RELATIVE STABILITY AND FREQUENCY RESPONSE OF MARINE STEAM TURBINE

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ABSTRACT

The relative stability measures and frequency response of a marine bled steam turbine with non-interacting controllers was analyzed. An investigation of the behavior of the multi-variable control system with ideal P-pressure controller and ideal P, PI, PD or PID - speed governors was carried out. A recommendation for selecting controllers for control plants subjected to periodical disturbances is proposed.

1.INTRODUCTION

By "frequency response", is meant the steady-state response of a system to a sinusoidal input. In the frequency response methods, the most conventional methods available to control engineers for the analysis and design of control systems, the frequency of the input signal may be varied over a certain range and the resulting frequency response can be studied. The frequency response methods enable us to investigate both the absolute and relative stabilities of linear closed - loop systems from a knowledge of their open-loop frequency response characteristics [1].

This is one advantage of the frequency response approach. Another advantage of this approach is that frequency response tests are, in general, simple and can be determined experimentally [2]. In addition, the frequency response approach has the advantages that a system may be designed so that the effects of undesirable noise are negligible and that such analysis and design can be extended to certain non-linear control system [3].

The fluctuation of the marine steam power plant is affected by the load disturbance on the propeller due to sea waves as well as ship motion [4].

There are some difficulties associated with obtaining a mathematical modelling for the load disturbance on the propeller. Although the different approaches described by the naval architect represent a logical step towards the evaluation of the disturbance caused by ship motion on the propeller, they have their own limitations and there are also difficulties associated with their application [5]. Furthermore, as yet there is no real proof that the system of ship motion equations represents a good model of load disturbance on the propeller of ship.

Due to the previous remarks, the most widely used singularity functions for the study of control systems, namely the sinusoidal function with a specific frequency is going to be assumed as load disturbance. This frequency depends on wave frequency, ship's speed, wave number and encounter angle [6,7]. This sinusoidal disturbance represents an external load on the propeller torque.

The regulating system of bled steam pressure and turbine's speed is a multi-variable system with two input signals, namely the propeller load due to sea waves and the amount of extracted steam.

The pressure of extracted steam is also assumed to be of a sinusoidal nature with the same frequency as the load [7].

Both time domain [8] and frequency domain analysis [7] of a marine bled steam turbine with interacting controllers were studied through a comparison for the dynamic behavior of the closed loop control system with different controller's properties.

Time and frequency domain methods give completely complementary pictures of many important problems, e.g. system identification [9]. Frequency domain methods seemed to dominate theory and practice of control systems in control engineering applications. Now, the interest in time domain methods has increased, and the literature on control engineering is much dominated by time domain methods. Needless to say, frequency domain methods are of course still in successful use in practical applications.

The purpose of the present paper together with the study of time domain analysis presented in [10] is to bridge over the gap between the two domains. In other words an analysis of the dynamic behavior of controllers in frequency domain for the regulating system of bled steam pressure and turbine's speed with non-interacting controllers has been carried out.

2. MATHEMATICAL MODELLING OF THE CONTROL SYSTEM

Figure (1) shows a regulating system for the extracted steam pressure and turbine's speed with non-interacting controllers. P_0 pressure controller and $(PID)_0$ speed controller are connected separately each with a hydraulic servo-motor, which has P_1 control property. Both hydraulic servo-motors actuate the steam valves in each portion of the turbine.

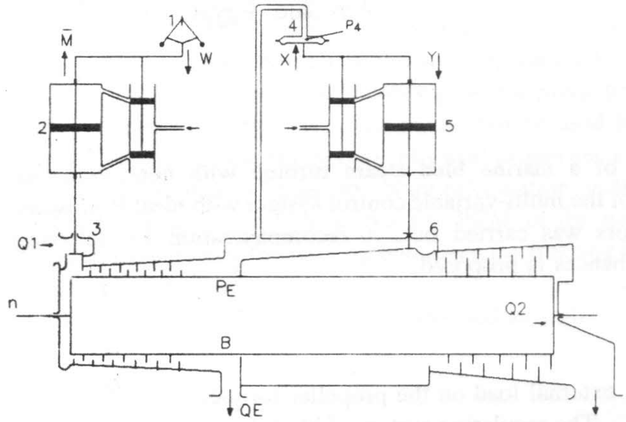


Figure 1.

