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SIMULTANEOUSLY - INTERFERING CONTROLLERS FOR A MARINE STEAM TURBINE - A STUDY ON CONTROLLER'S BEHAVIOR IN FREQUENCY DOMAIN

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Abstract

A frequency domain study on a multivariable closed loop control system of a marine bled steam turbine with simultaneously-interfering controllers was carried out. The dynamic behavior of the closed loop with different controllers' properties was investigated revealing an inconventional concept for choosing controller properties in case of periodic disturbances.

1. Introduction

Frequency response methods represent a powerful tool for studying both absolute and relative stability, improving system response through compensation techniques or internal feedback and for the analysis and design of single input-single output or multivariable linear or nonlinear control systems [1].

Control systems can be identified in both time [2] and frequency domains [3,4] in order to determine experimentally [5] their mathematical models, especially for the complicated ones, and hence their control properties and dynamic behavior can be obtained. Moreover, by means of Fourier transform, time-frequency correlation can be expressed as [4]

$$f(t) = \frac{1}{2\pi} \int_{-\infty}^{\infty} g(\omega)\, e^{i\omega t} d\omega$$

$$g(\omega) = \int_{-\infty}^{\infty} f(t)\, e^{-i\omega t} dt,$$

which means that the time response can be obtained from the frequency response and vice versa.

Especially in marine applications, disturbances of periodic nature are common, e.g. resulting from the periodic change of the driving torque in a marine Diesel propulsion plant. However, the main source of periodic disturbance on marine power plants is attributed to the ship motion among waves [6,7]. Sea waves can be mathematically represented as sinusoidal waves with a specific frequency. However, the disturbance frequency depends on the so-called encounter frequency. This frequency depends on wave frequency, ship's speed, wave number and encounter angle [6,7,8].

Such sinusiodal disturbances will result in corresponding sinusoidal fluctuations in propeller torque, thus representing an external disturbance on the main engine.

The system considered here is a multi-variable control system for a marine bled steam turbine subjected to two sources of external disturbances, namely the fluctuation of propeller torque due to waves and the fluctuation of the amount of extracted steam due to changing demand and/or regional temperature variations affecting the condensate outlet temperature from feedwater heaters and consequently the amount of bled steam.

The fluctuations in the extracted amount of steam, hence the pressure in the extraction steam room, are considered in this treatment to be also of sinusoidal nature with the same frequency superimposed on the sinusoidal torque fluctuation.

In [9] the time domain analysis of a marine bled steam turbine with simultaneously-interfering controllers was studied. It was shown that such control systems improve the time domain characteristics compared to similar control plants with independent control actions.

In this work it is aimed to calculate this improvement in the frequency domain. Besides, an analysis into the dynamic behavior of controllers in frequency domain as compored to their behavior in the time domain is carried out. The same mathematically simulated multivariable closed loop control system as in [9] with its inherent poor dynamic properties will be considered.

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2. The Considered System and Its Mathematical Model

Fig. (1) shows the control system for regulating both turbine's speed and extracted steam pressure with simultaneously-interfering controllers. Both speed and pressure controllers are connected through a walking beam with the joints taken equidistant. The walking beam is connected in turn to both pilot values of the two hydraulic servomotors which actuate both steam governing valves. Both hydraulic servomotors have proportional control property with first order time delay. This is achieved by the movable casing of each pilot valve which is connected to the power piston through a minor feedback pivoted lever.

A decrease in resisting torque on the propeller, resulting in an increase in the speed, will lead to the partial closure of both steam governing valves thus reducing the turbines' speed. On the other hand, an assumed increase in the amount of extracted steam, due to a change either in demand and/or in regional temperature, leads to a decrease in the steam pressure in the extraction room. The pressure controller will interfere on both servo-motors through the walking beam in order to close partially the steam admission valve after extraction - i.e. regulate the steam pressure in the extraction room - and to open partially the steam main governing valve to compensate for the partial loss in turbine's power and speed.

If each controller were to interfere independently on only one of the two governing valves, a decreases in resisting torque would have resulted in the partial closure of the main governing valve only. Consequently, the response would be slower than when both controllers interfered simultaneously. Moreover, with both controllers acting

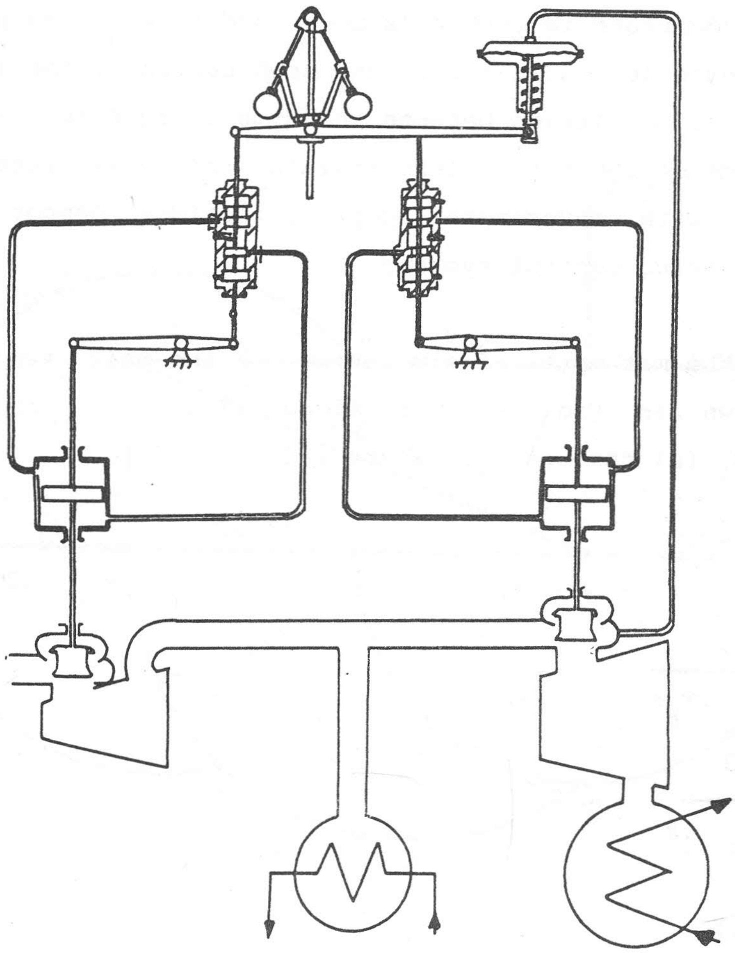

Fig. (1) A typical multi-variable control system of a marine steam turbine with simultaneously-interfering speed and bled steam pressure controllers independently, an increase in the amount of extracted steam would result in only partial closure of the steam governing valve located after the extraction room. Thus in order to regulate the extracted steam pressure, a decrease in turbine's power and speed takes place urging the speed controller to interfere and open partially the steam governing valve. The time elapsed between the drop in turbine's speed and the interference of the controllers explains the slower response of control systems with independent control actions compared to simultaneously-interfering control systems.

