Low-Speed Hunting of the Pneumatically Governed Compression Ignition Engine*
(Effects of Various Parameters on the Amplitude and Frequency of Limit Cycle)

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For a better understanding of hunting, it is desirable to obtain the amplitude and frequency of limit cycle analytically. In the author's last two reports, as a first step toward nonlinear approximate analysis, the effects of individual parameters of a closed-engine-governor system on the instability of the equilibrium state, especially on the frequency and increment of amplitude, were linearly estimated. In the present report, results show that the limit cycle disappears over the whole speed range if the moment of inertia of the crankshaft system exceeds 1.8 times that in the actual engine ; this phenomenon resembles the disappearance due to minimized subventuri pressure lag. Results also show that the amplitude decreases, except at extremely low speeds, with decreasing mass of the governing system, although mass reduction is difficult in reality . Further, it is found that the limit cycle does not disappear no matter what value the damping of the governor may take.

Key Words: Vibration, Instability, Shaft Speed Hunting, Limit Cycle, Internal Combustion Engine, Compression Ignition Engine, Pneumatic Governor, Engine Speed Control, Numerical Simulation

1. Introduction

In a governed engine, the idling speed cannot remain constant and is followed by a low frequency noise of its own[1]~[10]. This fluctuation of engine speed is called low-speed hunting.

A pneumatically governed engine controls the fuel control rack by reducing pressure in a narrow passage, called the subventuri, beside the throttle valve. Reduced pressure due to increased engine speed displaces the rack in the direction of decreasing fuel delivery through a diaphragm combined with a spring. The engine speed depends on the throttle valve opening.

It was revealed by the present author that the low-speed hunting in the pneumatically governed engine is the self- excited oscillation caused by the phase lag of the suction pressure for displacement of the fuel control rack. Further, computer simulation yields a transient process during which a small oscillation develops into a sustained oscillation with a large amplitude[9][11]~[14]. However, for a better understanding of hunting, it is desirable to obtain the amplitude and frequency of the limit cycle analytically.

The author's previous reports[15][16] linearly estimated the instability of the equilibrium state of a governed engine, compared it with the limit cycle given by a numerical, nonlinear simulation with respect to the frequency and increment of amplitude, and estimated the effects of individual parameters of a closed-engine-governor system on the instability of the equilibrium state, especially on the frequency and increment of amplitude, as a first step toward nonlinear approximate analysis.

The present report estimates the effects of three parameters besides a subventuri pressure lag, i.e., the moment of inertia of the crankshaft system and mass and damping of the governing system, that have a marked influence on the limit cycle characteristic as a step toward analytically explaining the mechanism of limit cycle evolution.

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2. Outline of Numerical Calculation

For an engine speed to be examined, the value of each variable at the equilibrium state must be determined as the initial value for calculation. The transient behavior of each element of the closed-engine-governor loop is computed, responding to a small step increment of the throttle opening or the small disturbance of engine speed, so as to investigate the amplitude and frequency of the limit cycle.

Let N_e rpm denote the engine speed. Then the rate of engine speed increase during idling is given by the expression

$$J_e\frac{dN_e}{dt}=T_e \qquad (1)$$

where I_e is the moment of inertia of the crankshaft system, $J_e=2\pi I_e/60$, and T_e is the accelerating torque. T_e is assumed to be constant throughout each 180° crank angle for a four-cylinder engine and becomes a function of fuel rack displacement and engine speed. The quantity of fuel in each injection is assumed to be determined by the engine speed N_e and rack displacement X at the moment when the corresponding piston reaches top dead center. The subroutine program gives a map of the measured torque as a function of the rack displacement and engine speed, and the torque T_{ei-1} corresponding to the values of X_{i-1} and N_{ei-1} is found by interpolation to be

$$T_{ei-1}=T_e(X_{i-1},N_{ei-1}) \qquad (2)$$

The response of rack displacement X to the governing pressure P can be written as