Although the block diagram is useful for pictorial representation of control systems, the block diagram reduction process becomes quite time-consuming for complicated systems. An alternative approach for finding the relationship among the system variables of a complicated control system is the signal flow graph approach, developed by S.J. Mason.

Nodes of signal flow graph represent the variables and the transmittance are the transfer functions between two nodes.

The signal flow graph of the system is shown in Figure (2), representing the mathematical simulation. The transfer function of each control element is indicated on each branch.

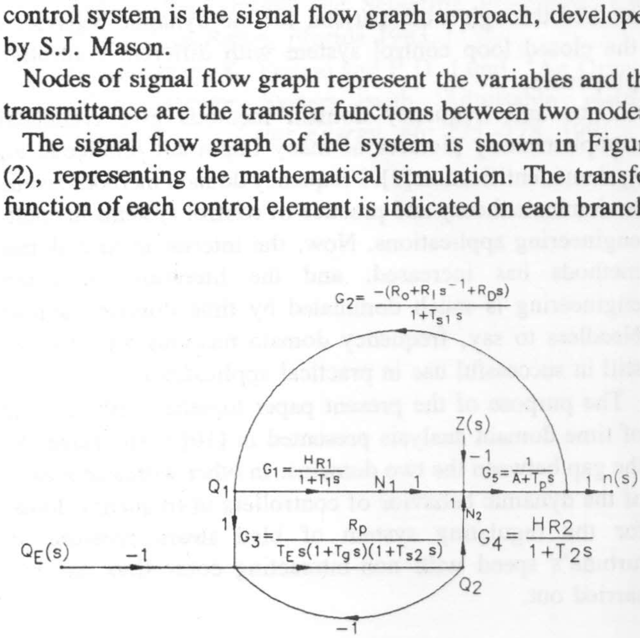


Figure 2.

The nodes which have only outgoing branches are the input node corresponding to an independent variable, namely the relative steam extraction $Q_E(S)$ and relative load disturbance Z(S), whereas, the node which has only incoming branches is the output node corresponding to a dependent variable, namely the turbine's relative speed deviation n(S).

According to Mason's gain formula [3], the overall transfer function is given by:

$$\text{T.F.} = \frac{1}{\Delta}\sum_{k} P_k \, \Delta_k$$

where

P_k = path gain or transmittance of k-th forward path,

Δ = determinant of graph.

$$= 1 - \sum_{a} L_a + \sum_{b,c} L_b L_c - \sum_{d,e,f} L_d L_e L_f + \ldots\ldots,$$

$\Sigma_a L_a$ = sum of all different loop gains,

$\Sigma_{b,c} L_b L_c$ = sum of gain products of all possible combinations of two non-touching loops,

$\Sigma_{d,e,f} L_d L_e L_f$ = sum of gain products of all possible combinations of three non-touching loops,

Δ_k = factor of the k-th forward path determinant of the graph with the loops touching the k-th forward path removed,

According to Mason's gain the transfer function for the system shown in Figure (2) is:

$$n(s) = \frac{1}{\Delta}\left[\sum_{k} P_k \Delta_k \quad \sum_{i} P_i \Delta_i\right]\begin{bmatrix} Q_E(s) \\ Z(s) \end{bmatrix}$$

$$L_1 = -G_3,\; L_2 = G_1\, G_2\, G_5,\; L_3 = G_2\, G_3\, G_4\, G_5$$

$$\Delta = 1 - (L_1 + L_2 + L_3) + L_1 L_2$$

Consider the relative steam extraction $Q_E(s)$ only as input variable, k = 1, and

$$P_1 = -\,G_3\, G_4\, G_5$$

$$\Delta_1 = 1$$

i.e.

$$\sum_{k} P_k \Delta_k = -G_3 G_4 G_5$$

For the relative load disturbance Z(s) only as input variable, i = 1, and

$$P_1 = -\,G_5,$$

$$\Delta_1 = 1 - L_1 = 1 + G_3$$

i.e. $$\sum_i P_i \Delta_i = -G_5 (1 + G_3)$$

3. NOMENCLATURE NUMERICAL RESULTS AND DISCUSSIONS

In order to investigate the frequency domain, sinusoidal disturbances for both the load and the extracted steam amount disturbances may be assumed.

