(s)\, G_9(s) + G_3(s)\, G_5(s)]$$

$$F_2(s) = 1 + 0.5\ G_2(s)\ [G_3(s) + G_9(s)]$$

$$F_3(s) = 0.5\ G_2(s)\ G_{11}(s)\ [G_3(s)\ G_5(s) - G_1(s)\ G_2(s)]$$

$$F_4(s) = 0.5\ G_{10}(s)\ [G_5(s) - 3G_9(s)]$$

$G_1(s)$ to $G_{11}(s)$ are the transfer functions indicated in Fig. (6). It is noted that both a_{11} and a_{12} are negative. This reveals that an increase in $Z(s)$ and/or $G_E(s)$ would result in a decrease of speed.

4. Numerical Treatment

The 6th degree characteristic equation of the system was factorized in search of all its roots. A search algorithm based on Newton general method for finding the roots of a polynomial [14] was adopted to pick up the roots to an accuracy of 10^{-6}.

Roots were found to be both pure real as well as complex. Since all the real parts of all roots were found to be negative, the absolute stability of the system was assured.

The general formula for the inverse laplace transform of the quotient of two polynomials giving the speed deviation n(t) can be expressed as [15]:

$$n(t) = \sum_{i=1}^{m} a_i e^{-\alpha_i t} \sin\ (\beta_i t + \theta_i) + \sum_{i=m+1}^{k} b_i e^{-\alpha_i t},$$

where k is the total number of roots of the denominator, m is the number of complex root pairs, and a_i, b_i, α_i, β_i and θ_i are the inverse Laplace coefficients. n(t) was then finally evaluated at different time intervals. The selection of the used time increments was based on optimizing the intervals depending on the shape of the time response curve [15].

Single precision results were found to be inacurate, e.g. having non-zero values at t=0, a condition not complying with the initial value theorem or with the expected transients according to time domain system identification through graphical analysis. The inconsistency of the results was eliminated through more refinement of the roots and using double precision arithmetic.

All numerical computations were carried out with programs written in FORTRAN IV on the PDP 11 computer of the Faculty of Engineering, Alexandria.

5. Numerical Evaluation of Time Response for Different System Parameters

For the numerical evaluation of the time response, different control system parameters were considered.

These parameters and their used values are given below:

T_1, T_2 time delays due to steam accumulators after the main steam governing valve and the second governing valve, respectively.

$$= \frac{\text{Vol. of accumulator}}{\text{steam vol. flow rate}}$$

$T_1 = 0.3$ sec; $T_2 = 0$ sec

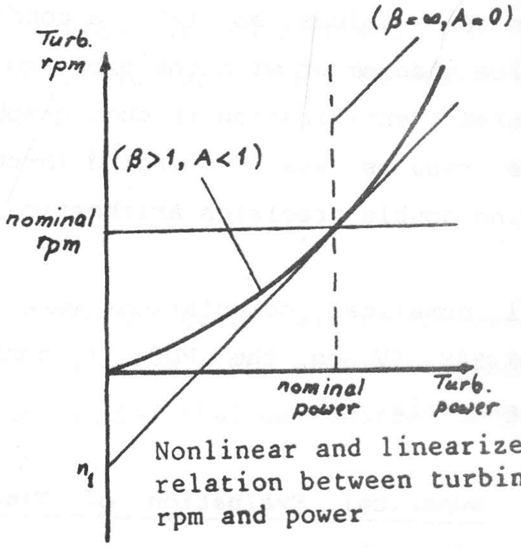
T_p rotor time constant

$$= \frac{\text{rotor polar mass moment of inertia x nominal ang. vel.}}{\text{nominal driving torque}}$$

$T_p = 16$ sec

A rotor's proportionality constant

$$A = \frac{1}{\beta}\ ; \quad \beta = \frac{(rpm)_{nom.} - n_1}{(rpm)_{nom.}}$$



Nonlinear and linearized relation between turbine's rpm and power

If A is nonzero, it leads to a (P_1) control property. If A = 0 ($\beta = \infty$), it leads to (I_o) control property, which is more difficult to control.

A = 0

T_E = (extraction room vol. x max. extracted steam press.)/(superh. steam const. x abs. steam temp. x max. mass rate of steam extraction).