Fig. (2) represents the mathematical simulation of the multi-variable control system shown in Fig. (1) in signal flow graph form. The transfer functions G_1 (s) through G_{11}(s) are [9]:

$$G_1 (s) = \frac{H_{R_1}}{1 + T_1 s}$$

$$G_2 (s) = \frac{R_p}{(1+T_g.s)T_E s}$$

$$G_3 (s) = \frac{1}{1 + T_{s_2} .s}$$

$$G_4 (s) = E = 0.5$$

$$G_5 (s) = \frac{H_{R_2}}{1 + T_2 . s}$$

$$G_6 (s) = B = 0.5$$

$$G_7 (s) = D = 0.5$$

$$G_8 (s) = C = 1.5$$

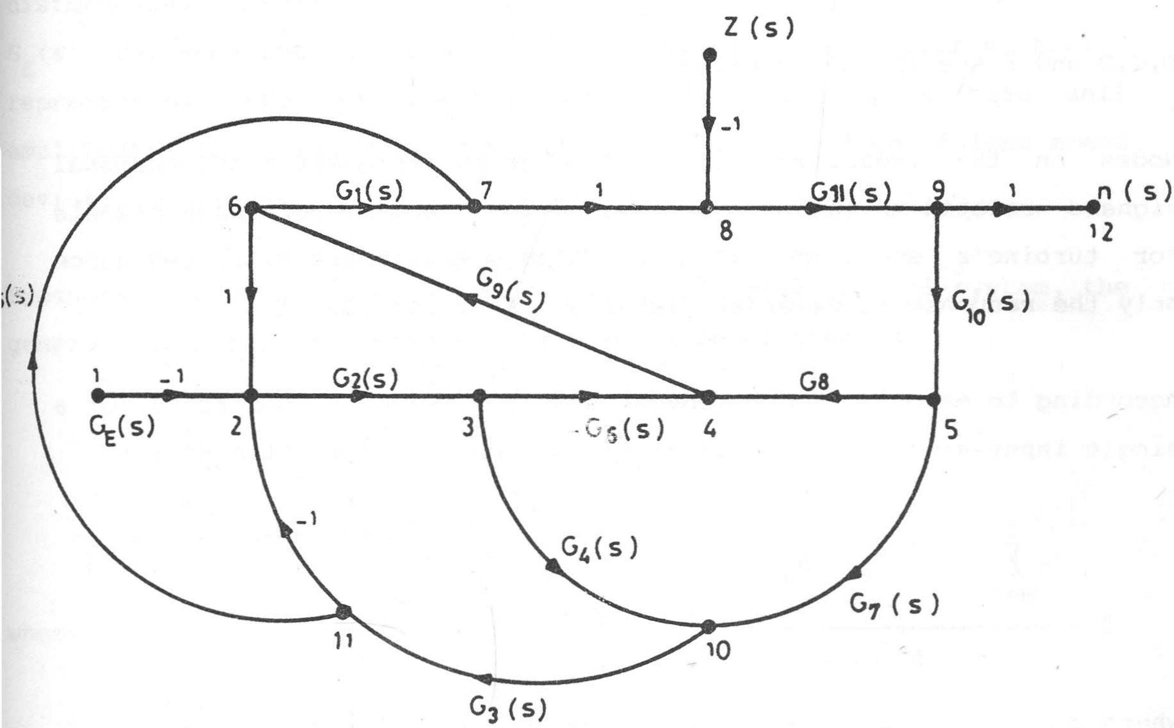

Fig. (2) Mathematical simulation of the control system in signal flow graph representation

$$G_9(s) = \frac{1}{1+T_{s_1}.s}$$

$$G_{10}(s) = R_o + R_D.s + R_I s^{-1}$$

$$G_{11}(s) = \frac{1}{T_p.s + A}$$

B,C,D and E are the lever ratios.

Nodes on the indicated signal flow graph represent nondimensional signals related to their respective nominal values. Reference signals for turbine's speed and extracted steam pressure are nullified since only the response to external disturbances is considered.

According to Mason's theorem the closed loop transfer function T for a single input-single output signal flow graph can be written as

$$T = \frac{\sum_{m=1}^{1} P_m \quad \Delta_m}{\Delta}$$

where :

1 = number of forward paths

P_m = the m-th forward path gain

Δ = signal flow graph determinant or characteristic function

$$\Delta = 1 - (-1)^{k+1} \sum_k \sum_i Q_{jk}$$

$$= 1 - \sum_j Q_{j1} + \sum_j Q_{j2} - \sum_j Q_{j3} + - \dots$$

= 1 - (sum of all loop gains) + (sum of all gain products of two non-touching loops)

- non-touching loops) + - ...

Q_{jk} = j-th possible product of k non-touching loop gains

Δ_m = Δ evaluated with all loops touching the m-th path eliminated

In the considered system here, there are two dimensionless disturbances, namely Z(s) representing the torque disturbance and $G_E(s)$ representing the amount of extracted steam disturbance, thus representing the two inputs. Both sinusoidal inputs have unit amplitude. The single output of system is the dimensionless speed deviation n(s).

Expanding Mason's theorem for a multi-variable control system, the general element of the transfer matrix may be written as

$$a_{ij} = \frac{\sum_{m=1}^{l_{ij}} P_m \Delta_m}{\Delta}$$

where j = 1,2, ... , no of outputs

i = 1,2. ... , no of inputs

l_{ij} = no. of forward paths between input node i and output node j

In our case this yields

$$\Delta n(s) = \begin{bmatrix} \dfrac{\sum_{m=1}^{l_{11}} P_m \Delta_m}{\Delta} & \dfrac{\sum_{m=1}^{l_{21}} P_m \Delta_m}{\Delta} \end{bmatrix} \begin{bmatrix} Z(s) \\ G_E(s) \end{bmatrix}$$

where l_{11} and l_{21} are number of forward paths for the first and second input, respectively.

For the first input Z(s) there exists only one forward path (l_{11} =1). The gain of this path is

$$P_1 = - G_{11}(s)$$

For the second input $G_E(s)$ there are two forward paths (l_{21} = 2), their gains are given by

$$P_1 = G_1(s)\ G_2(s)\ G_6(s)\ G_9(s)\ G_{11}(s)$$
$$P_2 = G_2(s)\ G_3(s)\ G_4(s)\ G_5(2)\ G_{11}(s)$$

There exist six loops, their gains are given by:

$$L_1 = - G_2(s)\ G_3(s)\ G_4(s)$$
$$L_2 = - G_2(s)\ G_6(s)\ G_9(s)$$
$$L_3 = G_2(s)\ G_3(s)\ G_4(s)\ G_5(s)\ G_8(s)\ G_9(s)\ G_{10}(s)\ G_{11}(s)$$
$$L_4 = G_3(s)\ G_5(s)\ G_7(s)\ G_{10}(s)\ G_{11}(s)$$
$$L_5 = G_1(s)\ G_8(s)\ G_9(s)\ G_{10}(s)\ G_{11}(s)$$
$$L_6 = G_1(s)\ G_2(s)\ G_3(s)\ G_6(s)\ G_7(s)\ G_9(s)\ G_{10}(s)\ G_{11}(s)$$

The only two non-touching loops are L_2 and L_4. There exist no combinations of three non-touching loops.

