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ON THE TRANSIENT BEHAVIOUR OF CONTROLLED PROPULSION PLANTS

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Abstract

This paper makes known the results of the investigations carried out into the transient behaviour of closed control loops of a ship propulsion plant. Transient characteristics of various speed governors as well as their applicability are compared. Comparison is made on the basis of transient characteristics gained by computer simulation, and represented in figures.

Keywords: transient characteristics, speed governors, propulsion plant.

1. Introduction

A series of computer simulations has been carried out at our department, in order to gain information on the transient behaviour of a ship's propulsion plant in waves.

Closed control loops, containing the following speed governors, have been compared:

- a direct acting mechanical one,
- a PI-type hydraulic one, equipped with a compensating vanishing feed-back.
- two versions of a two-pulse electronic speed governor, co-operating with a constant-pressure fuel injection system.

The following characteristics as well as decisive phenomena of the closed control loops have been calculated and compared: momentary speed variation, time constants, frequency response, stability, optimum setting of compensation, influence of the fuel injection system, the effect on the system dynamics of the deteriorating conditions of the fuel pumps.

2. Models of the System Components

2.1. Basic Attributes

The applied mathematical models of the engine and of the speed governors are linearized and quasi-stationary. The matching point. i. e. the basis of linearization is defined by 67% fuelling and 90% engine speed, with reference to the nominal values. In order to gain better-conditioned models, relative variables are applied, instead of physical characteristics. Having been divided by the matching point value, the change, referring to the matching point, of a variable yields its relative value.

The investigations have been carried out by means of state-space models written in MATLAB supported by its own 'Control Toolbox'.

The excitation function has been considered sinusoidal, deterministic function of two variables. The wave pattern has been presumed to be regular. The ship's advancing has been considered perpendicular to the crests. In most cases the angular frequency and the relative amplitude of the excitation have been $\omega_g = 1.08$ [rad/s] and $\alpha_e = 0.325$, respectively.

In the course of our investigations friction forces on the moving parts of the speed governors as well as that of the fuel injection system have been assumed to be proportional to the relative velocity of the moving parts, applying a wide range of ϑ coefficient of proportionality. This approximation is based upon the vibrations of the lubricated moving parts.

The model accuracy in steady state condition, on the base of the test bench diagrams of the engine, proved to be sufficient for system dynamics application, while its correct transient properties have been justified by sea trials.

2.2. Models of a Supercharged Marine Diesel Engine

A four-stroke, medium-speed, supercharged marine diesel engine 6NVD48A-2U type of SKL has been chosen, as the physical basis of the mathematical model.

Two different state-space models of the engine, for different applications, have been set up, applying two different methods [1]. The applied setting-up methods of the state-space models are founded on the transfer function form of the mathematical model of a supercharged diesel engine developed by KRUTOV [2]. The layout and the block diagrams in transfer function representation of the model are shown in *Fig. 1* and *2*, respectively. According to these figures, κ relative displacement of the fuel rack and α_e relative load are the input signals, while φ relative speed of the engine is the output signal of the model. Most of the system components are of first-order proportional type.

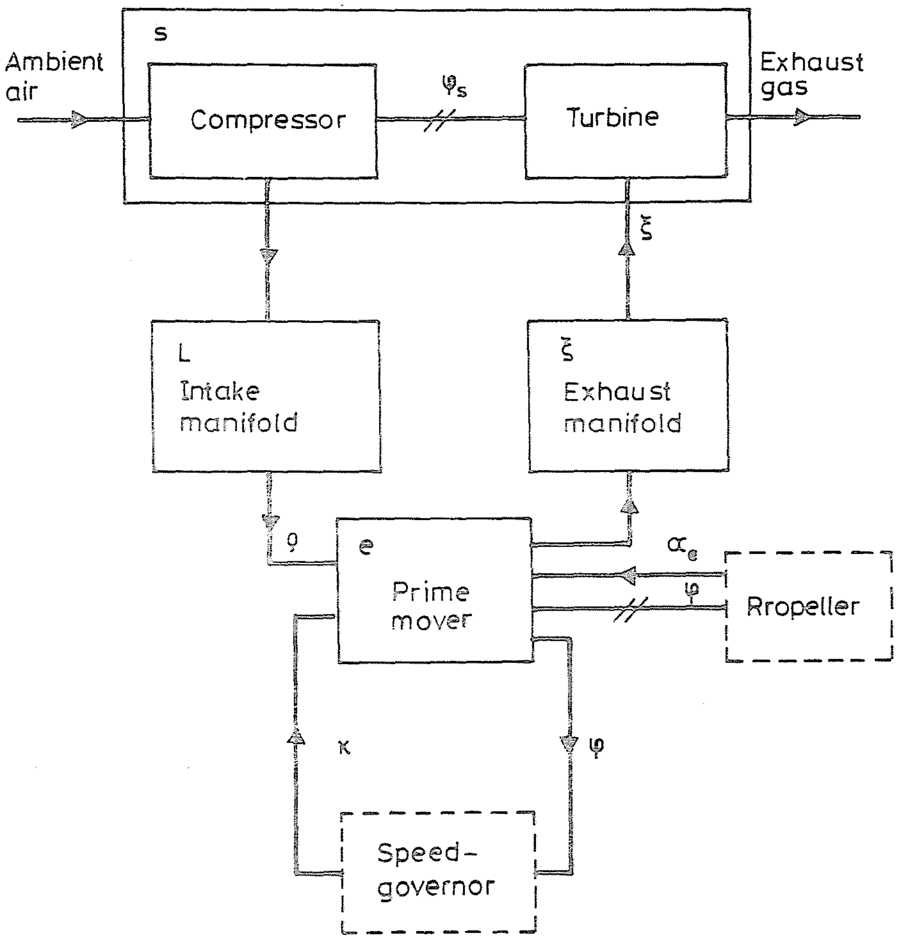

Fig. 1. Layout of the model of a supercharged diesel engine

2.3. Model of a Direct-acting PT2-type Mechanical Speed Governor

The model is represented by its partial transfer functions connected in parallel

$$Y_g^{\varphi}(p) = \frac{1}{d_g(p)} = \frac{1}{T_g^2 p^2 + T_d p + \delta_z} \tag{1}$$

of φ as one of the input signals, and

$$Y_g^{\alpha}(p) = -\frac{\Theta_g}{d_g(p)} = -\frac{\Theta_g}{T_g^2 p^2 + T_d p + \delta_z} \tag{2}$$

of the α_g relative tensioning of the flyweight springs considered the second input signal. The output signal of the model is η relative displacement of