$T_E = 10$ sec

T_g time delay due to pneumatic resistance capacitance(RC) effect in the pipe between extraction chamber and pressure controller

T_g = (extraction room vol.)/(superh. steam const. x abs. steam temp. x orifice coeff. x orifice cross-section)

$T_g = 0$ sec

H_{R_1}, H_{R_2} ratio of heat drop before and after steam extraction related to total heat drop

$H_{R_1} = 1/3$, $H_{R_2} = 2/3$

R_o speed controller's gain

$R_o = 20, 25, 30$ (non-dimensional)

R_I integral property coefficient of speed controller

$R_I = 0, 0.8 \ sec^{-1}$

R_D derivative property coefficient of the speed controller

$R_D = 0, \ 8$ sec

R_P extracted steam pressure controller's gain

$R_P = 20$ (non-dimensional)

T_{s1}, T_{s2} time delays of hydraulic servo-motors interfering on the main steam governing valve and the second governing valve, respectively.

T_s = (cross sectional area of power piston)/ (Proportionality coeff. of the uncovered portion of pilot valve).

$T_{s1} = 0.3$ sec, $T_{s2} = 0.3$ sec

B,C,D,E lever ratios for transfering control signals from speed and pressure controllers to servomotors

B = 0.5

C = 1.5

D = 0.5

E = 0.5

6. Results and Discussion

Results for the time domain response to a superposition of unit step load and steam extraction disturbances were evaluated for the considered system parameters. The results are illustrated in Figs. (7)

to (13).

Fig. (7) illustrates the comparsion of the time domain behavior under control of P- or PD-controllers. It can be seen that the P-controller results in higher speed deviation together with larger peak time compared with PD-controller, for the same proportionality gain of the controller R_o. Further, increasing R_o and/or R_D decreases maximum speed deviation and peak time. Significant increase only in R_o, however, may have more appreciable effect than inereasing only R_D while keeping R_o constant.

It is worth mentioning that for plants having poor control characteristics such as small rotor time constant, large time delays (either in order or in magnitude) or subjected to severe disturbances, such as in the case considered here, the D property is incorporated, and even increased, in order to improve the time domain requirements.

The results for P- and PI-controllers are shown in Fig. (8). The effect of increasing only R_I does not have a considerable effect either on the maximum speed deviation or on the peak time.

Fig. (9) illustrates the results for PI and PID controllers. From this curve it can be seen that with $R_D = 0$, increasing R_o from 20 to 30 results in a reduction of about 20% in the maximum speed deviation. On the other hand, introducing D property with $R_D = 8.0$ sec, results in a reduction in the maximum speed deviation also by about 20%, irrespective of the value of R_o. Since a PID-controller means much more manufacturing cost compared to a PI controller, it can be said that increasing R_o alone is economically preferable to introducing D-property, at the same value of R_I.