The frequency responses of the considerable regulating system are directly affected by the control system's time constants and

parameters, which are defined and given in [10]. It can be summarized as follows:

H_{R1}	$= 1/3$	(-)
H_{R2}	$= 2/3$	(-)
R_0	$= 20$ and 25	(-)
R_I	$= 0, 0.5$ and 0.8	s^{-1}
R_D	$= 0{,}5$ and 8	s
R_P	$= 20$	(-)
T_1	$= 0.2$	s
T_2	$= 0.0$	(-)
T_E	$= 10$	s
T_g	$= 0.0$	(-)
T_{s1}	$= 0.1$ and 0.3	s
T_{s2}	$= 0.2$	s
T_p	$= 16$ and 20	s
A	$= 0.0$ and 0.2	(-)

The numerical solutions were carried out using FORTRAN programs listed in Ref. [11], executed at the computer center of the faculty of Engineering, Alexandria University.

The frequency response analysis was carried out for both the open and closed loops and displayed in polar and Bode plots. Measures of the relative stability, namely the gain and the phase margins were computed from the open loop frequency response in polar and Bode plots, and Nichols chart, which are indicated in Figures (3) to (8).

Figure (3) and (4) illustrate the concept of the Nyquist stability criterion or the absolute stability as well as the relative stability indicators, namely the gain and the phase margins.

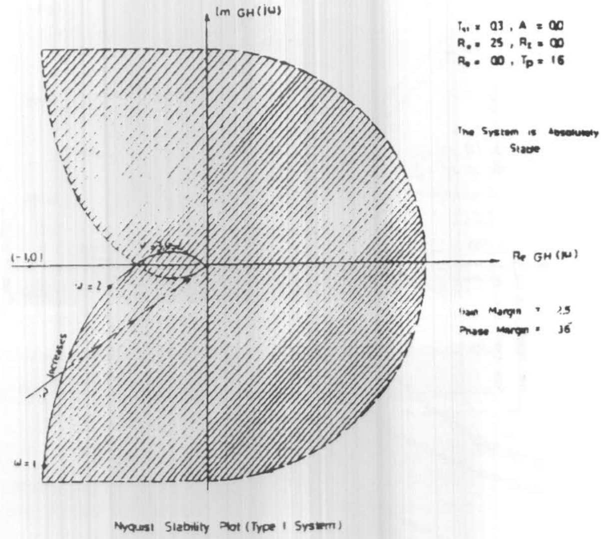


Figure 3.

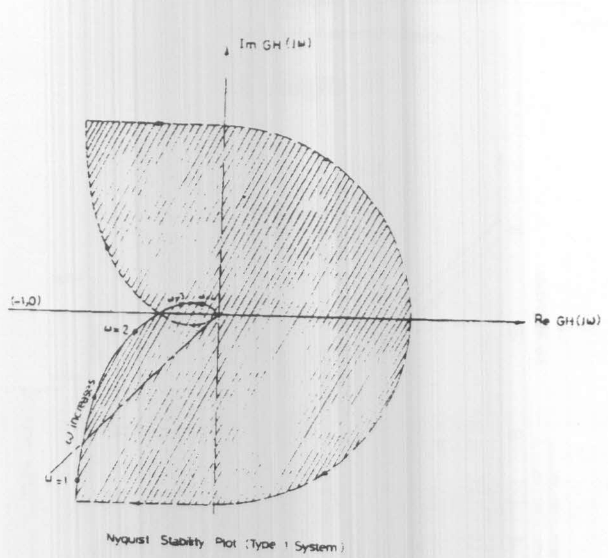
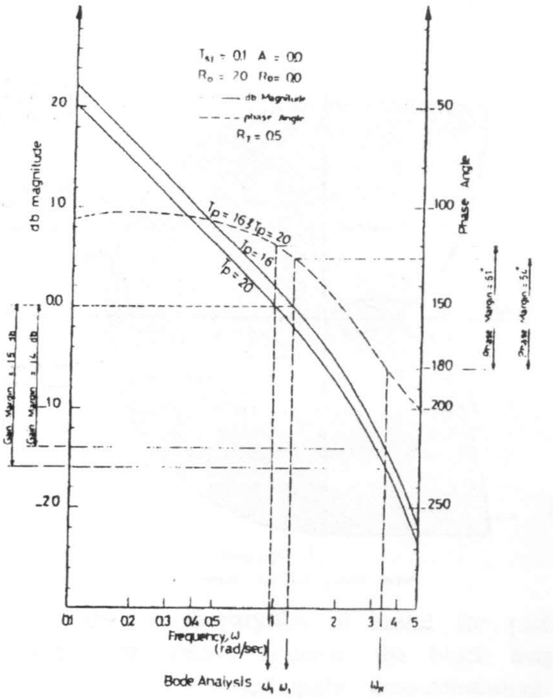


Figure 4.