$$m_e\frac{d^2X}{dt^2}+C_e\frac{dX}{dt}+k(X+L_0-L)=A_dP \qquad (3)$$

where m_e is the equivalent mass of moving parts of the pneumatic governor system, C_e the equivalent viscous damping coefficient, k the stiffness of the rack spring, L_0 the length of rack spring at $X=0$, L the free length of rack spring, and A_d the effective diaphragm area[12]. Under a steady running condition without hunting, the mean value of the governing pressure shows a linear relation to the mean engine speed at a given throttle opening. In transients, however, in order to satisfy a large number of experimental data, the mean governing pressure P_π for each 180° crank angle at each throttle opening requires the following equation, where N_e, α, and T_p are the engine speed, the sensitivity of governing pressure to the steady-state engine speed at each throttle opening, and the time constant of the first order, respectively:

$$T_p\frac{dP_\pi}{dt}+P_\pi=-\alpha N_e \qquad (4)$$

Since the higher frequency component caused by the suction stroke of each piston somewhat affects the governing of fuel delivery, the waveform of reduced pressure between the instant of the i-1 th injection t_{i-1} and that of t_i is given as

$$P=P_{\pi_{i-1}}(1-\cos\omega_{m_{i-1}}t) \qquad (5)$$

where $\omega_{m_{i-1}}$ is the frequency caused by the suction stroke depending on N_e, and the origin of t is set at the instant when the piston leaves top dead center.

Figure 1 shows the calculated transient behavior on a phase plane plotting rack displacement at the instant of injection versus engine speed, indicating development of a clockwise limit cycle of the closed-engine-governor loop responding to a small step increment of the throttle opening at an initial engine speed of 800 rpm. The trajectories near the limit cycle spiral on to it from its inside and outside; accordingly, this limit cycle is stable[13]. The reference values in the calculation are those of the test engine and fuel injection pump[9][13].

3. Effects of Various Parameters on the Limit Cycle of the Closed-Engine-Governor Loop

3.1 Effect of phase lag of subventuri pressure

Figure 2 shows the measured limit cycles on the phase plane plotting rack displacement at the instant of injection versus mean engine speed throughout each 180° crank angle, while Fig. 3 shows the variations under the suction pressure control with the minimized phase lag, indicating disappearance of the limit cycles. The phase lag of subventuri pressure responding to the low-frequency fluctuation of engine speed is considered to be a time constant of the first order. It has

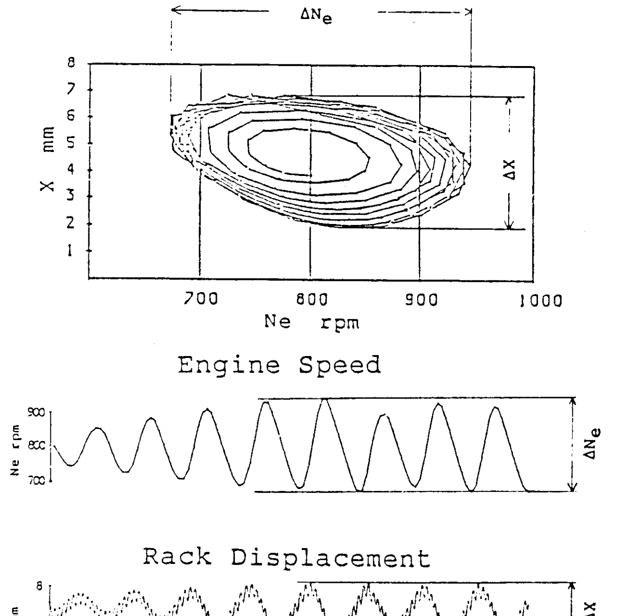

Fig. 1 Calculated limit cycle in a phase plane

already been reported that the calculated engine speed fluctuation, i.e., the amplitude of the limit cycle, is in good agreement with the experimental results, and that the instability of equilibrium does not appear in the linearized calculation in the case in which the phase lag is not taken into account[11][13][16]. The amplitude of limit cycle increases and the frequency of it becomes lower with an increasing value of T_p.