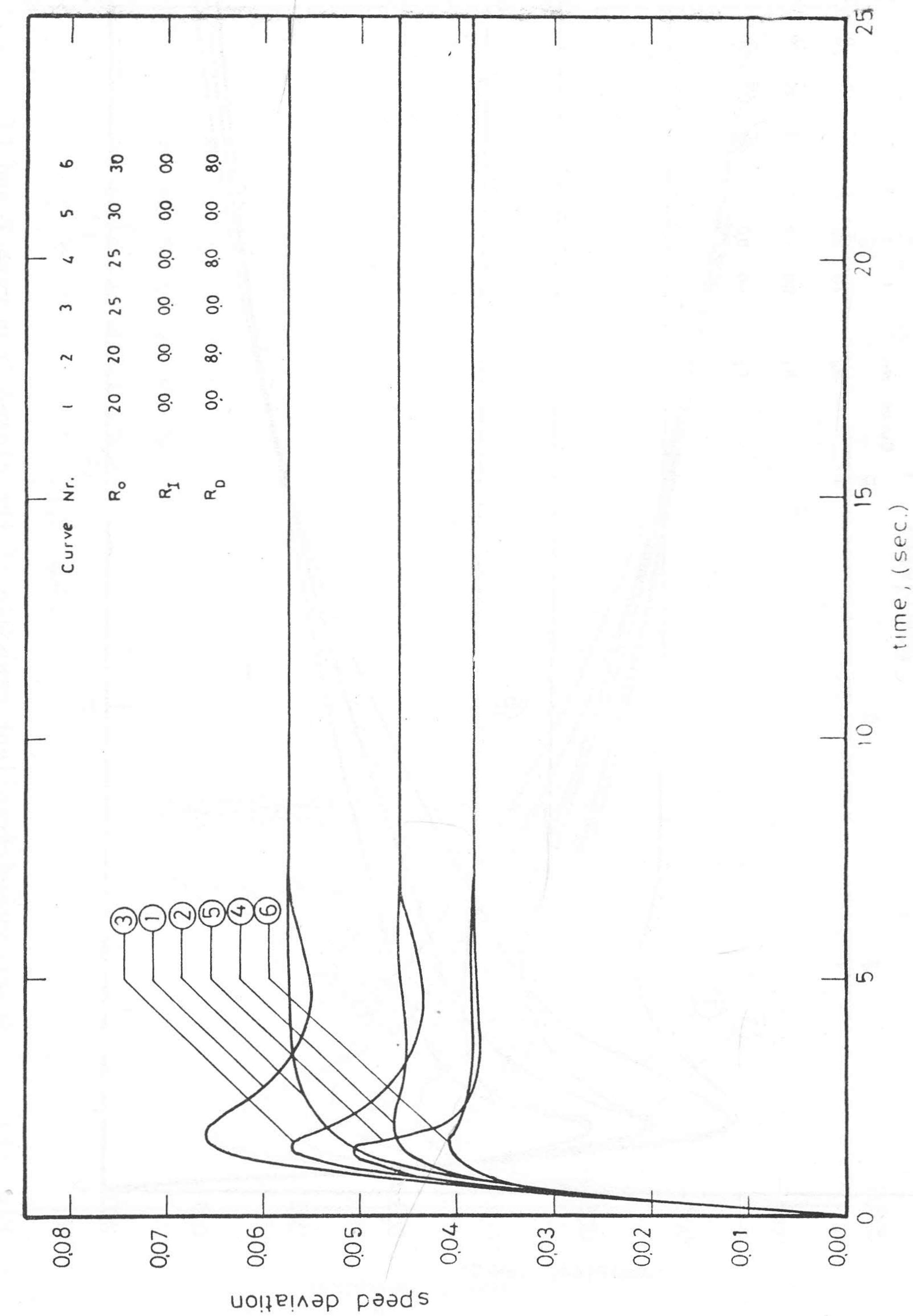


Fig. (7) Superimposed transient response of the closed loop with P and PD speed controlles