Δ is given by

$$\Delta = 1 - \sum_{i=1}^{6} L_i + L_2 . L_4$$

For the first element of the transfer matrix:

$$\Delta_1 = 1 - (L_1 + L_2).$$

For the second element of the transfer matrix :

$$\Delta_1 = 1$$
$$\Delta_2 = 1$$

The elements of the transfer matrix correspond to the elements $a_{11}(s)$ and $a_{12}(s)$ as given in [9].

3. Numerical Treatment

Both closed loop frequency responses due to the considered disturbances, namely variations in resisting torque and amount of extracted steam were analysed numerically using FORTRAN programs modified from [10] utilizing the complex algebra facility included in this programming language. The programs were run on the PDP 11 computer of the Faculty of Engineering, Alexandria University. The output of the computer programs yields the closed loop frequency response for each of the two considered disturbances in both polar and Bode forms.

To obtain the superimposed frequency response to both disturbances, the phase angle between both outputs should be taken into consideration. The worst condition occurs when the output vectors have zero phase angle shift.

The question was raised as to which range of excitation frequencies to be considered. The frequencies were chosen such that they represent actual excitation frequencies from statistical wave data [11]. However, attention was paid to the concept of encounter frequency [6,7,8]. From wave statistics, the frequency of wave oscillations ranges from 0.05 to 1 Hz, i.e from about 0.3 to 6 rad/sec. The encounter frequency ω is given by [8]

$$\omega = \omega_W - Uk \sin \varphi$$

where

ω_W = wave frequency (rad/sec)
U = ship's speed (m/sec)
k = wave number $= 2\pi/\lambda$ (rad. m^{-1})
λ = wave length (m)
φ = encounter angle

For the above mentioned wave frequencies and for a ship travelling at, say, 15 knots, both for head and following seas the corresponding range of encounter frequency is from about 0.03 to 6.76 Hz, i.e from about 0.2 to 40 rad/sec. It is further to be noted that for high wave frequencies the encounter frequency may turn to be negative, in which case the absolute value is to be considered [8].

For our study, the frequency range was chosen to represent two decades, namely from 0.1 to 10 rad/sec. The following decade of the exciting frequency (i.e from =10 to 100 rad/sec) was discarded due to the fact that control systems possess much more poles than zeros. This means that with higher frequencies the system response rapidly decays.

The definitions of the control system's time constants and parameters are given in [9]. Here we summarize the numerical values chosen to represent poor control characteristics of the plant:

$T_1 = 0.3$ (sec)

$T_2 = 0$ (sec)

$T_p = 16$ (sec)

$A = 0$ (-)

$T_E = 10$ (sec)

$T_g = 0$ (sec)

$H_{R_1} = \frac{1}{3}$ (-)

$H_{R_2} = \frac{2}{3}$ (-)

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

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

$R_D = 0,8$ (sec)

$R_p = 20$ (-)

$T_{s_1} = T_{s_2} = 0.3$ (sec)

$B = D = E = 0.5$ (-)

$C = 1.5$ (-)

4. Results and Discussion

The results are illustrated in Fig. (3) to (9).

Fig. (3) to (7) show the closed loop frequency response (speed deviation) due to load (torque disturbance) and steam (extracted steam amount disturbance).

Considering Fig. (3) to (5) which show the effect of adopting P, PI, PD and PID controllers with different coefficients, it can be noted