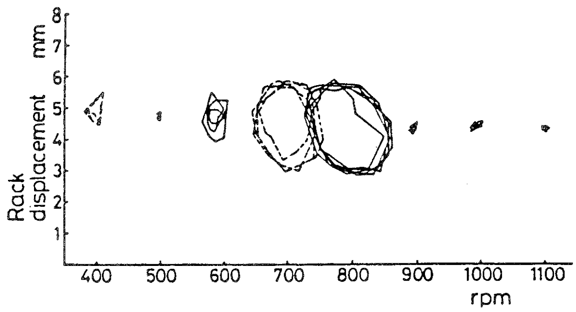

Fig. 2 Measured limit cycles on the phase plane plotting rack displacement at the instant of injection versus engine speed

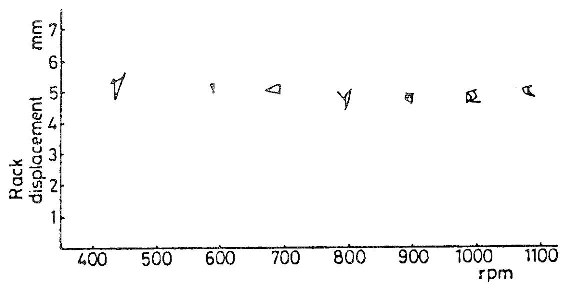

Fig. 3 Variations under the suction pressure control with the minimized phase lag

3.2 Effect of moment of inertia of the crankshaft system

Figure 4 shows the calculated results of limit cycles on the phase plane for the values of the moment of inertia of crankshaft system J_e from 0.5 times to 1.5 times the reference value $J_{eo}(=0.261\ \mathrm{kgm^2})$, where the limit cycles run through in a clockwise direction, because engine speed increases with increasing rack displacement. As the value of J_e increases, the fluctuation at each equilibrium engine speed becomes smaller and also the hunting region shifts toward lower engine speed with a narrower speed range. Finally, for $J_e/J_{eo}=1.5$, the fluctuation becomes small over all speed ranges, and for $J_e/J_{eo}=1.8$, the hunting completely disappears. This result is consistent with the result of the linear calculation that the critical J_e/J_{eo} ratio between stable and unstable is 1.8. Figure 4 also shows that the amplitude of engine speed relative to that of rack displacement decreases with increasing J_e.

Figure 5 shows that the frequency decreases with increasing values of J_e. Figure 6 shows that the maximum amplitude of engine speed decreases with increasing values of J_e. Figure 7 is the result of a computer simulation for $J_e/J_{eo}=1.8$, where the initial disturbance converges rapidly to the equilibrium state with the frequency of about 1.8 Hz.

3.3 Effect of equivalent mass of the governing system

The equivalent mass m_e of the governing system is composed of an equivalent mass of the moment of inertia of plungers, pinions and sleeves, and also of the