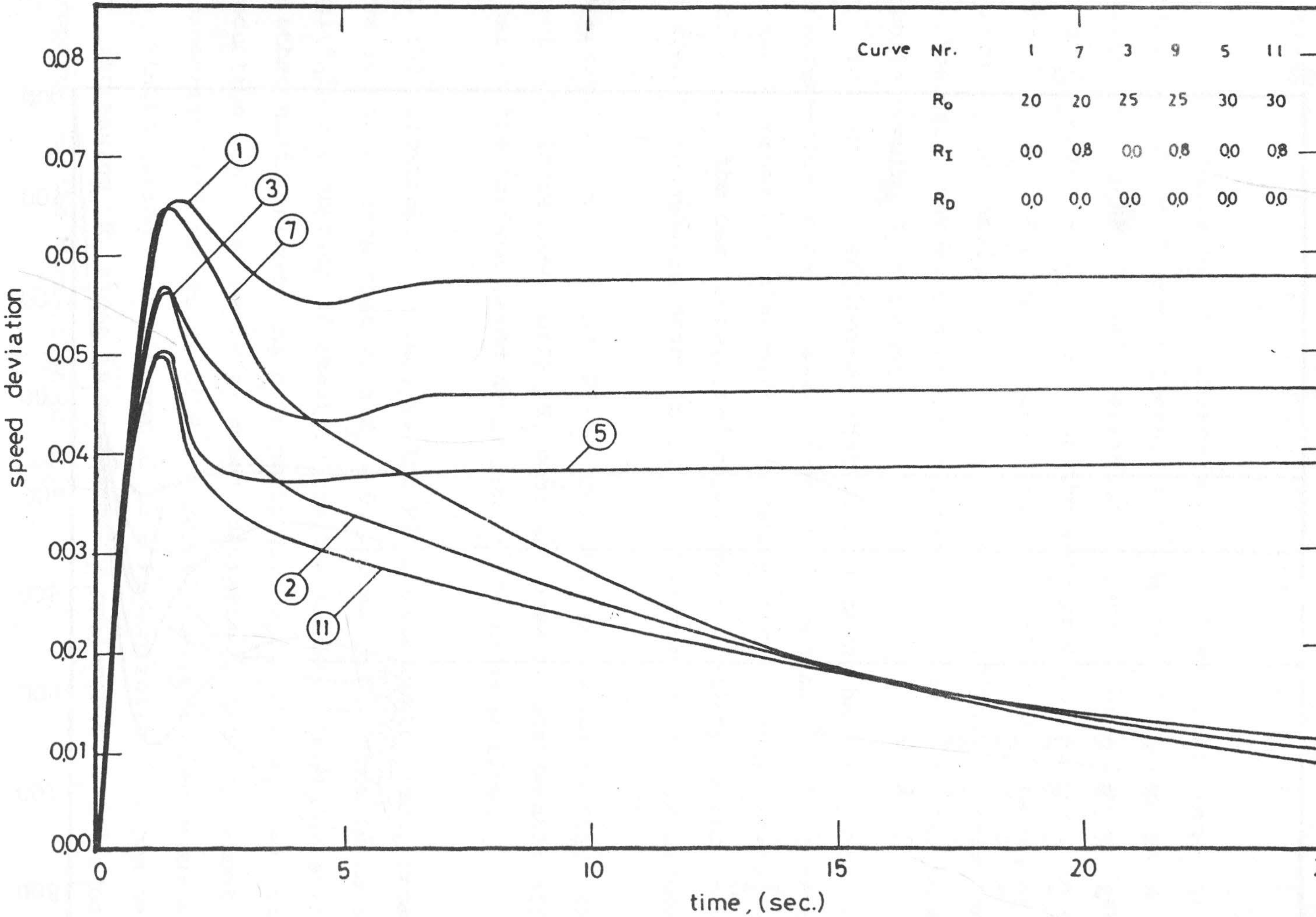


Fig. (8) Superimposed transient response of the closed loop with P and PI speed controlles.