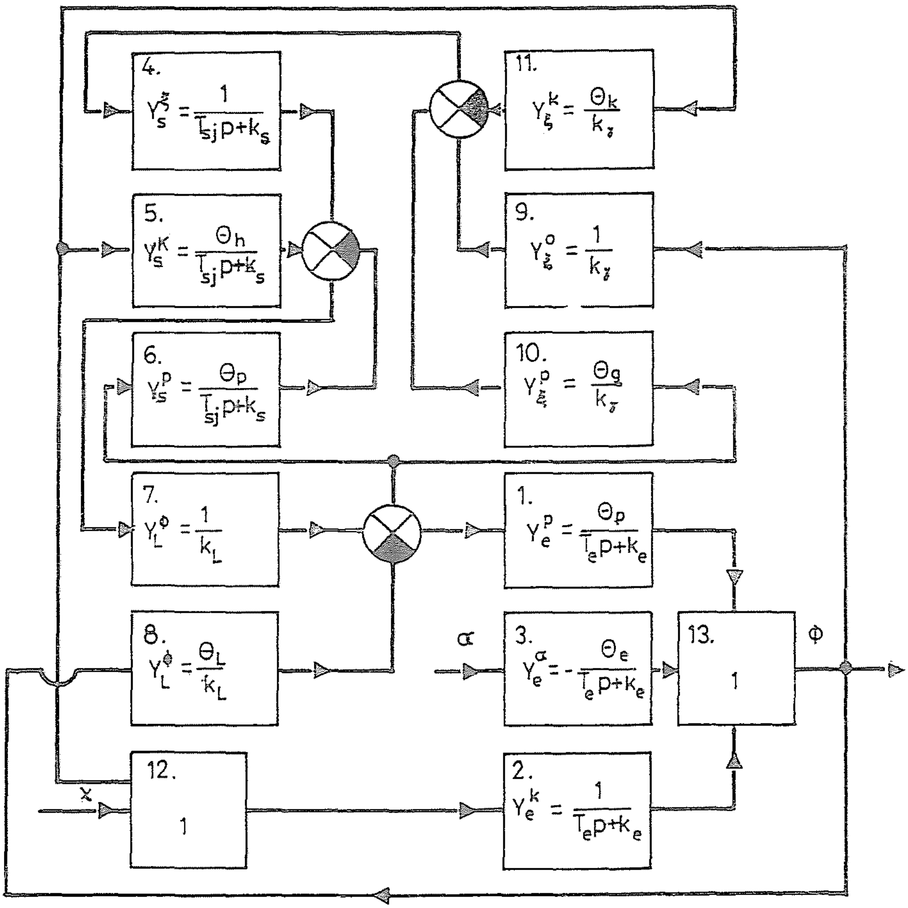

Fig. 2. Block diagram of the model of a supercharged diesel engine

the flyweight sleeve.

Both transfer functions (1) and (2) are of second-order proportional type. The state-space model of the governor has been set up on the basis of the above transfer function representation, by means of converting commands in MATLAB.

2.4. *Model of a PI-type Hydraulic Speed Governor*

This model is based upon the WOODWARD UG proportional plus integral type hydraulic universal speed governor equipped with a vanishing type compensating feedback.