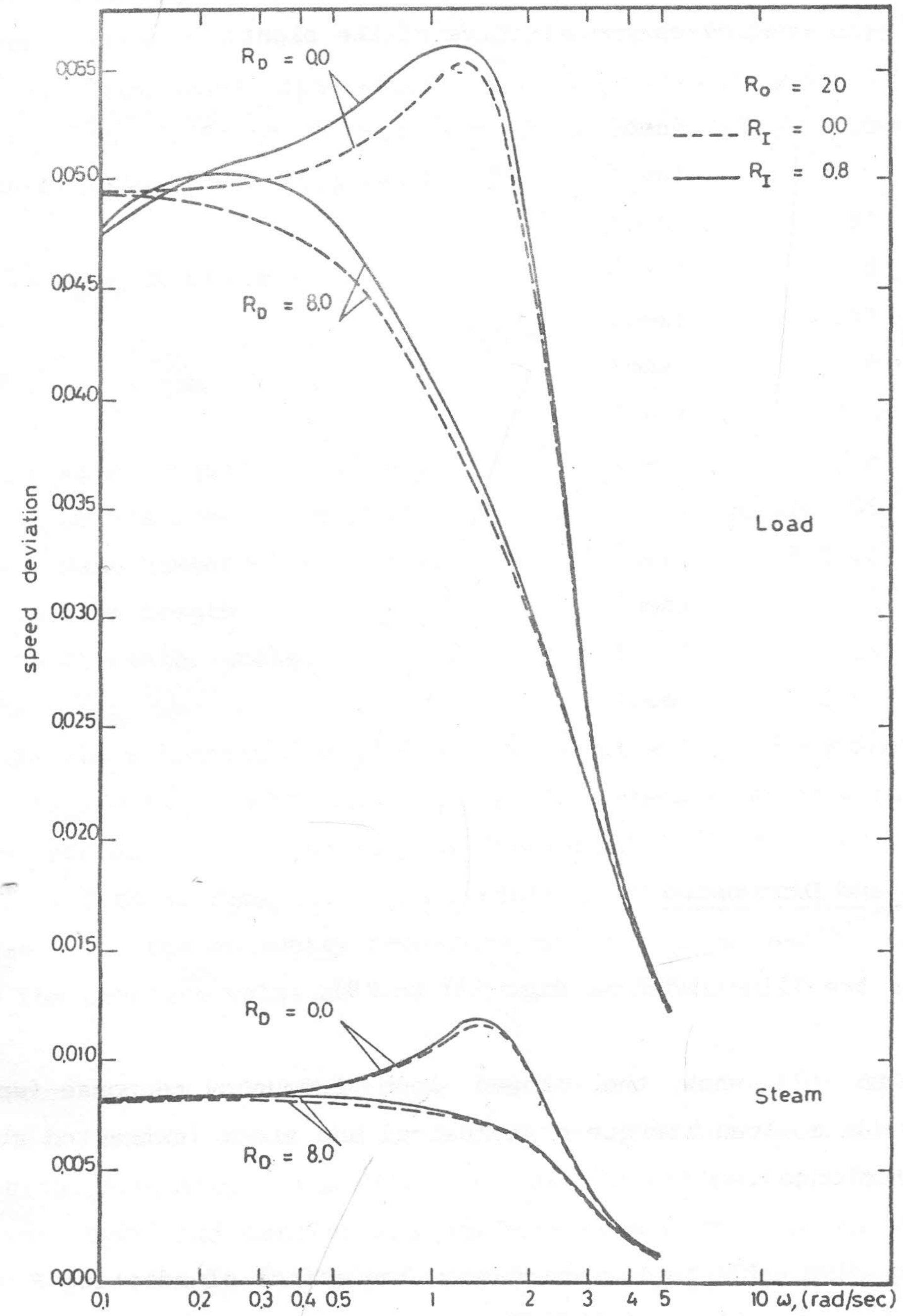

Fig. (3) Frequency response of the closed loop for load and bled steam amount disturnance for P, PD, PI, and PID speed controllers for $R_o = 20$

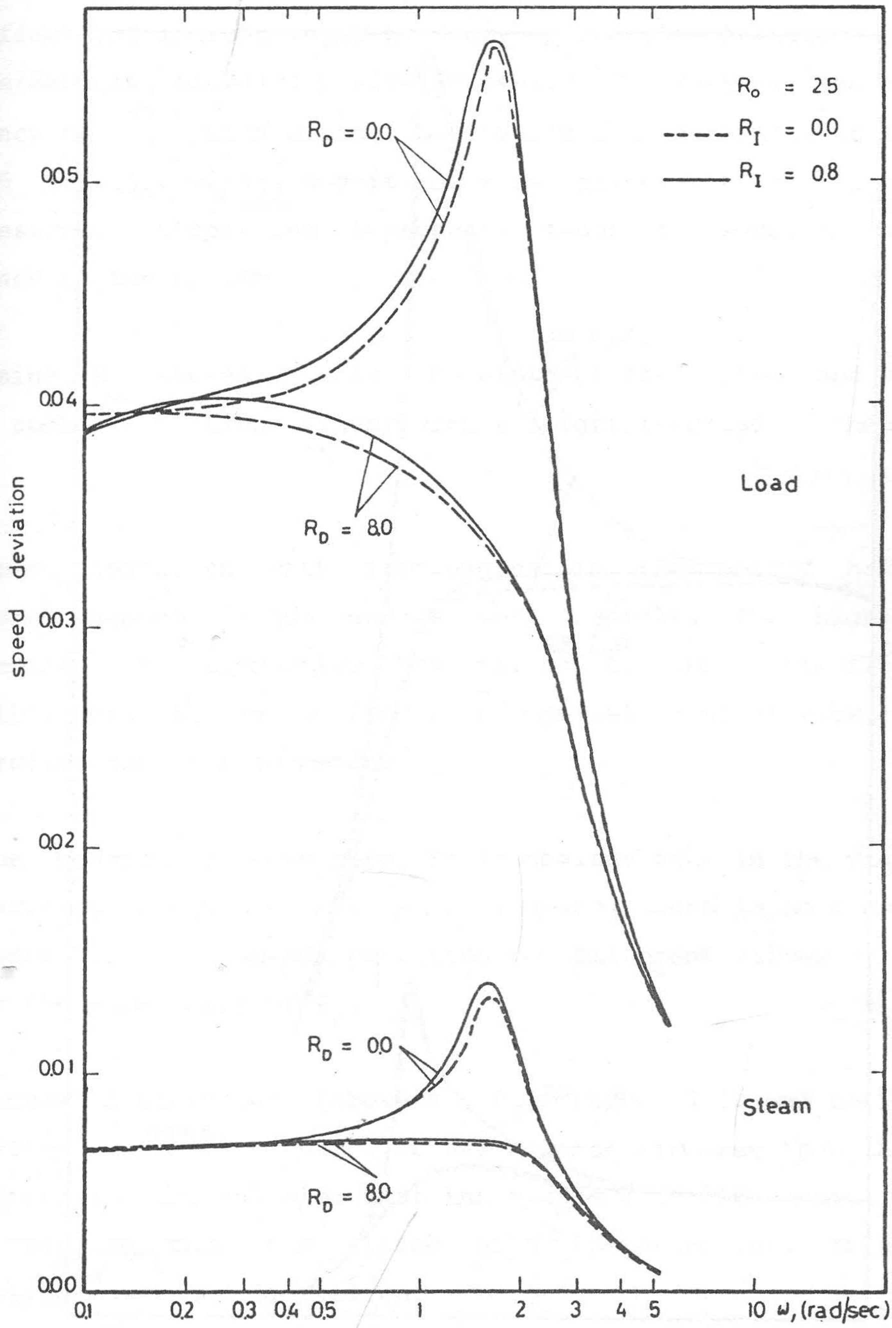

Fig. (4) Frequency response of the closed loop for load and bled steam amount disturbance for P, PD, PI and PID speed controllers for $R_o = 25$