Figure (5) and (6) illustrate the Bode analysis of the open loop control system under discussion.

Figure (7) and (8) represent the open loop frequency response plotted on a Nichols chart.

The closed loop frequency responses due to either sinusoidal load or sinusoidal extracted steam amount disturbances for combinations of the plants, regulating systems dynamics and the disturbances frequencies are plotted in Figures from (9) to (18) inclusive. All control parameters are illustrated on the drawn curves.



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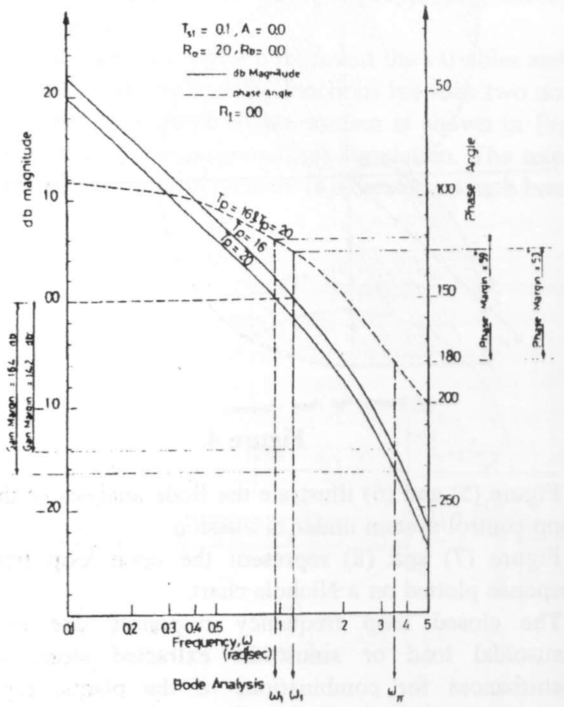

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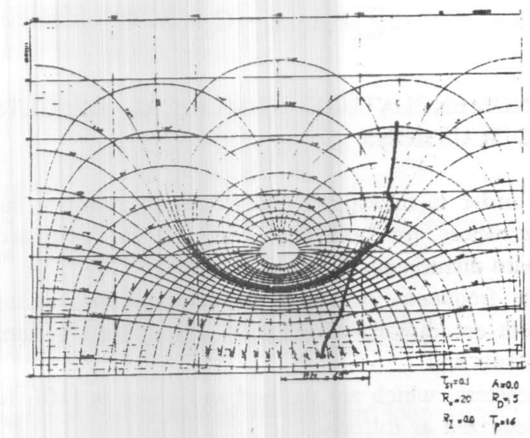

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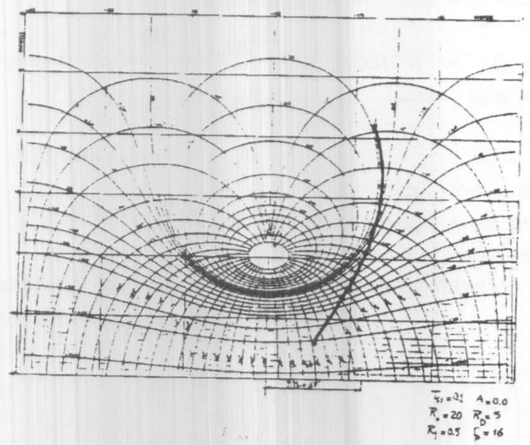

Figure 8.

In order to compute the percentage speed deviation due to both disturbances occurring simultaneously, the superposition principle applied to multi-input linear control systems can be adopted. In this case for sinusoidal inputs, the phase differences for the outputs due to each sinusoidal input should be taken into account by applying the vector algebra rules.

According to Nyquist stability criterion, the control system under consideration is absolutely stable with P-controller as shown in Figure (3) and (4). Increasing the time constant of the turbine's rotor T_p from 16 to 20 (s) increases the phase margin from 36° to 44.5° and increases the gain margin from 2.5 to 3.125.