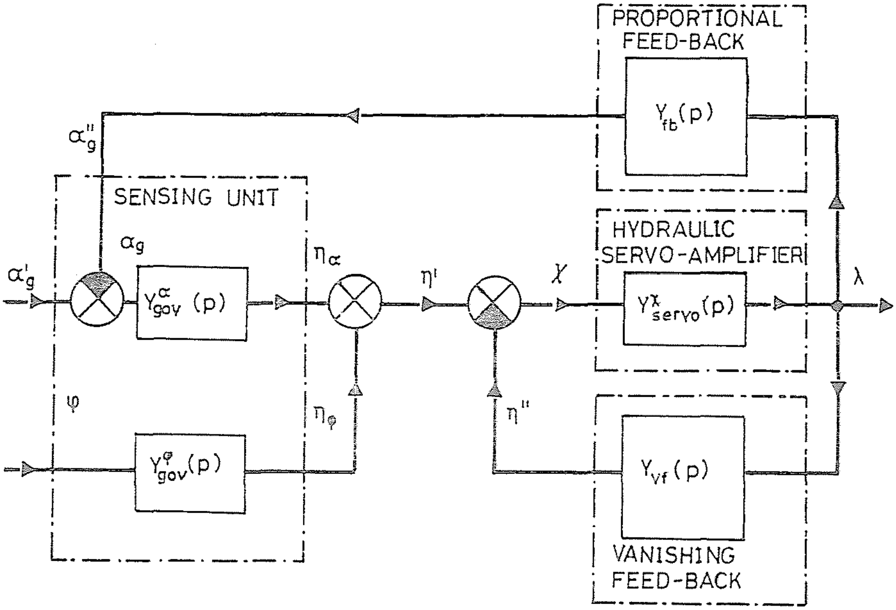

Fig. 3. Block diagram of a PI-type hydraulic universal speed governor

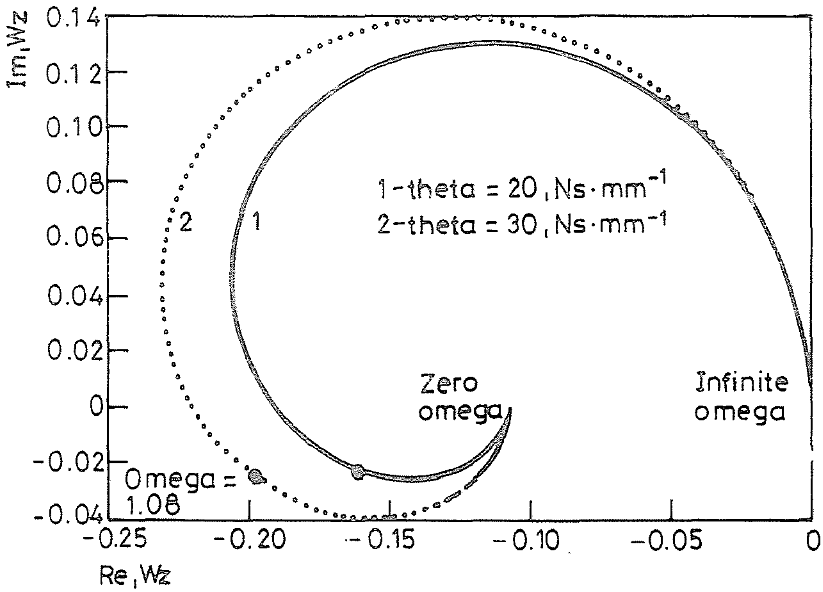

Fig. 4. Nyquist diagram of the controlled variable

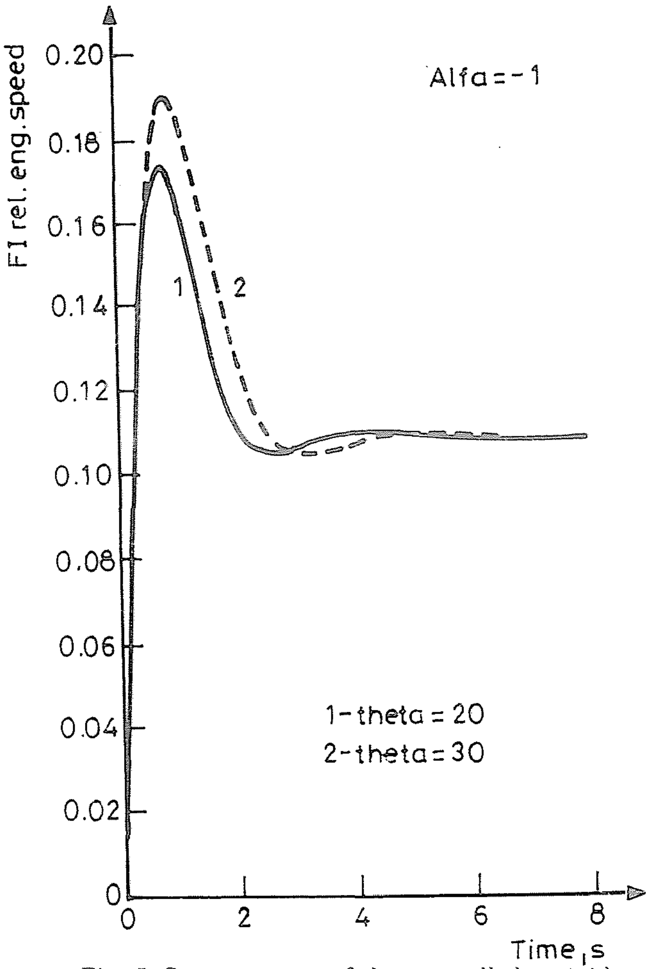

Fig. 5. Step response of the controlled variable

According to its layout in transfer function representation (*Fig. 3*), φ and α'_g partial relative tensioning of the flyweight springs are the input signals, while λ relative displacement of the servo-piston is the output signal of the model.

The transfer functions and the input-output signals of the subsystems in *Fig. 3* are as follows:

- The integral type transfer function related to χ relative displacement of the pilot valve, as the input signal of the hydraulic servo-amplifier unit, is

$$Y^{\chi}_{servo}(p) = \frac{1}{T_{servo}\, p}. \tag{3}$$

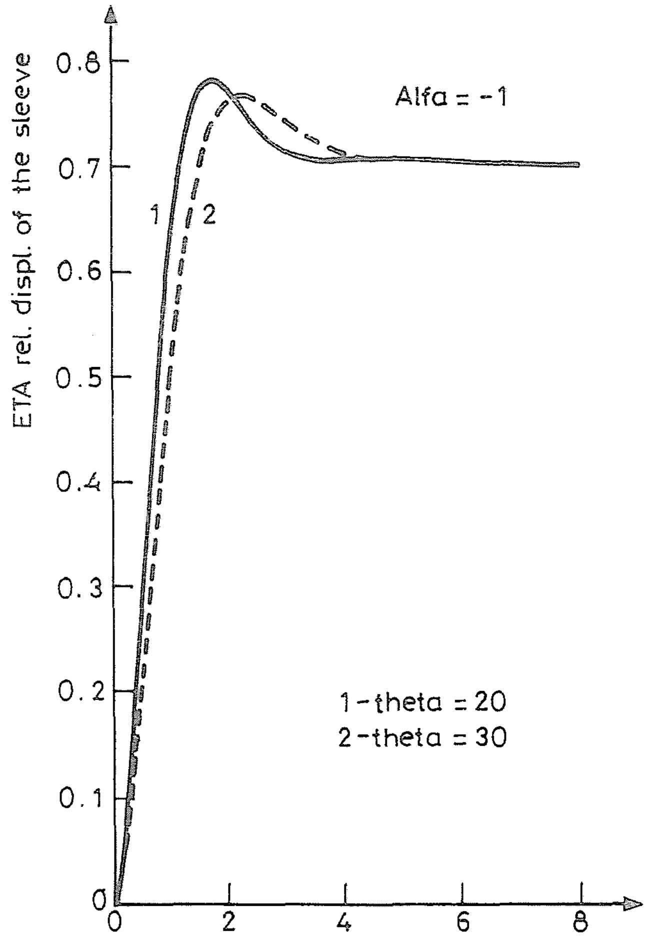

Fig. 6. Step response of η

The output signal of the subsystem is λ, the very output of the model.