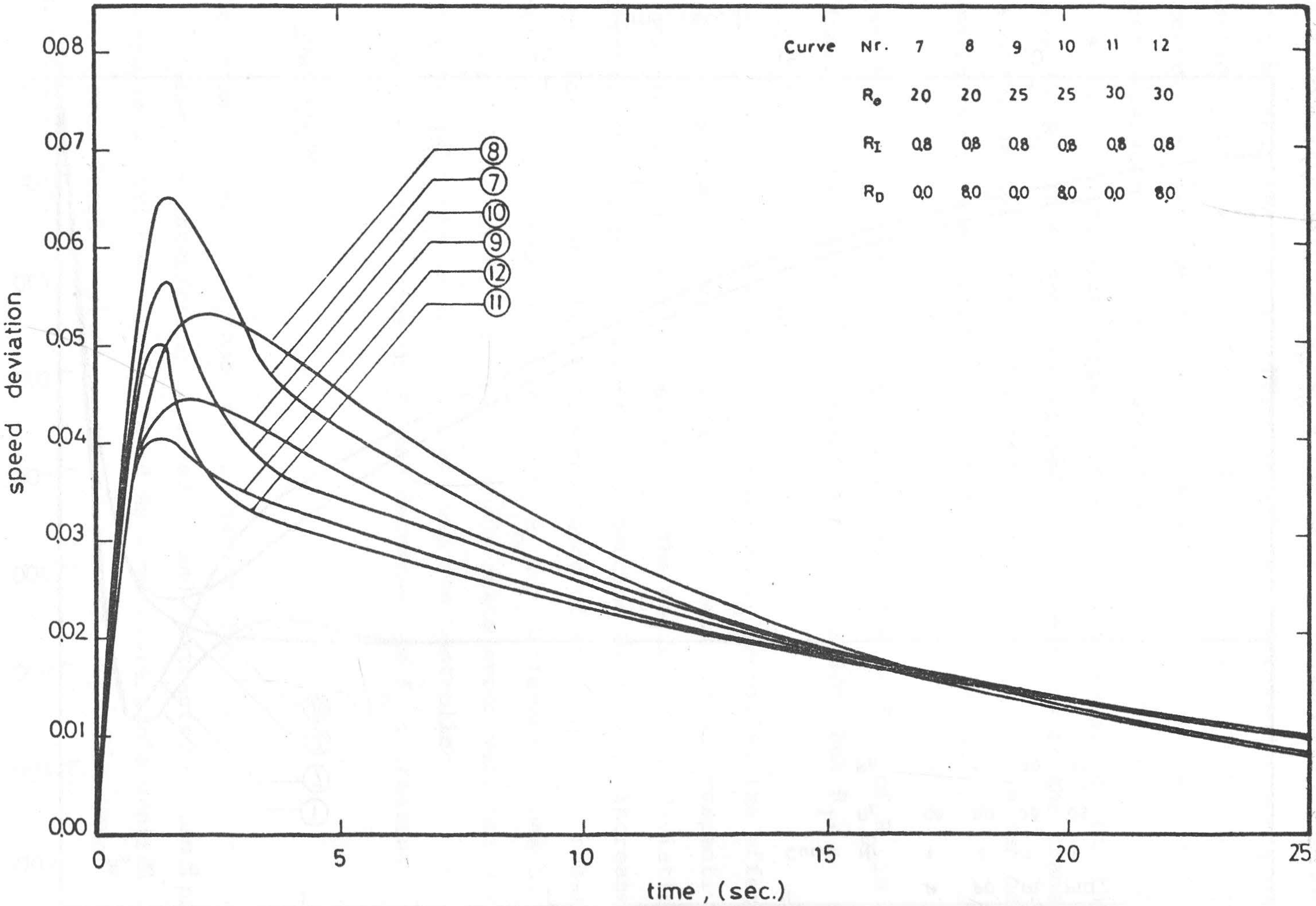


Fig. (9) Superimposed transient response of the closed loop with PI, PID speed controlles.

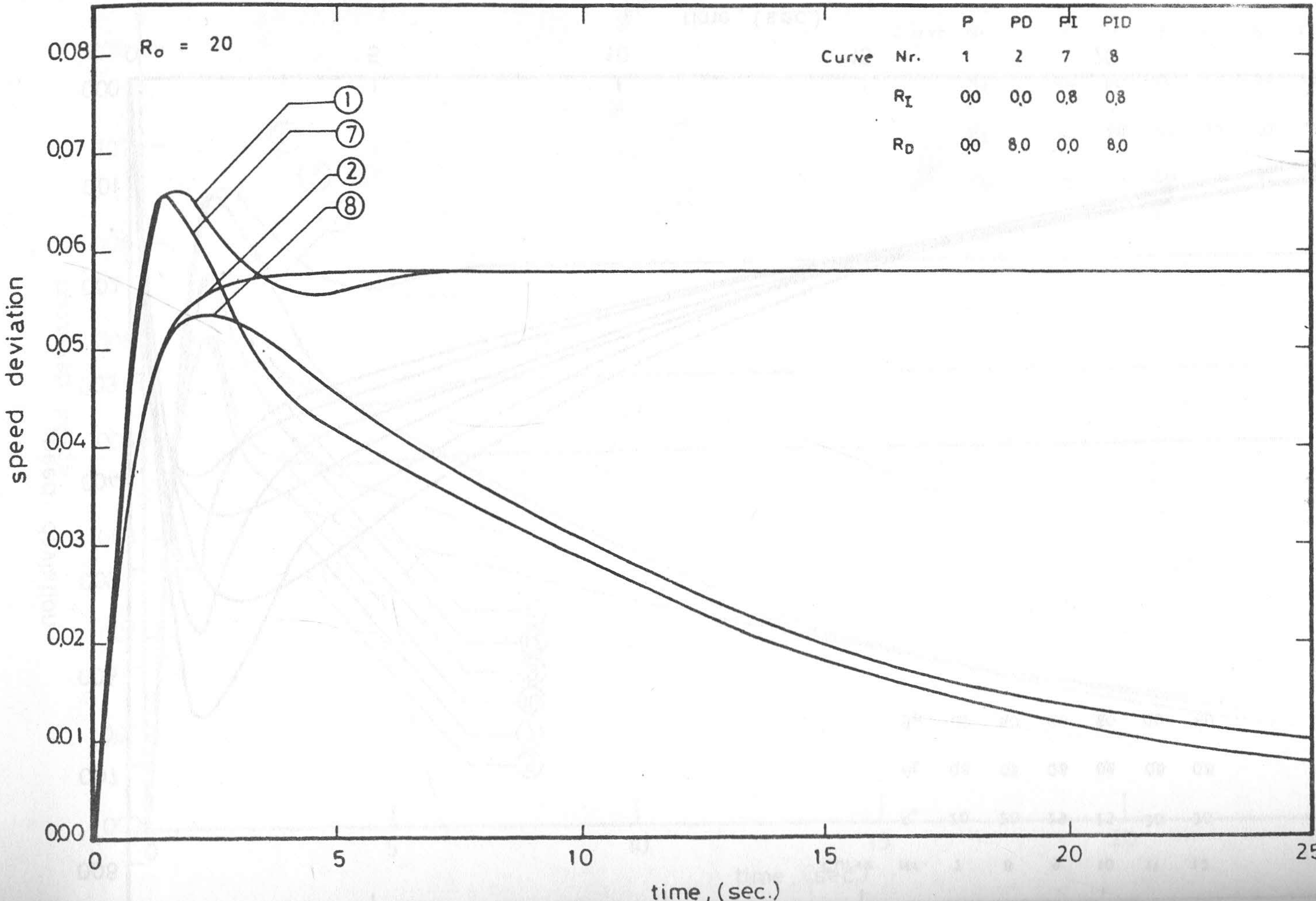


Fig. (10) Superimposed transient response of the closed loop with P,PD,PI,PID speed controllers for constant R_o

For a specific value of R_o (R_o = 20), Fig. (10) shows a comparison of the response of P, PD, PI and PID controllers. Apart from the question, whether it is required to keep constant the turbine's speed irrespective of the external disturbances as in the case of a turbo-generator, it can be seen that a PD - controller is preferable to a PI - controller with respect to speed deviation, for the same value of R_o. This fact is confined to the begining of the transient response, particularly with zero order of delay, as in our case. As time increases the effect of the I-property dominates.