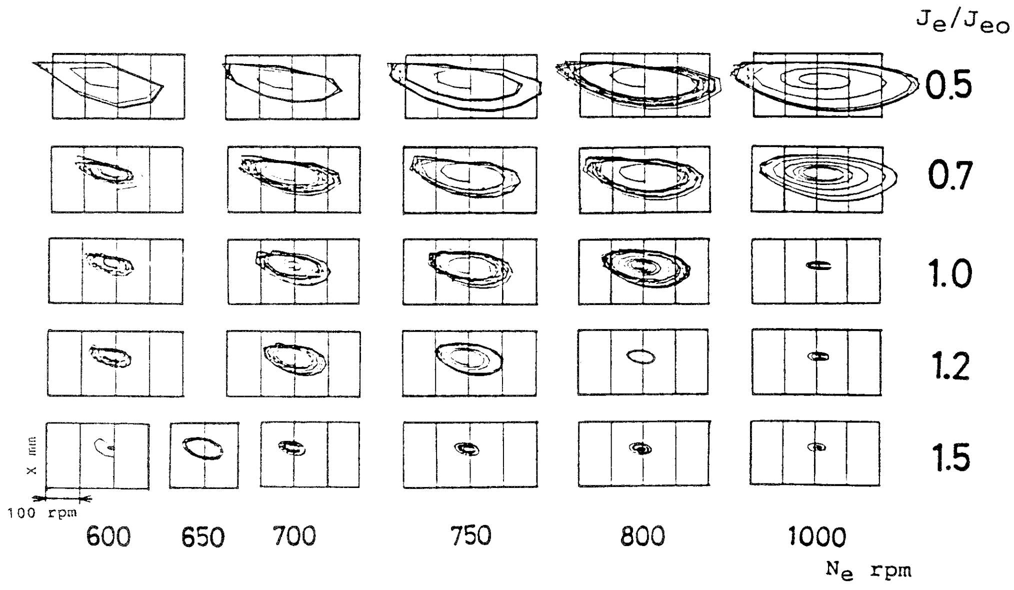

Fig. 4 Effect of moment of inertia of the crankshaft system on the limit cycle

mass of the control rack itself and of the moving parts of the diaphragm[12].

Figure 8 shows the calculated results of the limit cycle at each equilibrium engine speed for values of mass from 0.1 times to 2.0 times the reference value m_{eo} of 0.273 kg, where the fluctuation and hunting region decrease with decreasing mass of the governing system, except at extremely low speeds of around 500 rpm. The fluctuation becomes very small for $m_e/m_{eo}=0.25$, and it disappears over almost all speed ranges for $m_e/m_{eo}=0.1$. This result differs slightly from the linearized result of small oscillation near the equilibrium state. The linearly calculated logarithmic increment of amplitude of small unstable oscillation becomes small with decreasing mass of the governing system, but the small oscillation never becomes stable. Figure 9 is the calculated result for frequency of the closed loop, which shows that the mass of the governing system has only a small effect on the frequency. As shown in Fig. 10, the maximum amplitude of engine speed increases with increasing mass of the governing system. Figure 11 is an example of computer simulation at 800 rpm for $m_e/m_{eo}=0.25$, where the initial

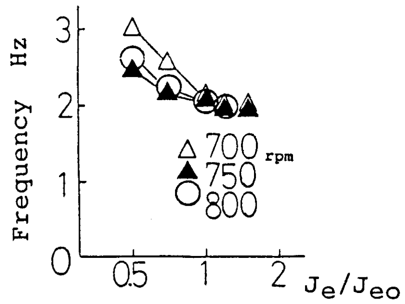

Fig. 5 Effect of moment of inertia of the crankshaft system on the frequency of limit cycle

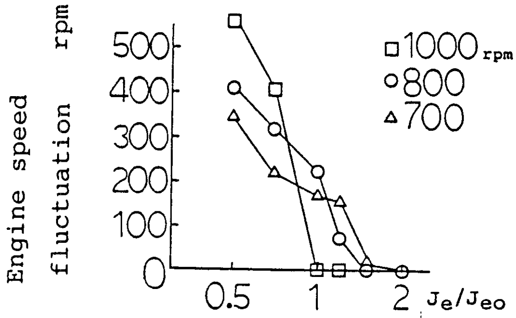

Fig. 6 Effect of moment of inertia of the crankshaft system on the amplitude of engine speed