- The second-order proportional type partial transfer functions, connected in parallel, of the sensing unit subsystem are

$$Y_{gov}^{\alpha}(p) = \frac{\Theta_{gov}}{T_g^2 p^2 + T_d p + \delta_z} \tag{4}$$

of α_g as one of the input signals, and

$$Y_{gov}^{\varphi}(p) = \frac{1}{T_g^2 p^2 + T_d p + \delta_z} \tag{5}$$

of φ considered the second input signal. The output signal of the subsystem is η' partial relative displacement of the flyweight sleeve.

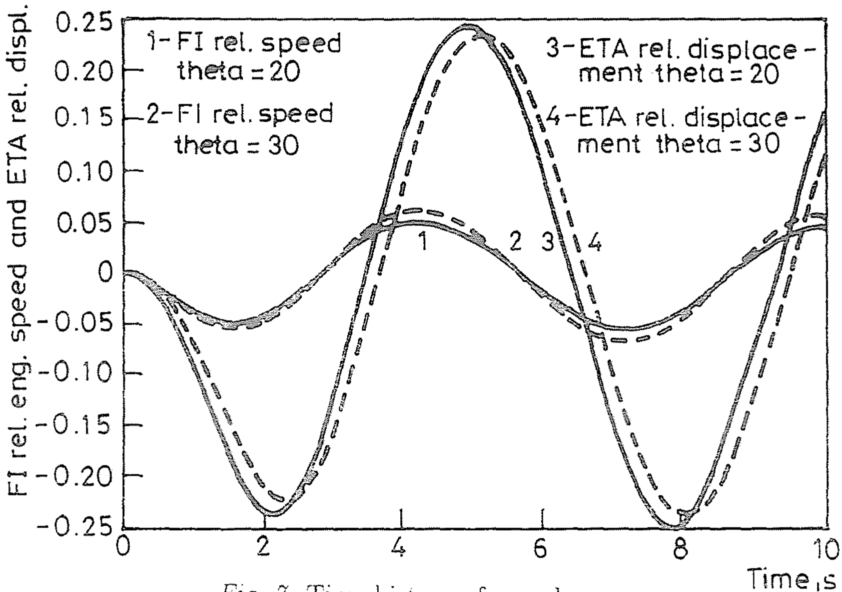

Fig. 7. Time history of φ and η

– The proportional type transfer function of the inner proportional feedback is

$$Y_{fb}(p) = \Theta_{fb}. \tag{6}$$

λ is the input signal, and α_g'' partial relative tensioning of the flyweight springs is the inner feedback signal, as the output one of the subsystem.

– The first-order derivative type transfer function of the compensating vanishing feedback

$$Y_{vf}(p) = u\beta_{PI}\frac{T_{PI} \cdot p}{1 + T_{PI} \cdot p} \tag{7}$$

has λ input signal and η'' partial relative displacement of the flyweight sleeve, where T_{PI} and u are the time constant and the rate of the compensation, respectively.

The state-space model of the governor has been developed by simple conversion in MATLAB. Detailed information on the models of the speed governors is available in [3], [4].

2.5. *Models of a Two-pulse Bing-bang Type Electronic Speed Governor*

The operating method of this governor is supposed to be similar to that of the mechanical speed governors, operating on two-stroke, low-speed marine

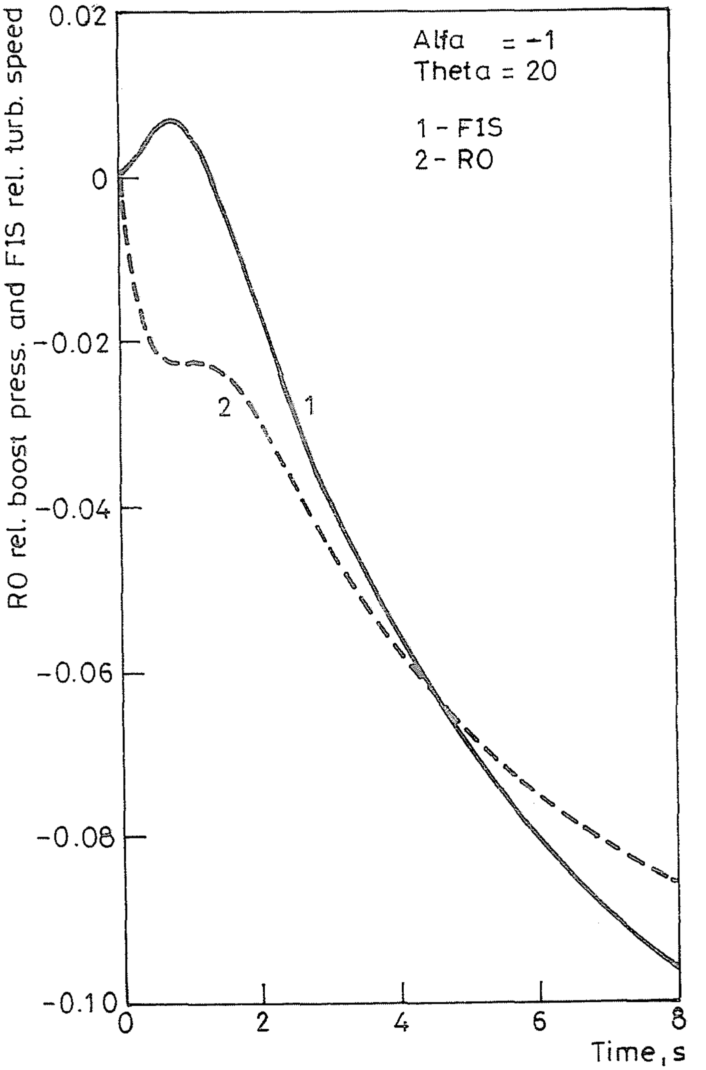

Fig. 8. Step response of φ_s and ρ

diesel engines. Thus, value of the controlled propeller speed is maintained by the governor between upper and lower limits, applying successive cut-out of injection or successive reduction in amount of the injected fuel.