From Figure (5) and (6), using PI and P-speed controllers respectively, the Bode analysis displays that increasing T_p from 16 to 20 (s) with PI-speed controller, the phase angle plot remains unchanged with the same phase crossover frequency ω_1. On the contrary, the db magnitude plot is shifted downwards by about 2 db. Furthermore, the gain margin is increased from about 14 to 16 db, whereas, the phase margin is also increased from 54° to 61°.

If compared to PI-speed controller, the adoption of only P-controller reveals the slight decrease of both gain and phase margins of the control system, however such variations are not significant.

From Figure (7) and (8), the Nichols chart illustrate a comparison between PD and PID-speed controllers on the relative stability measures for a specified control system. Choosing PID instead of PD-speed controller will improve the phase margin from 65° to 69°.

In what concerns the analysis of the automatic feedback control loop in the frequency domain, Figure (9), (10) and (11) illustrate the effect of adopting P, PD, PI or PID-speed controllers for different values of derivative and integral property coefficients of the speed governor. It is to be noted that the closed loop frequency response due to load disturbance is always higher than that for the extracted steam amount. Also, the resonant frequency lies in the proximity of 1 rad/s for both load and bled steam disturbances. It is concluded that, varying R_I has no significant effect on both resonant frequency and frequency response of the system. Furthermore, introducing D-property either to P or PI-speed controller leads to reducing the frequency response of speed deviation and consequently reducing both the resonant and frequency peak.

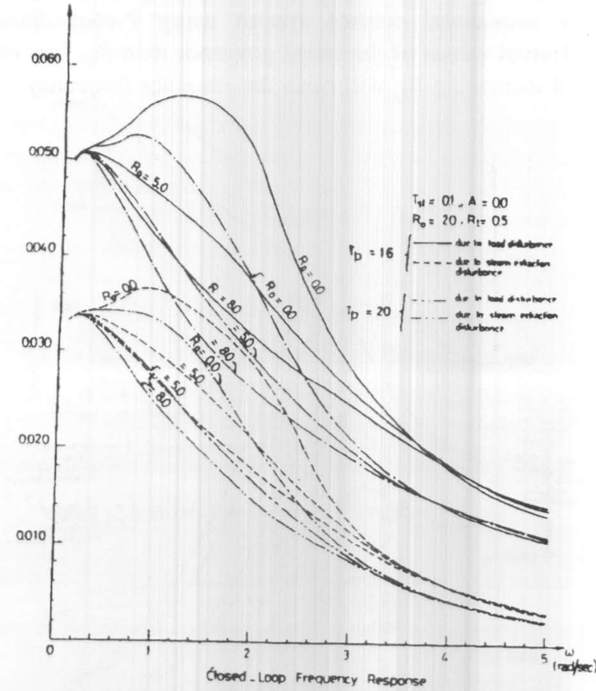

Figure 10.

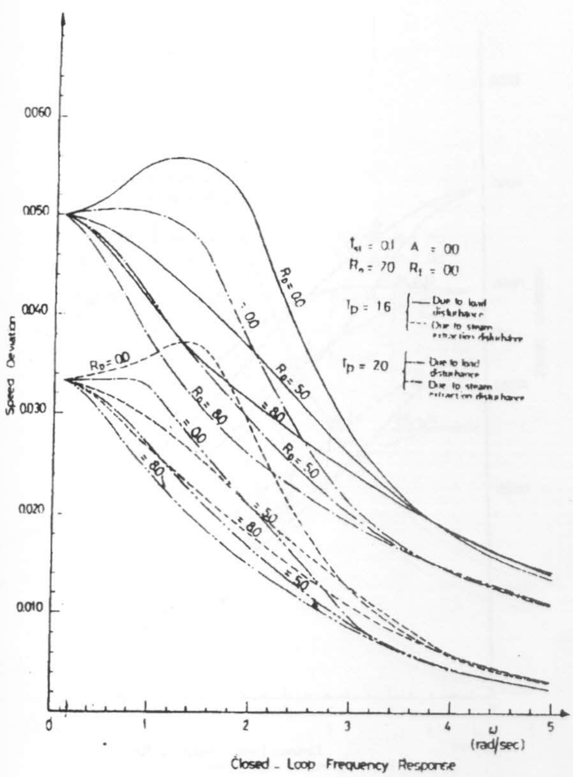

Figure 9.