From Fig. (11) it can be seen that the effect of R_D is more effective in improving the maximum speed deviation than R_I.

Figs. (11) and (12) show that increasing R_D magnifies the effect of R_I on maximum speed deviation and peak time, respectively, particularly for lower values of R_o. The maximum speed deviation is improved by D-property. However, introducing D-property increased peak time for most of the considered range of R_o. Fig. (13) shows the steady state error with respect to R_o for different values of R_I and R_D. It is evident that the steady state error vanishes when an integral property is incorporated into the controller. Further, it is independent of the value of R_D and decreases as R_o increases.

7. Conclusion

The time domain response of a marine bled steam turbine with simultaneously-interfering speed and extracted steam pressure controllers has been investigated as a multi-variable control system. The considered plant was chosen to represent poor control characteristics.

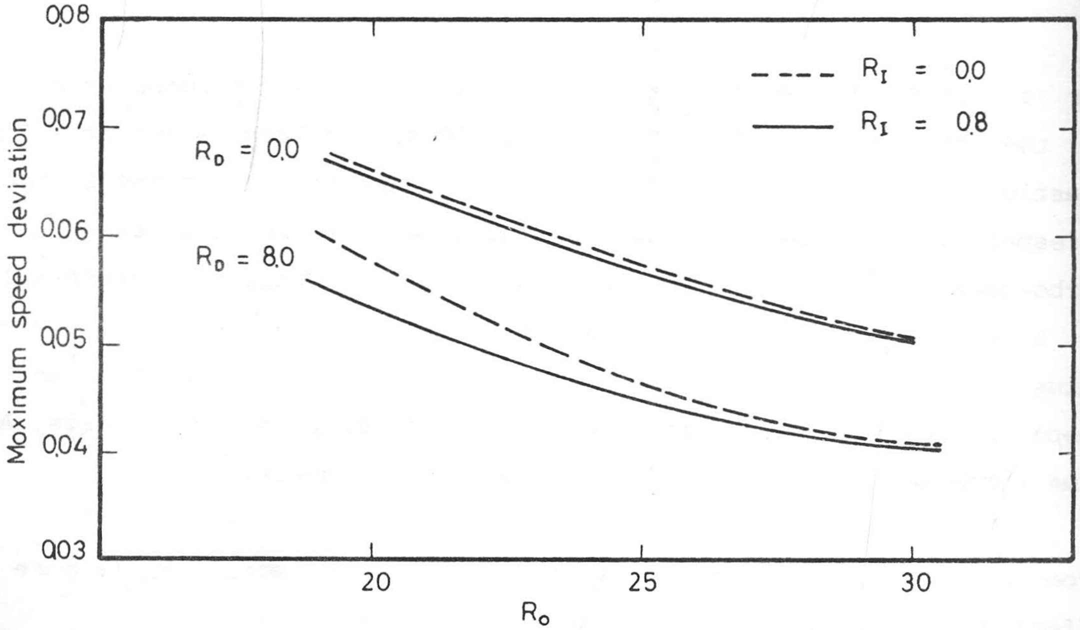


Fig. (11) Maximum speed deviation versus R_O

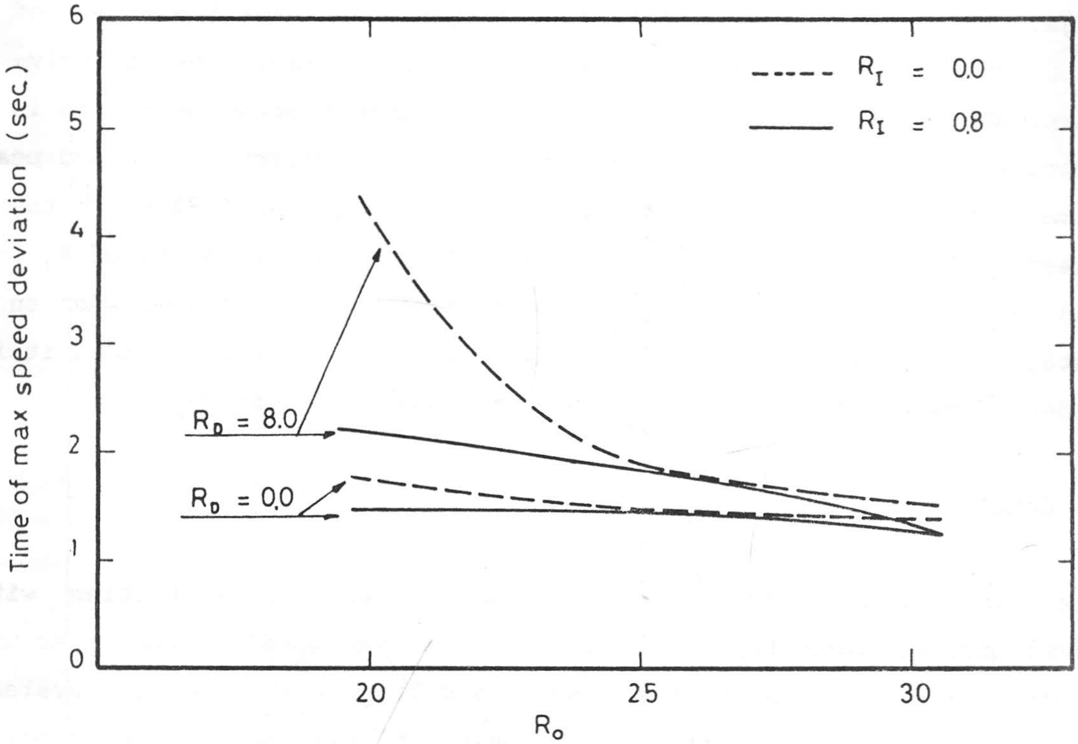


Fig. (12) Peak time versus R_O for different combinations of R_I and R_D

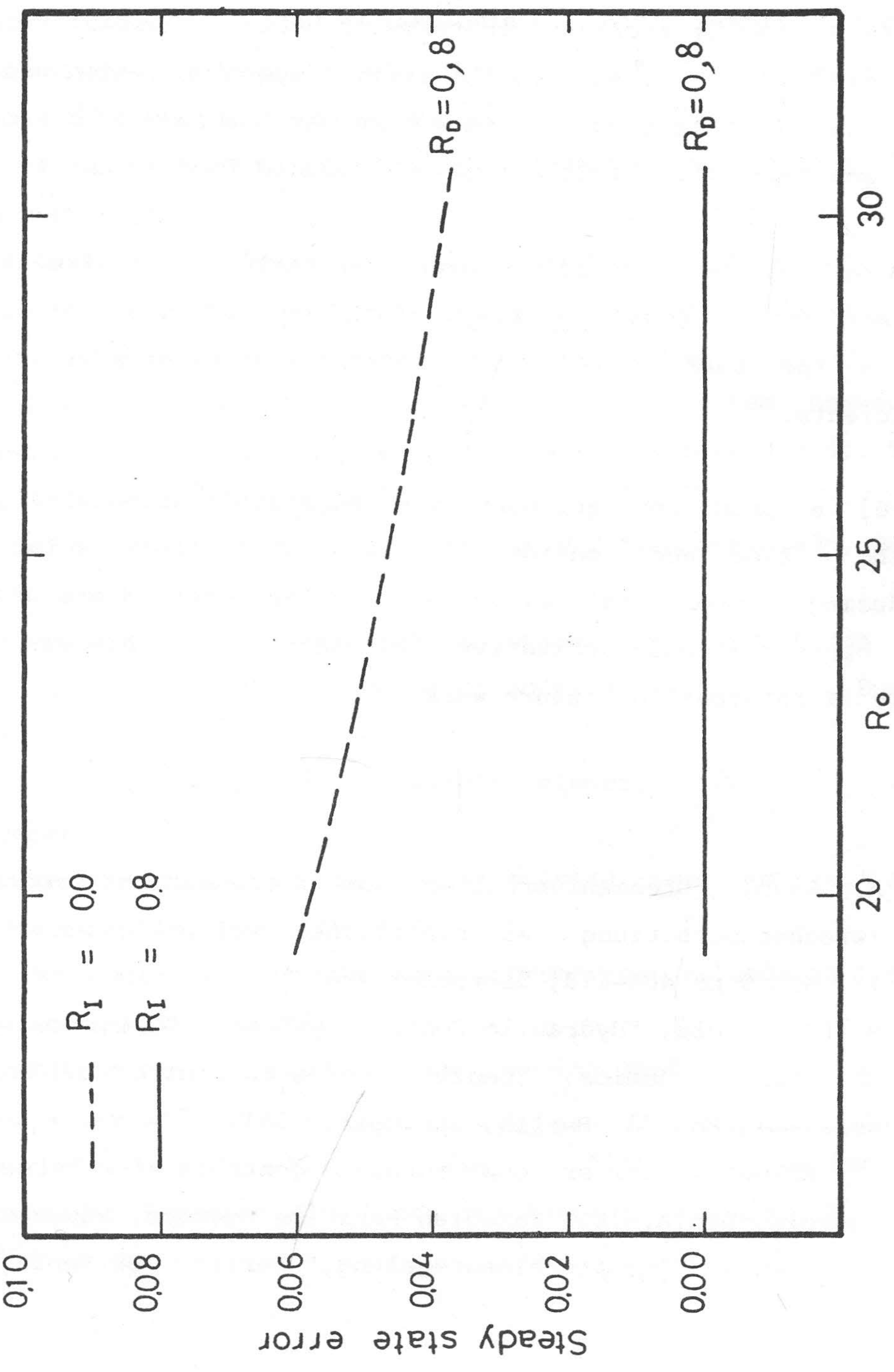


Fig. (13) Steady state error versus R_o for different combinations of R_I and R_D

The study displays the improved performance as compared with controllers having independent control action, as treated in [9]. This improvement can be expressed in terms of better transient performance specifications. For instance, the maximum speed deviation was reduced from about 10.3% to 6.5%. The peak time was improved from about 5 sec to 1.5 sec. Also the settling time was reduced from 12 sec to 7 sec.

A comparison between simultaneously-interfering controllers with different control properties was also carried out. The analysis should help in the task of selecting properly controller's properties and coefficients.