Two models of this kind were developed, both of them have been written in form of M-files in MATLAB, by using logical variables. The only difference between these models refers to the rate of fuelling during the cut-out periods, one of them cuts out injection in these periods completely.

Speed governors of this type have been supposed to be co-operating

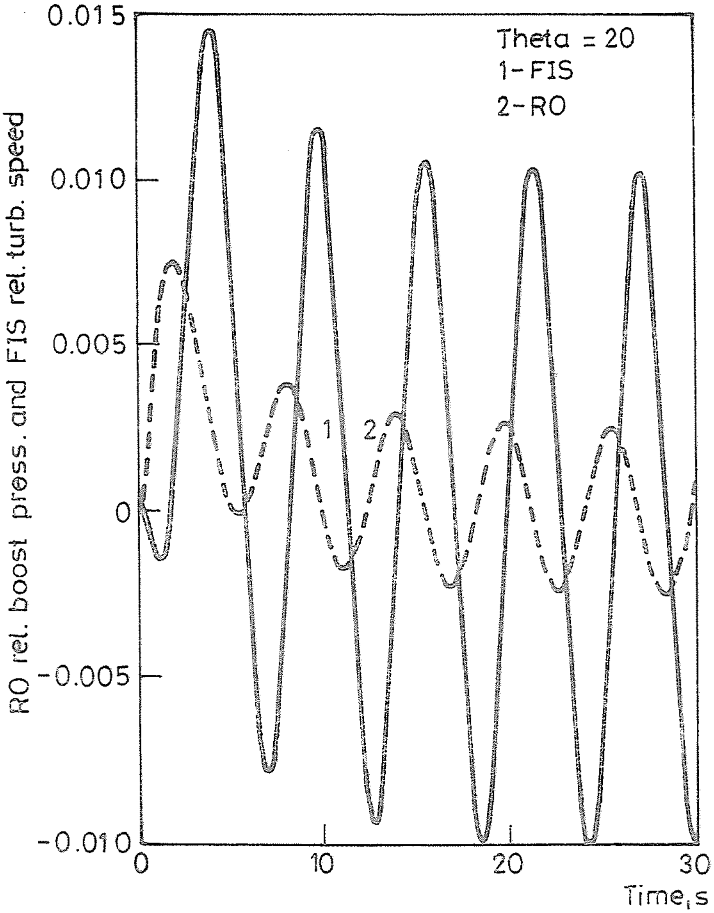

Fig. 9. Time history of φ_s and ρ

with electronically controlled constant-pressure injection systems [5]. Time lag represented by the applied electro-hydraulic injector valves has been neglected.

3. Results of the Investigations

3.1. Closed Control Loop, Comprising a Direct-acting PT2-type Mechanical Speed Governor

Figs 4–7 represent the influence of the ϑ coefficient of proportionality, i.e. the effect of the changing condition of the fuel injection system, on

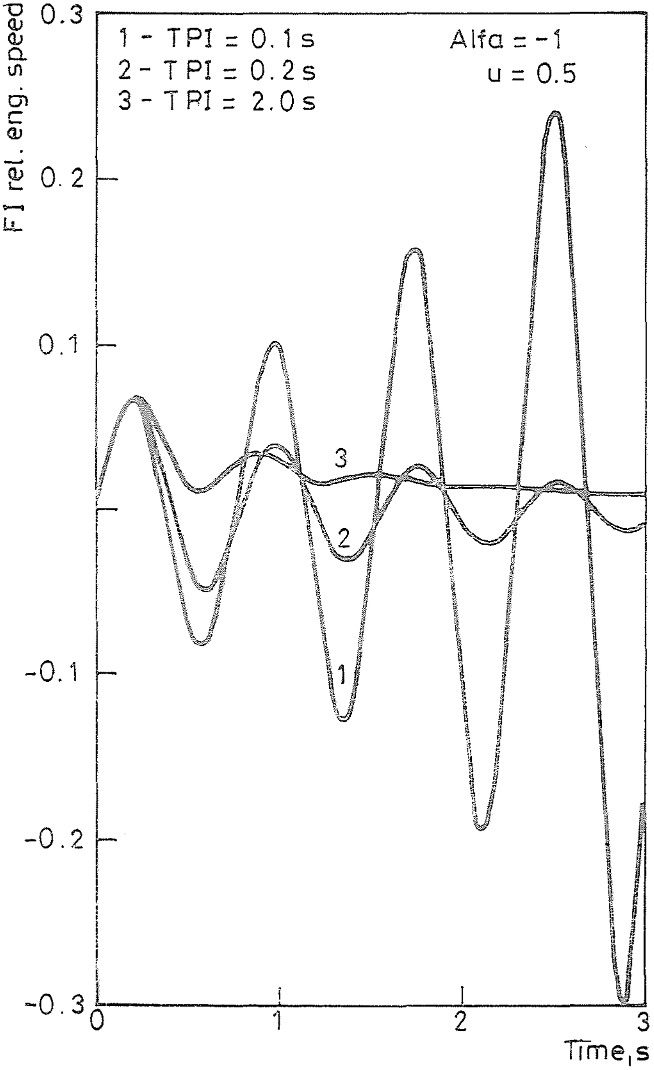

Fig. 10. Step responses at various pairs of u and T_{PI}

- the Nyquist gain-phase characteristics of the closed control loop (*Fig. 4*),
- the step response of φ controlled variable at $\alpha_\varepsilon = -1$ sudden load rejection (*Fig. 5*),
- the step response of η relative displacement of the flyweight sleeve at $\alpha_\varepsilon = -1$ (*Fig. 6*),
- the time history of φ and η at sinusoidal excitation of $\omega_g = 1.08$ [rad/s] angular frequency (*Fig. 7*).