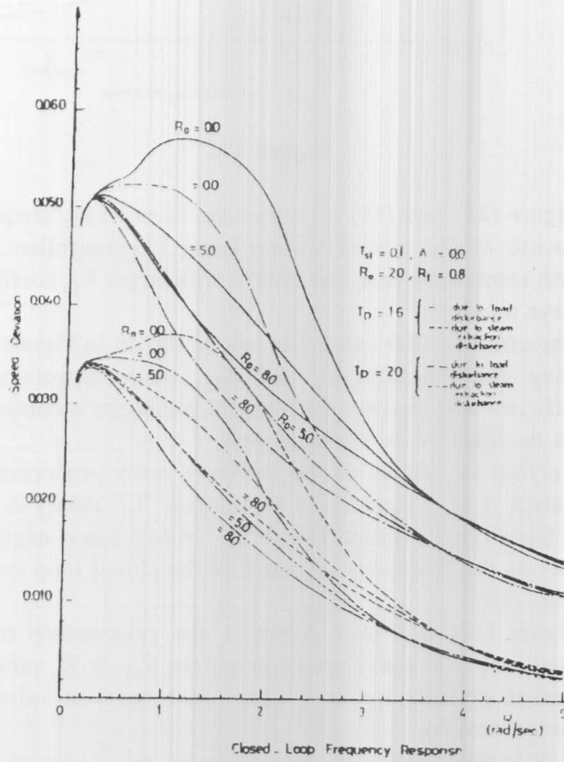

Figure 11.

Figure (12) shows the closed loop frequency response for the considered control system using P-controllers with different values of the speed governor gain R_0. It is obvious that increasing R_0 will raise the resonant frequency.

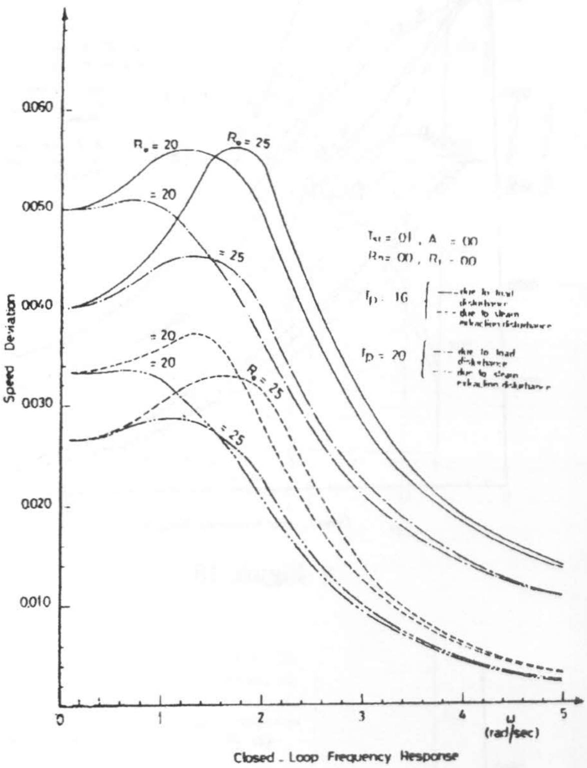

Figure 12.

Figure (13) and (14) represent the closed loop frequency response of the control system using PD-controllers. It is worth mentioning that, the system with larger R_D coefficient decays rapidly.

The effect of PID controller can be shown in Figure (15). Fixing the constants R_I and R_D, while increasing R_0 coefficient will increase the resonant frequency accompanied by a decrease of the resonant peak.

Varying the values of the turbine's rotor proportionality constant A = 0 (specifying an integral "I_0" rotor) to A = 0.2 (specifying a proportional "P_1" rotor) has a negligible effect on the frequency response of the closed loop system, Figure (16).

Figure (17) and (18) represent the relationship of the closed loop frequency response versus R_D or R_I values at different exciting frequency lines and constant values of other parameters.

It is evident that the magnitude of speed deviation is mainly dependent on the value of the exciting frequency irrespective of the value of R_D or R_I.