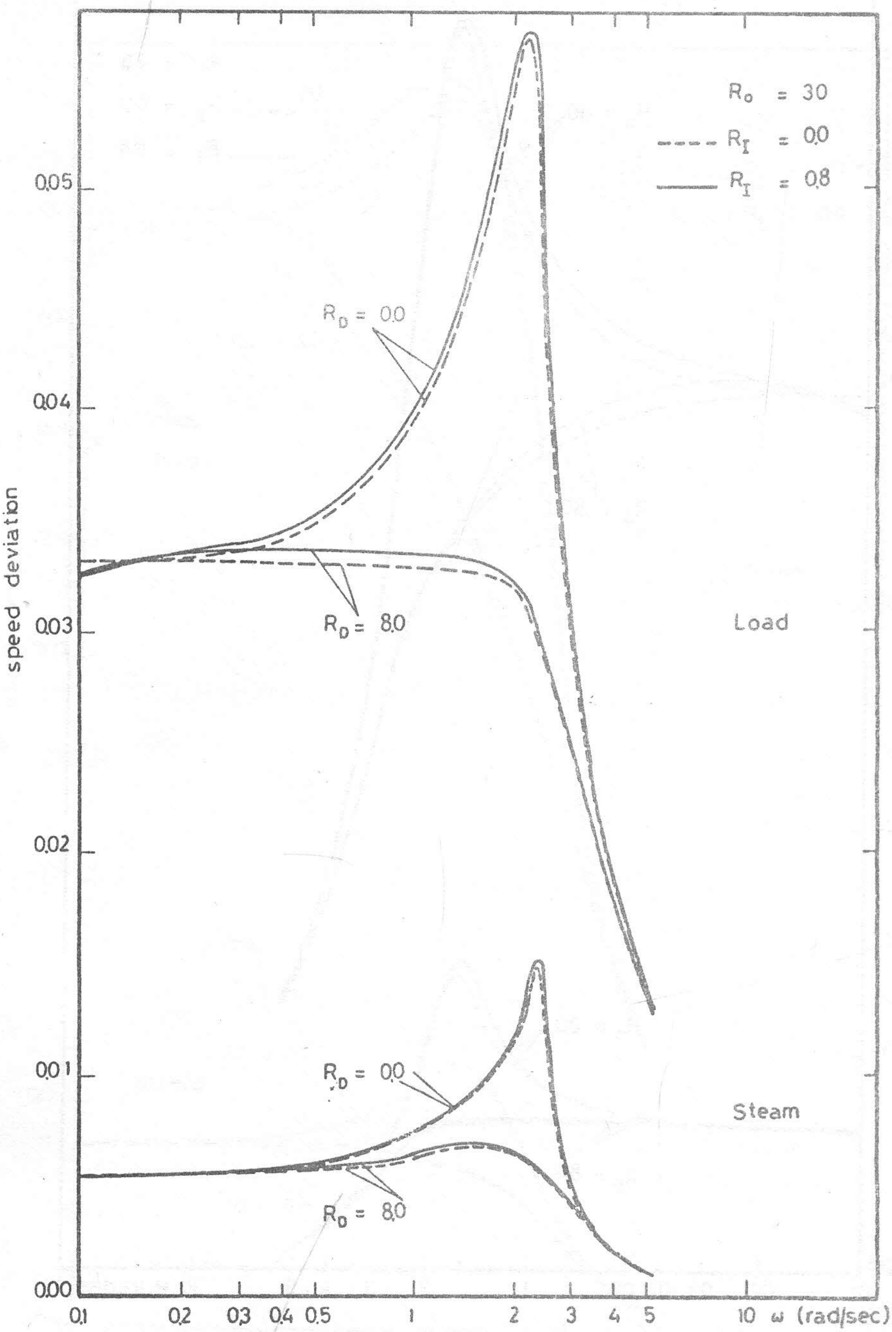

Fig. (5) Frequency response of the closed loop for load and bled steam amount disturbance for P, PD, PI and PID speed controllers for $R_o = 30$

that the effect of extracted steam amount disturbance is approximately one tenth to one fifth of the speed deviation due to load disturbance. The effect of introducing D-property on reducing the speed deviation is considerable, especially with increasing R_O. Further, the natural frequency of the system without D-Property lies generally in the range of 1.5 to 2.5 rad/sec, and it increases slightly with an increase in R_O. Besides, introducing D-property tends to reduce the natural frequency of the system.

Increasing R_O increased the band width of the system, and yielded lower speed deviation values with a slight increase in the resonant peak.

The speed deviation with frequencies in the second half of the considered second decade decays very rapidly. For high exciting frequencies, the properties and values of the controller are of negligible effect, hence from an economical point of view, a simple P-controller can be recommended.

For the considered case here, it is obvious that in the vicinity of the exciting frequency $\omega = 0.16$ rad/sec, there is no considerable difference in the speed deviation for different values of R_D and R_I, for the same value of R_O.

For higher frequencies (above $\omega = 0.16$ rad/sec) it was noticed that the system with PI controller has a worse response than the system with P-controller, and also that the system with PID-controller has a worse response than the system with PD-controller. This is an interesting point for discussion.

The error signal to the controller is the algebraic sum of the system output and the reference signal. In case of sinusoidal disturbances, linear control system's output in steady state can be represented as a series of sinusoidal signals obtained through Fourier expansion. This results in a sinusoidal response of the controller. Controller's frequency responses with different control properties can be consulted from fig. (1) [5,12]. The figure illustrates the frequency response of P, PD and PID controller both without and with time lags up to the third order. In our case all controllers are considered to have zero order time delays, i.e. ideal controllers.

When introducing the I-property in a controller, i.e. PI or PID-controller, it responds with a very large interfering action in magnitude at very low exciting frequencies, tending to infinity when the frequency vanishes. For ideal controller with I-property the interfering action decreases monotonically, reaching zero when ω approaches infinity. If time delays were incorporated in such a controller, the increase in ω reduces the interfering action up to a certain frequency, with a following increases, and eventually decreases again to zero.

An ideal P-controller has a constant finite frequency response, whereas for a P-controller with time delays it starts with finite interfering action, which decays (or increases and then decays) to zero as ω approaches infinity.

An ideal PD-controller starts from a finite value and tends to infinity as ω tends to infinity. For a $(PD)_1$- controller, the frequency response starts at a finite value at $\omega = 0$ and increases with increasing ω to a larger finite value when ω tends to infinity. For

PD-controllers with higher order of delay, the control action starts at a finite value and increases up to a certain frequency then decays to zero as ω tends to infinity.

From this frequency analysis of different controllers it is obvious that for very low exciting frequencies, incorporating the I-property produces the maximum interfering action followed by PD and P, especially for ideal controllers.

For frequencies beyond a certain limit (about 0.16 rad/sec in our case) the response with a PD-controller is to best followed by PID, P and finally PI-controllers.

Considering controllers incorporating time delays and after certain higher frequency, the preference of control properties may change according to the prementioned discussion. However, due to the rapid decay of the closed loop frequency response, this in of no or little significance.

Fig. (6) illustrates the closed loop frequency response for both torque and steam exciting disturbances with P and PI speed controllers. Fig. (7) shows the speed deviation with PD and PID speed controllers for different values of controller's coefficients. Both figures support the above mentioned discussion.

Fig. (8) and (9) represent the closed loop frequency response due to torque disturbance for different values of R_O, with R_I, R_D and ω was parameters. Fig. (8) shows the speed deviation with PI and PD speed controllers, while Fig. (9) is for P and PID speed controllers. It is further obvious that the exciting frequency has a significant