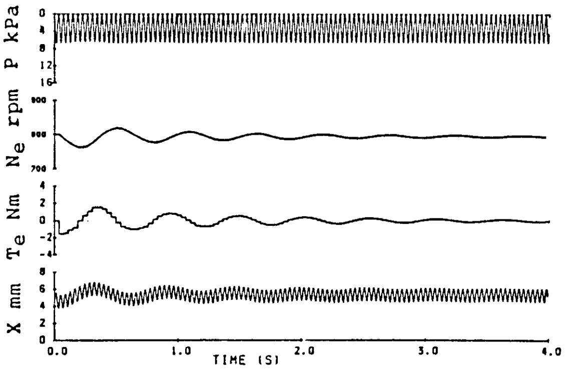

Fig. 7 Result of computer simulation in the case of $J_e/J_{eo}=1.8$ (800 rpm)

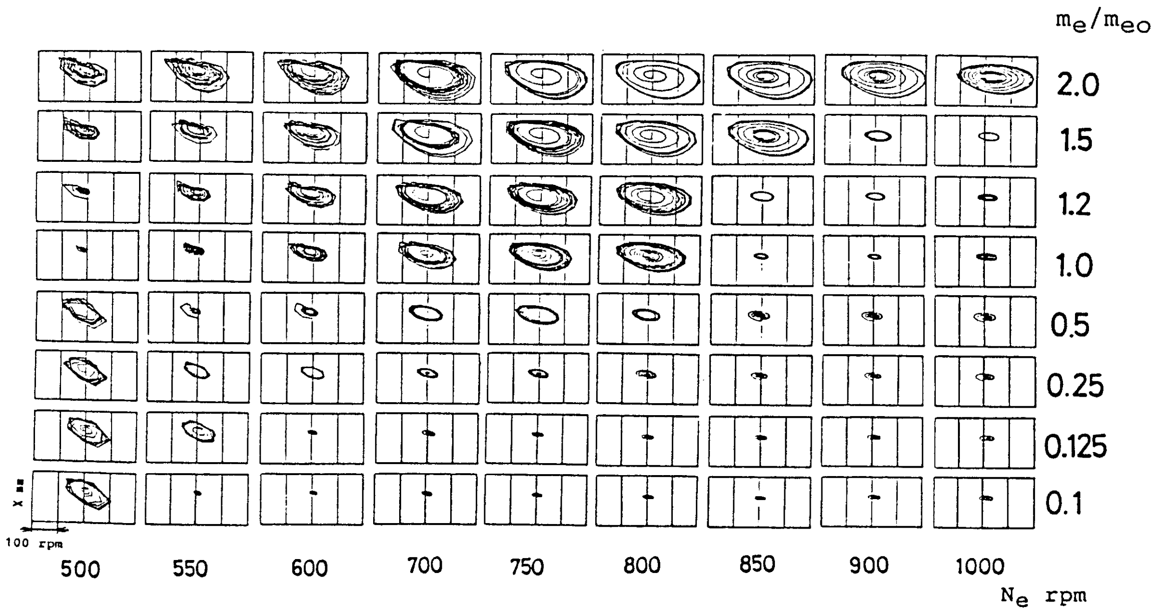

Fig. 8 Effect of equivalent mass of the governing system on the limit cycle

disturbance due to a small increment of throttle opening converges and hunting does not occur. However, it is difficult to decrease the mass in reality.

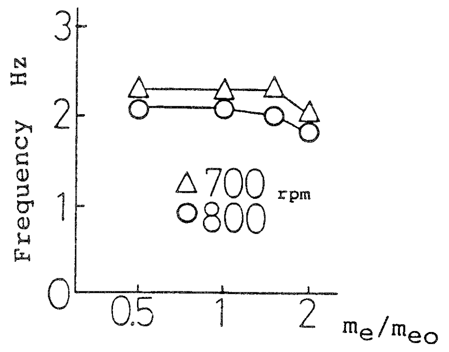

Fig. 9 Effect of equivalent mass of the governing system on the frequency of limit cycle