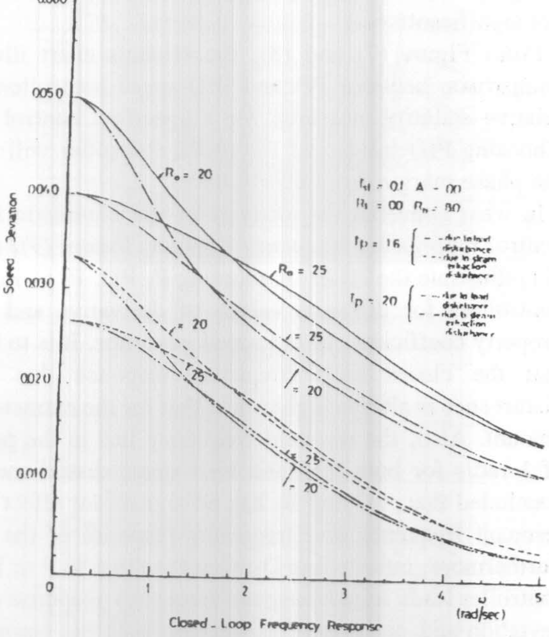

Figure 13.

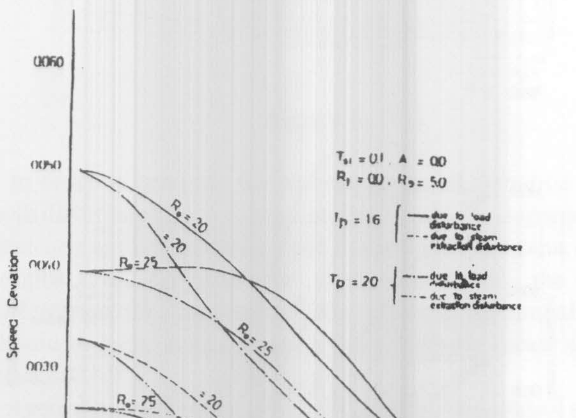

Figure 14.

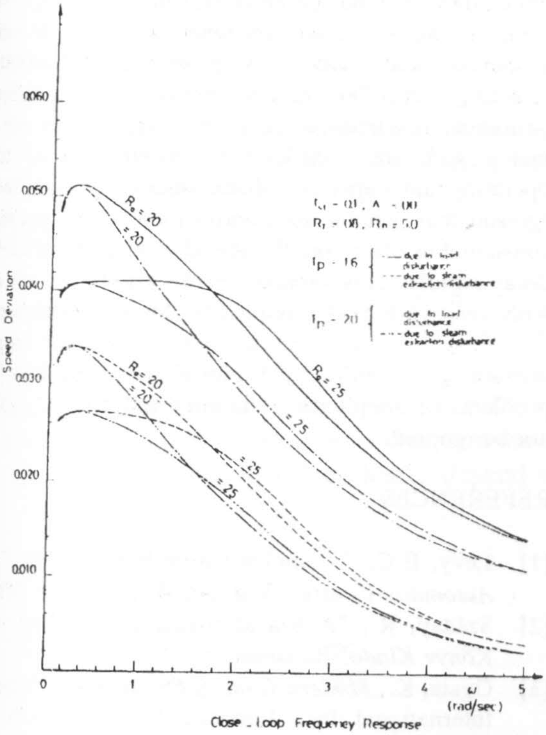

Figure 15.

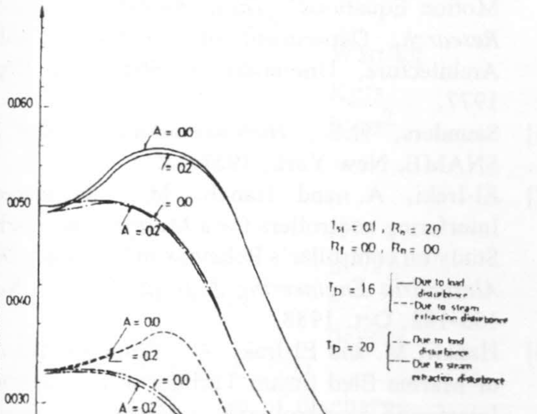

Figure 16.