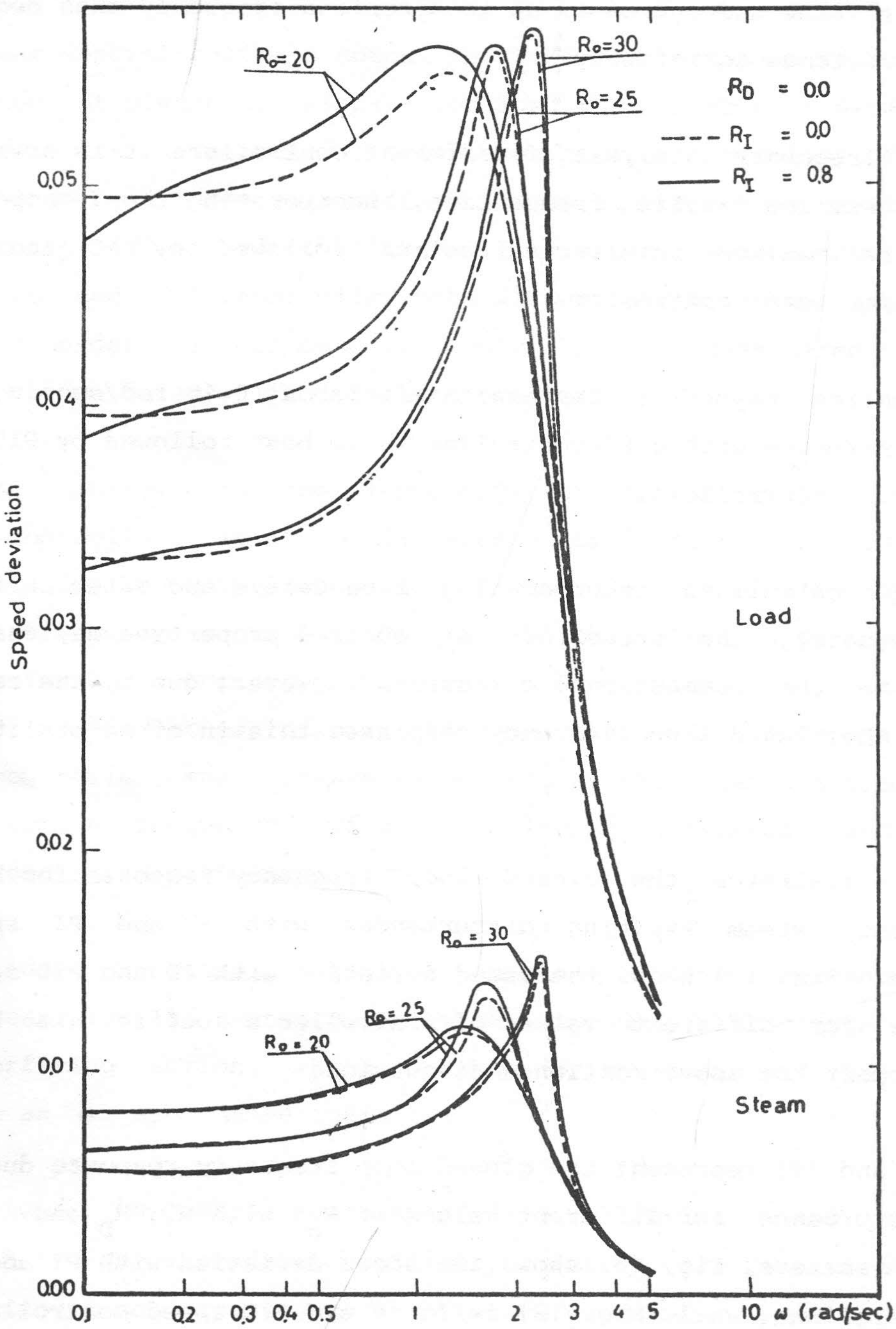

Fig. (6) Frequency response of the closed loop for load and bled steam amount disturbance for P and PI speed controllers with different coefficients

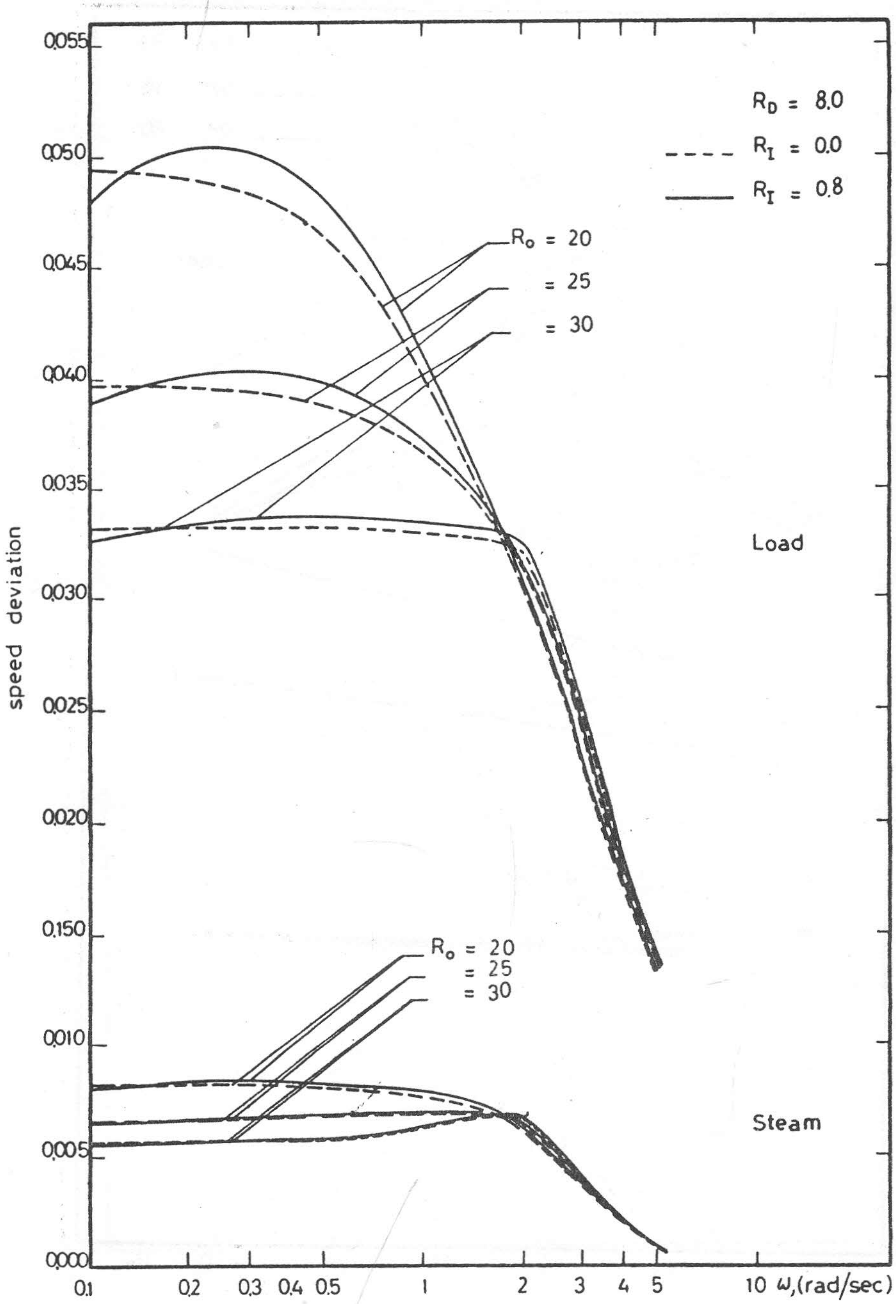

Fig. (7) Frequency response of the closed loop for load and bled steam amount disturbance for PD and PID speed controllers with different coefficients