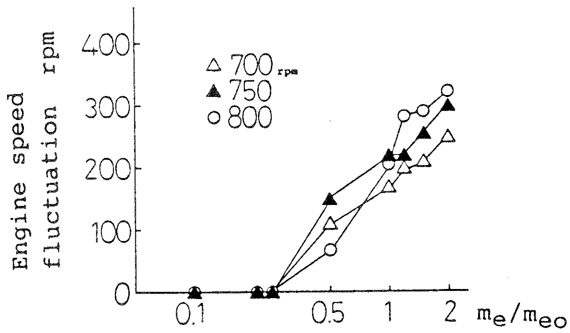

Fig. 10 Effect of equivalent mass of the governing system on the amplitude of engine speed

3.4 Effect of the equivalent damping of the governing system

The equivalent damping coefficient[12] $C_{eo}=24.9$ Ns/m of the governing system is used as the reference value with a chamber 50 cm^3 in volume, liaison pipe 40 cm long and 8 mm in inner diameter, and equivalent diaphragm area of 21.0 cm^2. In the linear approximation, the damping of the governing system does not stabilize the closed loop, but affects the frequency. In this section, the effect of C_e on the limit cycle is considered through computer simulation.

Figure 12 shows the calculated results for the limit cycle at each equilibrium engine speed for values of the damping coefficient from 0.1 times to 2.0 times the reference value C_{eo}, where the fluctuation increases and the hunting region extends to the lower speed range with increasing damping of the governing

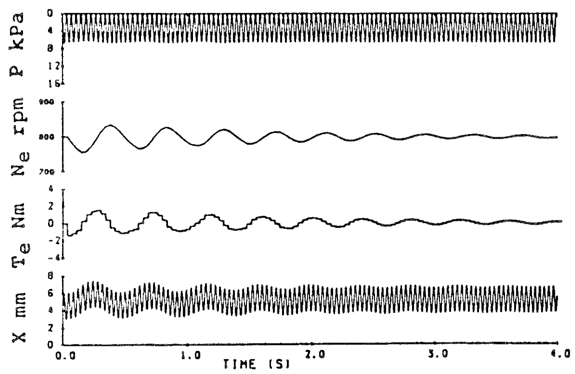

Fig. 11 Result of computer simulation in the case of $m_e/m_{eo}=0.25$ (800 rpm)