In [16] a promising approach was suggested for obtaining quicker settling time and considerably less ITAE index value through introducing proportional minus delay (PMD) controllers as compared with proportional plus derivative (PD) controllers. This may represent a topic of interest for future work.

8. References

[1] A. Raab, "Drehzahlverhalten von Kondensationsturbosätzen mit Zwischenüberhitzung bei plötzlicher Vollentlastung," BWK Vol. 13, No. 9 pp 404-413, September 1961.

[2] P. Dransfield, "Hydraulic Control Systems - Design and analsysis of their dynamics," Lecture notes in control and information sciences, No. 33, Berlin, Springer, 1981.

[3] L. McEwen, "Boiler and turbine controls for steam driven vessels,"S.N.A.M.E., Canadian Maritime Section, November 1968.

[4] O. Graul, "Dampfturbinenregelung," Berlin, VEB Verlag Technik,

1960.

[5] I. Kirilow, "Regelung von Dampf- und Gasturbinen," Berlin, VEB Verlag Technik, 1956.

[6] G. Mathias, "Regelungsaufgaben and Dampfturbinen- Eine Struktursystematische Untersuchung," VGB Kraft werktechnik, Heft 2 pp 119-124, February, 1979.

[7] W. Traupel, "Thermische Turbomaschinen," 2. Bd, 2. Aufl. Berlin, Springer, 1968.

[8] M. Hanafi S., "Behaviour of speed regulating systems of heat engines as control systems under external disturbances," Ph.D. thesis, Hungarian Academy of Sciences, Budapest 1979.

[9] M. Mosleh, "Compound regulation of bled steam pressure and turbines speed in marine steam power plants," M.Sc. Thesis, Alexandria University, 1983.

[10] A. Donko and I. Móriez, "Gözturbinák, "Tankönyvkiadó, Budapest 1961.

[11] R. Száday, "200 MW-os turbina-generátor egységy rotorja axiális elmozdulásának analóg számológépi modellezése," B.M.E., Budapest, 1974.

[12] R. Szaday, "A szabályozáselmélt elemei," Müszaki Konyvkiadó, Budapest, 1972.

[13] L. Helm [Ed.], "A szabályozástechnika kézi könyve," Müszaki Könyvkiadó, Budapest, 1970.

[14] J. McCormick and M. Salvadori, "Numerical Methods in FORTRAN", New York, Prentice Hall, 1971.

[15] D. McCracken, "FORTRAN with engineering applications," New York, John Wiley and Sons, 1967.

[16] I.H. Suh and Z. Bien, " Use of Time-Delay Actions in the Controller Design," IEEE Trans. on Automatic Control, Vol. Ac-25, No. 3, June 1980.