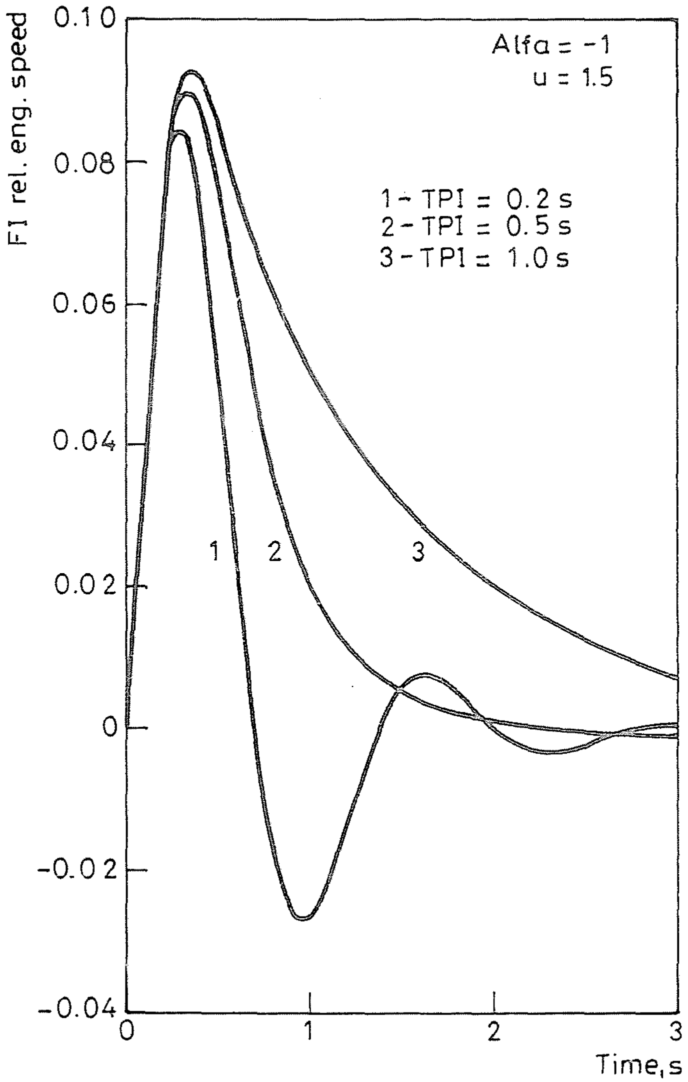

Fig. 11. Step responses at various pairs of u and T_{PI}

Figs 8 and *9* comprise the step response and the time history, at $\omega_g = 1.08$ [rad/s] sinusoidal excitation, of ρ relative boost pressure and φ_s relative speed of the supercharger.

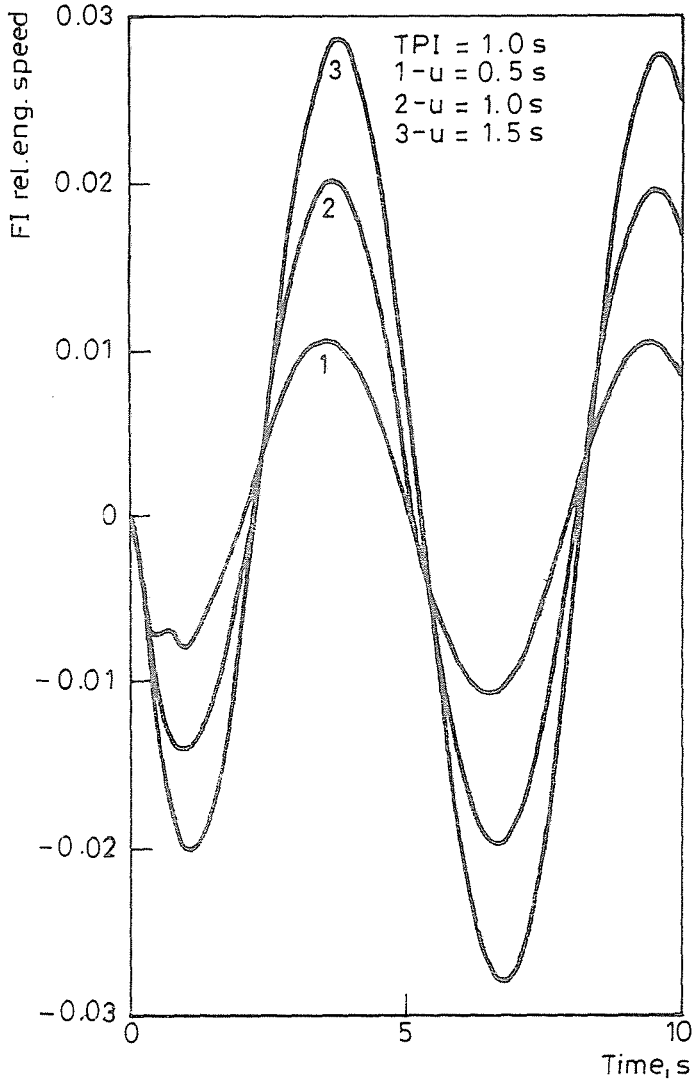

Fig. 12. Time history of φ at various pairs of u and T_{PI}

3.2. Closed Control Loop. Comprising a PI-type Hydraulic Speed Governor

Figs 10–12 represent the influence of the varying u, T_{PI} pairs on the step-response (*Figs 10* and *11*) as well as on the time history at $\omega_g = 1.08$ [rad/s] sinusoidal excitation, of controlled variable φ(*Fig. 12*).

Fig. 13 shows the time history of λ relative displacement of the servo-piston, at $\omega_g = 1.08$ [rad/s] sinusoidal excitation.