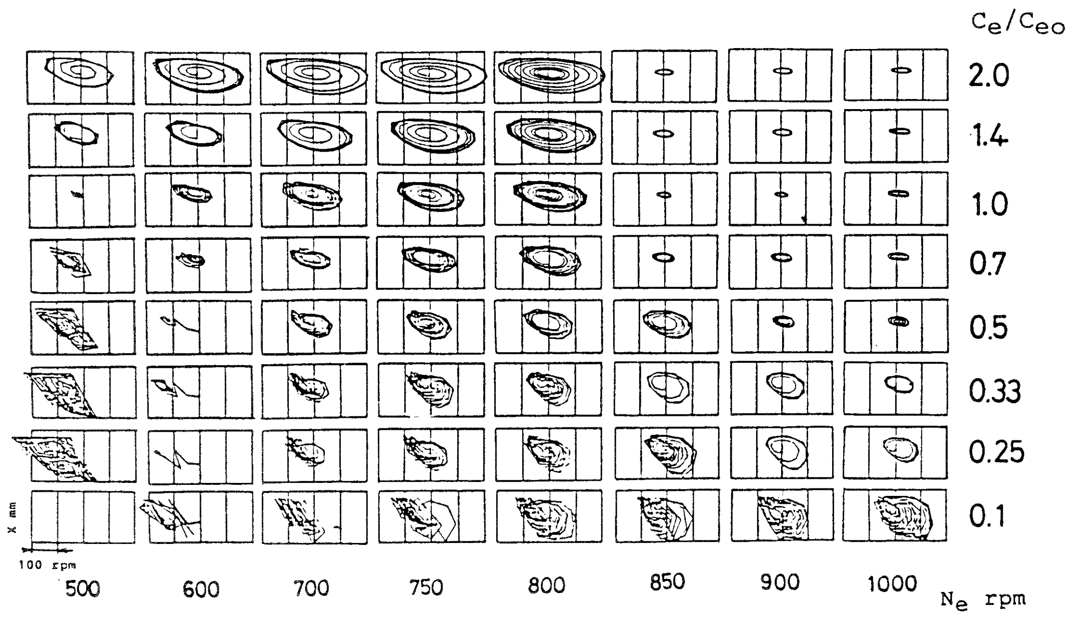

Fig. 12 Effect of equivalent damping of the governing system on the limit cycle

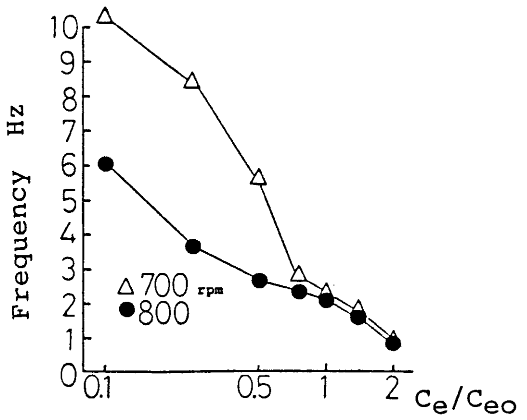

Fig. 13 Effect of equivalent damping of the governing system on the frequency of limit cycle

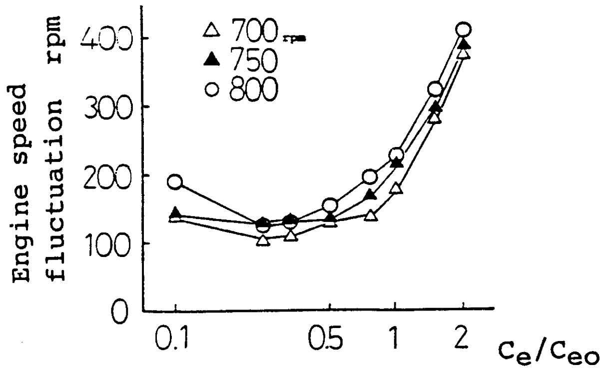

Fig. 14 Effect of equivalent demping of the governing system on the amplitude of engine speed

system. The fluctuation becomes smaller for $C_e/C_{eo}=0.5$, but the hunting region extends to the higher speed range and does not disappear even for $C_e/C_{eo}=0.1$; i.e., the limit cycle does not disappear no matter what value the damping of the governor may take. Further, Fig. 12 shows that the amplitude of rack displacement relative to that of engine speed increases, and the limit cycles exhibit complex behavior with decreasing value of C_e, because the damping coefficient becomes less than the critical damping of the governing system.

Figure 13 shows the calculated result for frequency of the closed loop; here the frequency decreases with increasing value of C_e. As shown in Fig. 14, the maximum amplitude of engine speed fluctuation increases with increasing damping of the governing system, and does not fall below 100 rpm, even with decreasing damping. That means that the increase of damping does not reduce the amplitude, but rather results in an increase.

4. Conclusions

The effects of three parameters that have a marked influence on the limit cycle characteristic were estimated as a step toward analytically explaining the mechanism of limit cycle evolution. Results show that the limit cycle disappears over all speed ranges for a supposed moment of inertia of crankshaft system 1.8 times that in the actual engine, as given by minimized subventuri pressure lag. Results also show that the amplitude decreases, except at extremely low speeds, with decreasing mass of the governing system, although mass reduction is difficult in reality . Further, it is found that the limit cycle does not disappear no matter what value the damping of the governing system may take.

The effects of the other parameters of the closed-engine-governor loop on the limit cycle behavior will be estimated in the next paper.

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