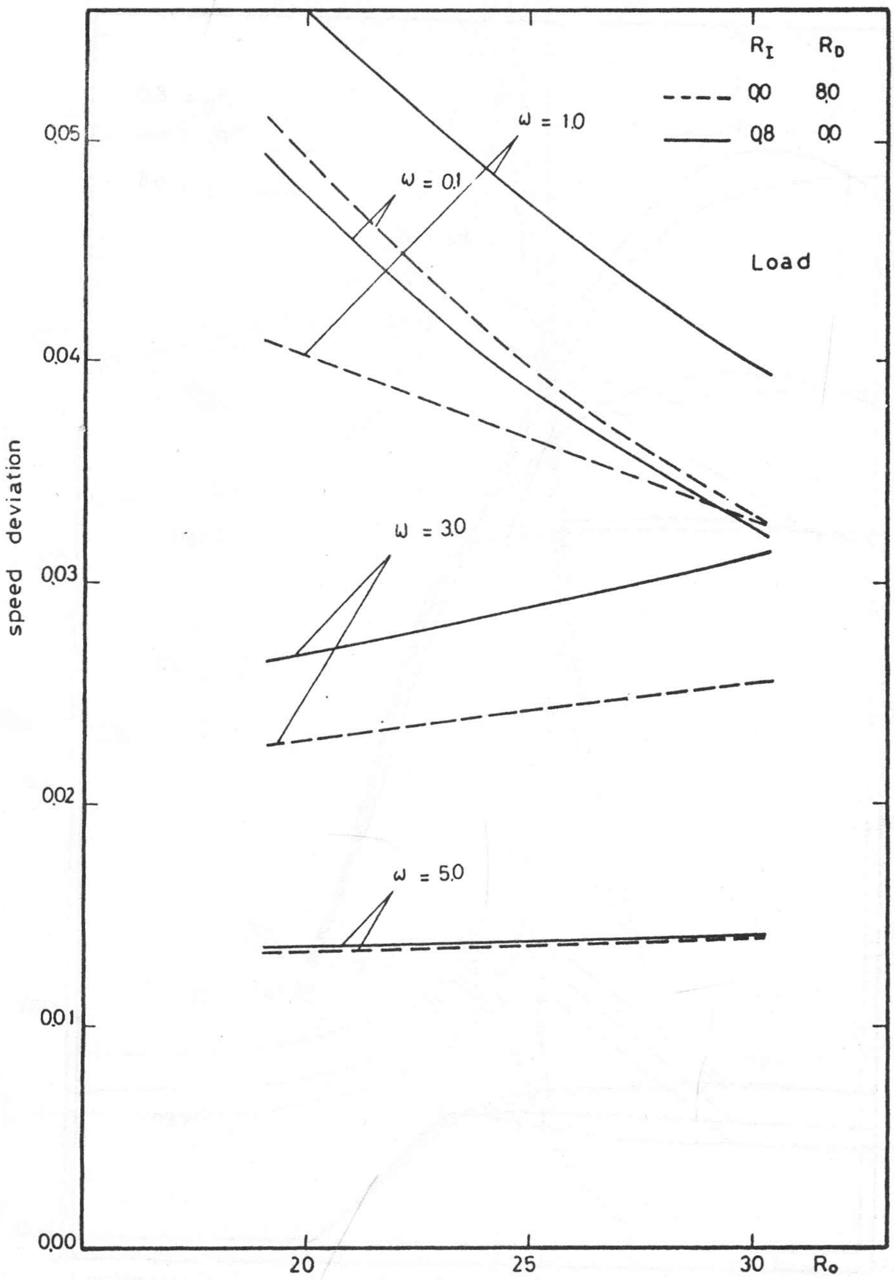

Fig. (8) Speed deviation due to load disturbance versus R_o for PI and PD speed controllers at different exciting frequencies

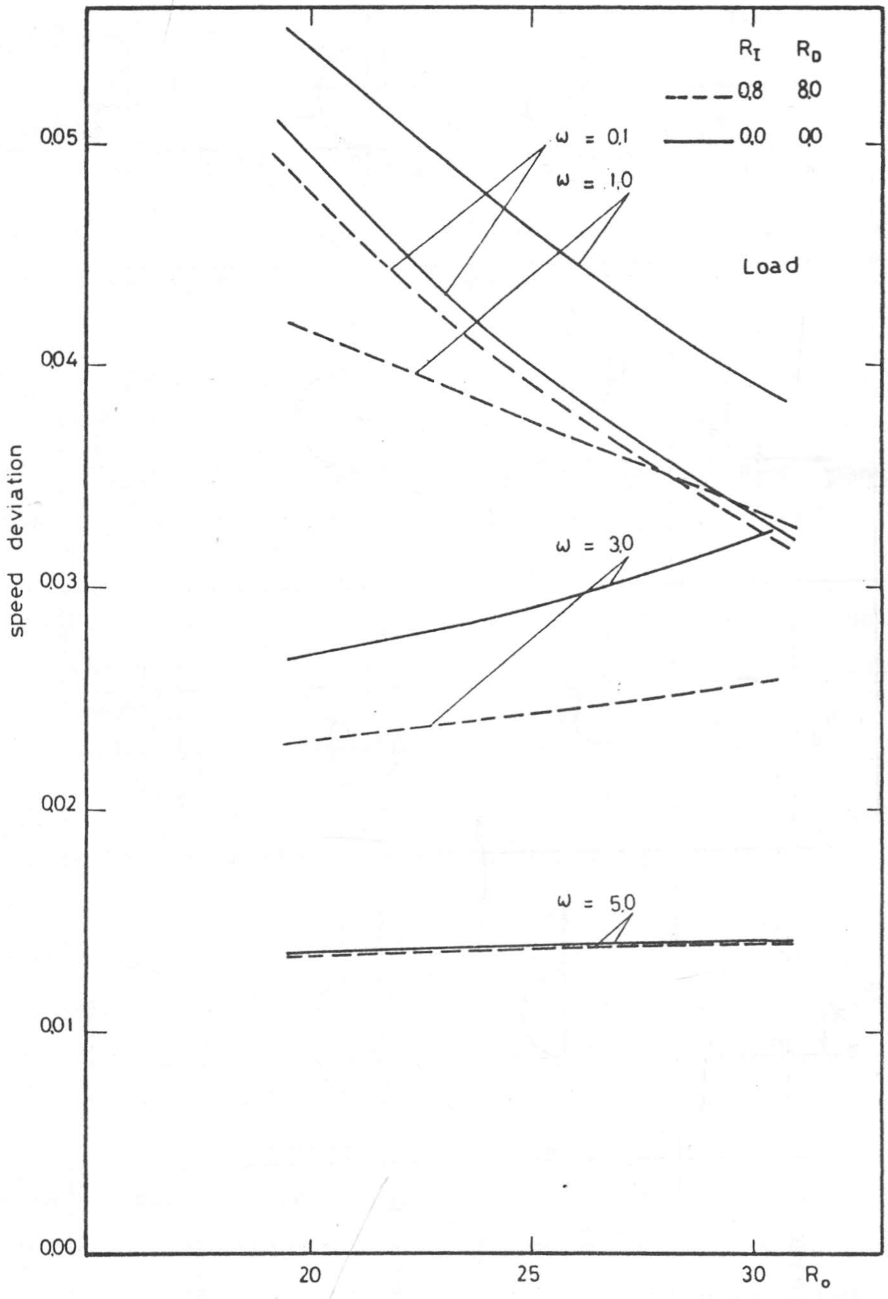

Fig. (9) Speed deviation due to load disturbance versus R_o for P, PID speed controllers at different exciting frequencies