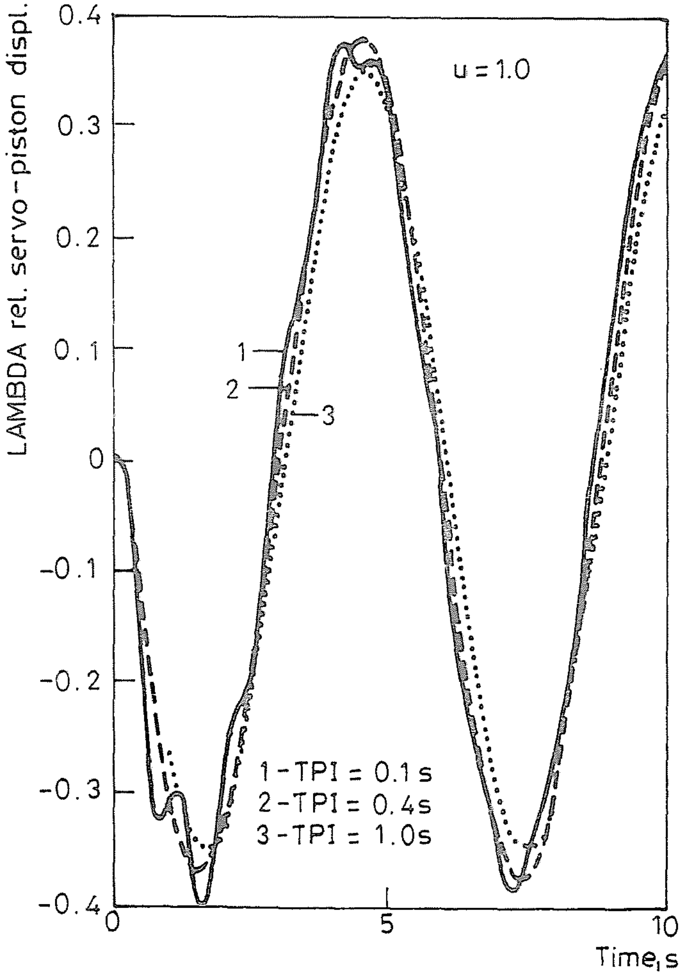

Fig. 13. Time history of λ at various pairs of u and T_{PI}

In *Fig. 10* curves 2 and 3 relate to stable settings, while curve 1 to an instable setting of the compensating system. Optimal compensation provides quick response of the system, without hunting or surging of the prime mover. In *Fig. 11* curve 1 corresponds to an undercompensated, curve 3 to an overcompensated, while curve 2 to the optimally compensated settings of the system, whereas all these settings yield stable operation. In *Fig. 12* on the curve 1, likewise on the curve 1 in *Fig. 13*, small wavelets have been superposed to the sine-shaped output signal. This phenomenon demonstrates hunting of the prime mover due to an undercompensated setting of T_{PI} and u.

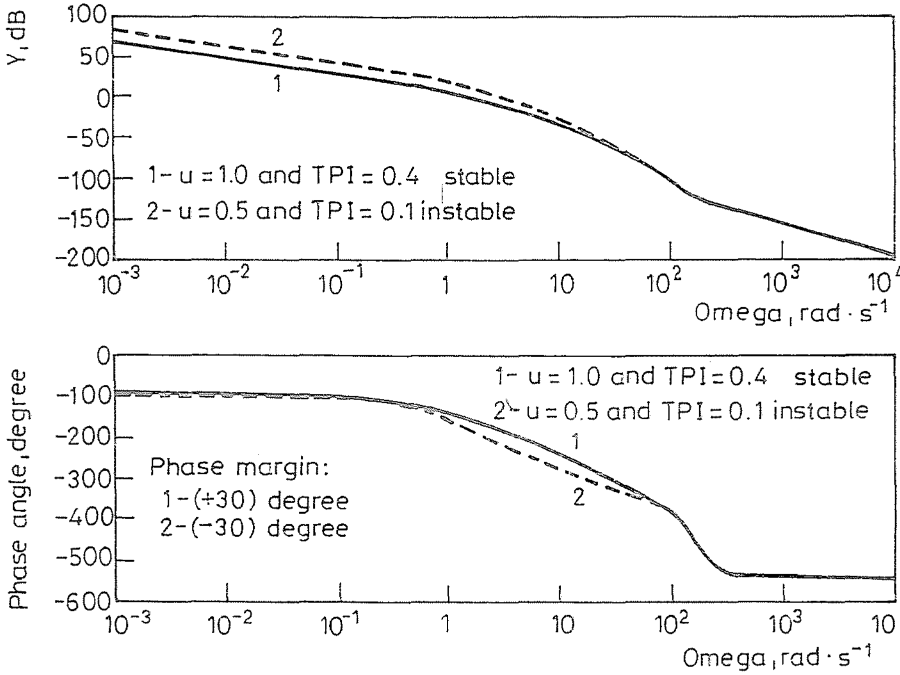

Fig. 14. Bode diagram of the open control loop

Fig. 14 contains the Bode logarithmic plots of the open control loop. These plots have been calculated at a stable, as well as at an instable pair of u and T_{PI} variables of the compensating vanishing feed back.

Figs 10–14 have been plotted assuming new condition of the fuel injection system, such as the fuel rack, linkage and the constant-stroke, edge-controlled fuel pumps.

3.3. Closed Control Loop, Comprising a Two-pulse Bing-bang Type Electronic Speed Governor

Figs 15 and *16* represent the time history of φ controlled variable and that of κ relative displacement of the fuel rack, at $\omega_g = 1.08$ [rad/s] sinusoidal excitation. Pre-set lower κ limit has been $\kappa = -1$ in *Fig. 15*, while $\kappa = -0.325$ in *Fig. 16*.

Comparing the time history of the φ controlled variable in *Fig. 15* to that in *Fig. 16*, the latter shows better control performance. By applying smaller gap between the upper and lower fuelling limits result fluctuation at lower frequency of the φ relative engine speed.