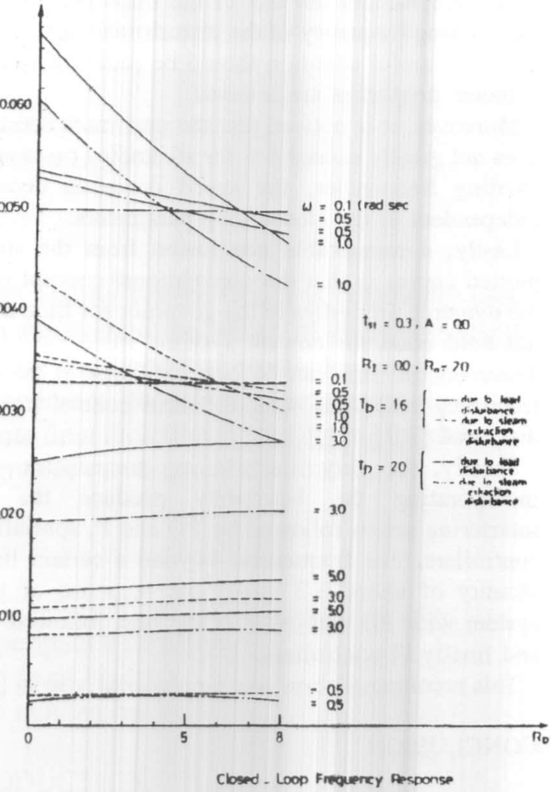

Figure 17.

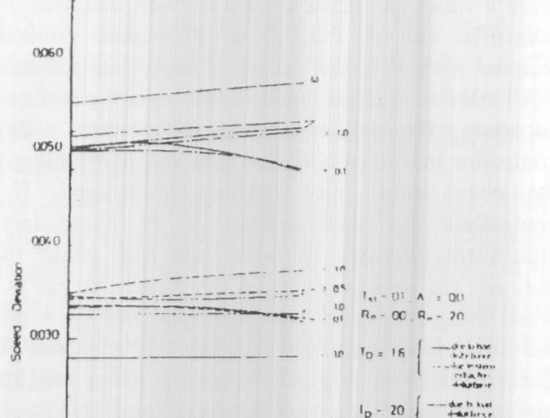

Figure 18.

This emphasizes the fact of the important role played by the exciting frequency of the disturbance on the control loop and that careful attention should be paid when the governor dynamic properties are selected.

Moreover, it is noticed that the maximum speed deviation does not greatly exceed 6% for all studied cases and for high exciting frequencies, the speed deviation decays rapidly independent of the closed loop parameters.

Lastly, a remarkable conclusion from the study of the plotted curves is that the conventional concept of selecting the dynamic properties of the governor for time domain does not hold good for control systems subjected to periodical disturbances. This can be best explained if the controllers frequency responses with different control properties are discussed [2,7].

Shortly, for very low exciting frequencies, controllers incorporating the I-property produce the maximum interfering action followed by PD and P, specially for ideal controllers. For frequencies beyond a certain limit (in the vicinity of about 0.2 rad/s) the response of the control system with PD-controller is the best followed by PID, P and finally PI-controllers.

This recommendation was emphasized also in [7].

CONCLUSION

Relative stability indicators as well as the frequency response of the automatic closed loop control system for compound regulation of extracted steam pressure and turbine's speed were discussed. A wide variety of the control plant's time and other constants was scanned. P-pressure controller and P, PD, PI or PID-speed governors were adapted with different values of controller parameters.

All control systems under study insure absolute stability. Increasing the turbine's shaft time constant with PI speed controller increases both the gain and phase margins, while the phase angle plot remains unchanged. If only P-controllers are used instead of PI controllers a non-significant decrease in both gain and phase margins is noticed. An improvement of the phase margin is observed when using PID instead of PD controllers. In what concerns the closed loop frequency response, the response due to load disturbance is much greater than that of the bled steam. The maximum speed deviation does not considerably exceed 6%. In general the resonant frequency is located in the neighborhood of 1 rad/s. A variation of R_I shows insignificant effect in both resonant frequency and speed deviation. The D-property if introduced to either P or PI controller reduces the speed deviation and decreases both the resonant and frequency peak. For P, PI, PD or PID-controllers, increasing the controller's gain R_o will increase the resonant frequency and decrease the resonant peak. The turbine's rotor proportionality constant (A) has a negligible effect on the frequency response of the closed loop. It is emphasized that the operating exciting frequency ω should be carefully taken into account when selecting the controller's properties and values. A proper recommendation for selecting controllers to suit control systems subjected to periodical disturbances is given, which does not exactly match with the well-known concept applied to plants operating under aperiodic disturbances. For high frequencies (greater than 5 rad/s) the control loop parameters have non-considerable effect on the speed deviation which tends to decay rapidly. This research reveals also the slower response with relatively poor frequency domain characteristics if compared to such multi-variable control loops with interacting controller's [7]. However, as stated in [10], the problems of simplicity, reliability and relative cost should not be ignored.

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