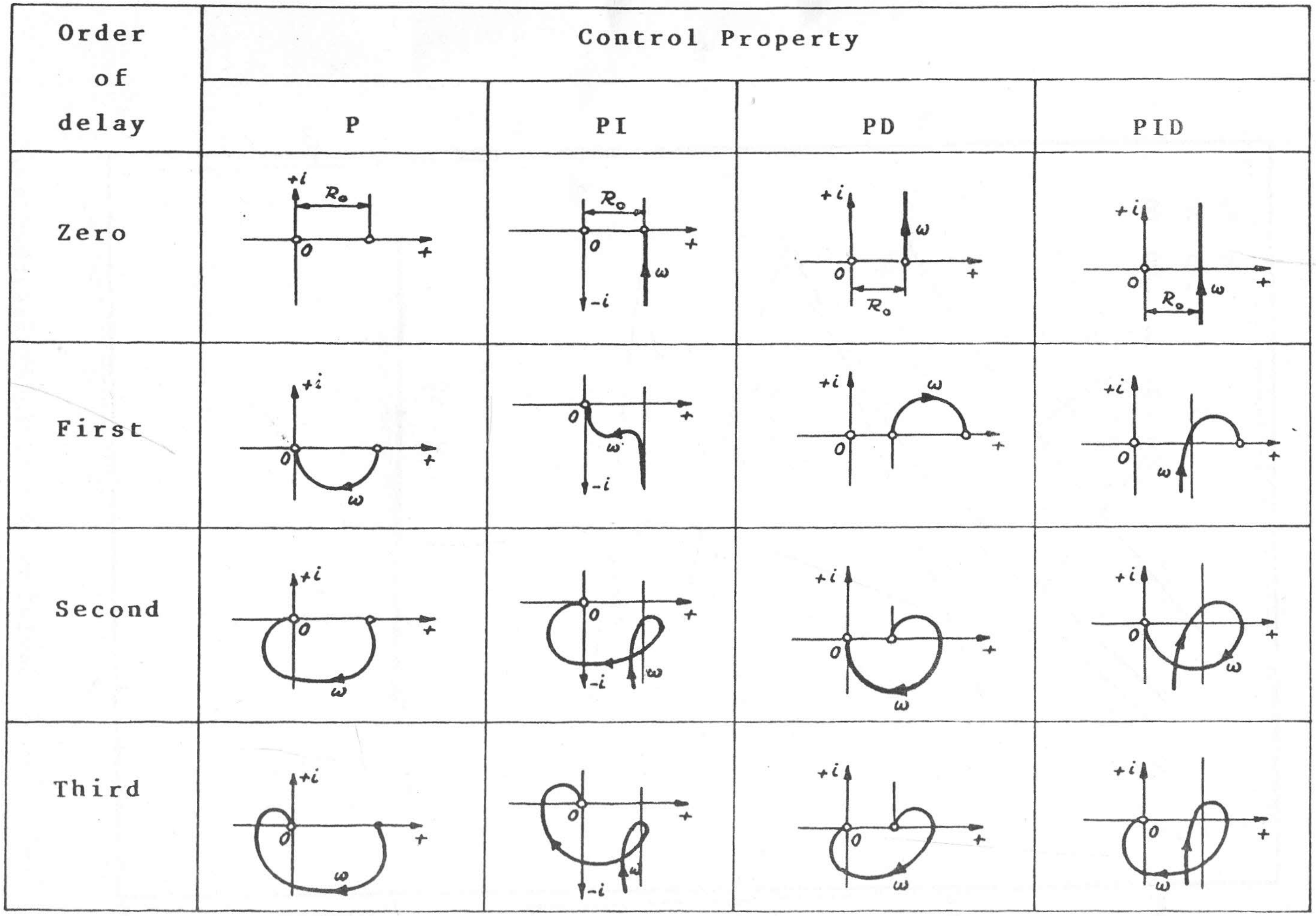

Fig. (10) Polar plots of controllers with different control properties and order of delay effect on the shape of the characteristic curves. Moreover, increasing R_o may decrease or increase speed deviation depending on the value of the exciting frequency.

5. Conclusion

A frequency domain study on simultaneously-interfering controllers adapted to a marine bled steam turbine was carried out.

It may be concluded that increasing R_o increases slightly the natural frequency and resonant peak, while generally reducing the speed deviation and increasing the control systems band width. Increasing R_D also reduces the speed deviation.

The exciting frequency has a dominant effect on the shape of the relation between speed deviation and system parameters. The natural frequency was found to be in the range of 1.5 to 2.5 rad/sec.

For rather high exciting frequencies, the speed deviation decays rapidly independent of the closed loop parameters.

Simultaneously-interfering controllers display much better behavior in terms of speed deviation compared to independently acting controllers as considered in [13]. For example, it was found that the maximum speed deviation due to load disturbance only was reduced by about 20%, while that for steam disturbance only the reduction amounted to about 66 %. Superimposing the resulting maximum speed deviation due to both disturbances considering the worst case of zero phase angle shift between both output responses, the obtained results for simultaneously -interfering controllers yield an overall reduction of 70 %. This reduction in resonant peaks can be interpreted as an improvement in the relative stability of the system. The reason why this considerable improvement of response takes place was discussed in section 2.

In what concerns the proper selection of controllers in both time and frequency domains, it is established that in the time domain, PID-controllers are known to give the best time domain characteristics followed by PD or PI and finally P-controllers. Introducing I-property is mainly to eliminate the steady state error. While in the time domain the proper selection of a controller depends on the dynamic behavior of the plant together with disturbance characteristics, in the frequency domain, however, the range of exciting frequency should in addition be given careful consideration.

In order to further decrease the speed deviation in control systems subjected to periodic or sinusoidal disturbances, consideration should be paid to controllers having asymetric D-property, i.e. having faster response in case of speed increase than in case of speed decrease. Similarly asymetric hydraulic servomotors may be adopted for improving systems response [14].

6. References

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