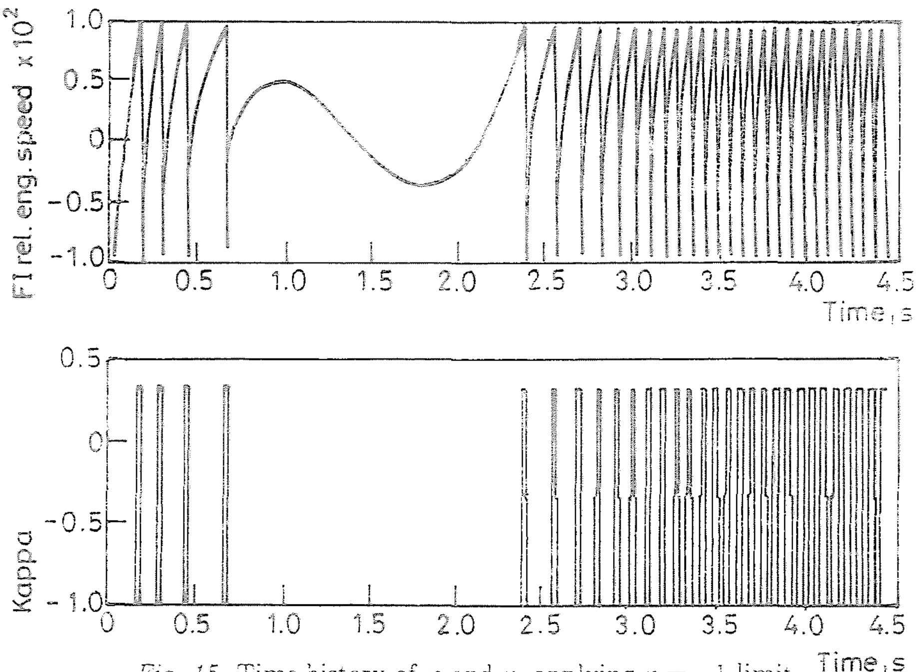

Fig. 15. Time history of φ and κ, applying $\kappa = -1$ limit

4. Concluding Remarks

On the basis of our investigations the following conclusions can be drawn:

- Applying a direct-acting mechanical speed governor, amplitude ratio of φ controlled variable, at regular sinusoidal waves as an excitation, is basically determined by the fuel injection system. Assuming deteriorated conditions of the fuel rack, linkage and of the fuel pumps, ξ_φ damping factor of the second-order system, representing the speed governor, is far from being optimum, even at robust dimensions of the governor. Application of constant-stroke, edge-controlled fuel pumps results considerable drawbacks, constant-pressure fuel injection systems are favourable in this regard. However, on the other hand, this governor is less influenced by the varying excitation frequency (*Figs 4–7*).
- Regarding direct-acting mechanical speed governors, at a sudden change in load, e.g. due to lost propeller, controlled variable probably cannot be maintained below 120% of its nominal value prescribed as a limit by the rules of registers, thus application of an independent overspeed protection is necessary. Applying a governor of this type, influence of the speed and torque fluctuations due to waves, concerning noise and vibration, cannot be disregarded (*Figs 5* and *6*).

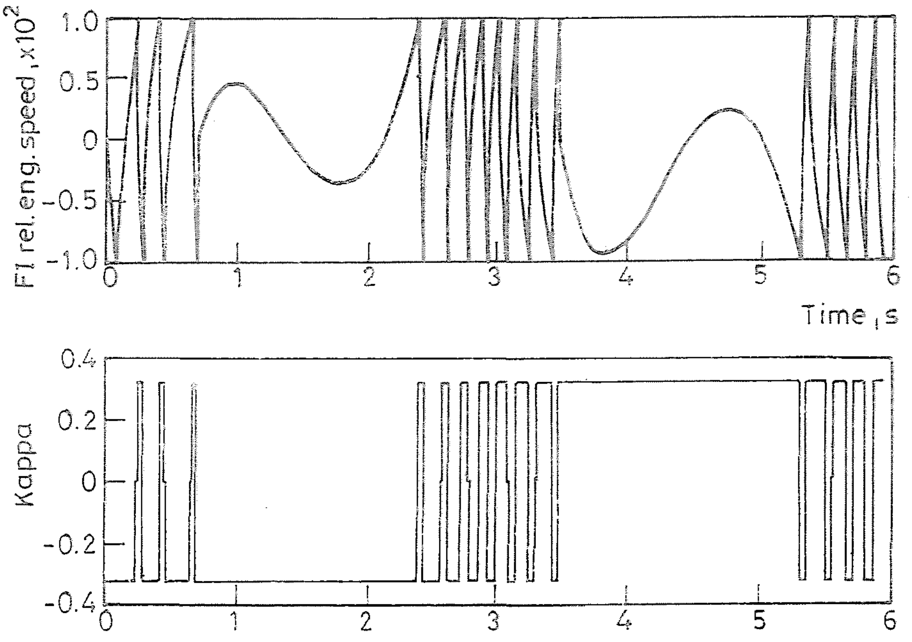

Fig. 16. Time history of φ and κ, applying $\kappa = -0.325$ limit

- Transient behaviour and stability of a closed control loop, containing a PI-type hydraulic speed governor, is highly influenced by the excitation frequency. Optimum setting of u and T_{PI} is basically frequency dependent. The momentary speed variation, the length of the transient process, surging and stability are sharply influenced by the pair of u and T_{PI} (*Figs 10–16*).
- Two-pulse speed governors are less sensitive to the excitation frequency and to the deteriorated conditions of the fuelling system. However, the scope of application for the two-pulse speed governors is limited by the torsional vibration of the shafting. Their application in propulsion plants, containing reverse-reduction gears or flexible couplings, is conditional (*Figs 15* and *16*).
- Surging in the angular velocity of the turbo-supercharger impeller proved to be negligible in this engine category, applying even the less favourable type of the investigated speed governors at a sudden load change (*Figs 8* and *9*).

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