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Low Speed Hunting of Pneumatically Governed Compression-Ignition Engine*

(3rd Report, Computer Simulation of the Low Speed Hunting)

By Yoshihiko KAWAZOE**

Most of the past research works on engine speed hunting have been devoted to discriminating the divergence of a small disturbance given at an equilibrium state, resulting in a hunting frequency estimation different from that of the actual system. In the present report, a numerical simulation is given on the basis of equations of elements of the engine-governor system. As a result of simulation, a self-sustained oscillation develops in the case of speed control with subventuri pressure which has a retarded hunting frequency component compared with the speed fluctuation, while hunting disappears under suction pressure control without phase lag, as predicted in the author's previous work. Calculations show good agreement with the experimental data regarding the frequency, amplitude, and hunting speed range.

Key Words: Vibration, Low Speed Hunting, Hunting, Pneumatic Governor, Compression-Ignition Engine, Computer Simulation, Stability, Self-Sustained Oscillation, Speed Control

1. Introduction

In a pneumatically governed compression-ignition engine its idling speed cannot remain constant in some speed range, followed by a low frequency noise of its own. This fluctuation of the engine speed is called low speed hunting. The purpose of this study is to reveal the mechanism of the low speed hunting peculiar to the pneumatically governed engine, and to devise its preventive measure. It was stated in the 1st report[1] that the key factor must be the phase lag of the governing pressure taken at a narrow passage called subventuri beside a throttle valve, because the hunting disappears when the phase lag is minimized by displacing the pressure source to the common intake duct just down the throttle valve and the subventuri. In the 2nd report[2] it was shown from the results of the frequency response that there exists a hunting as a limit cycle of a nonlinear system. No hunting occurs under the suction pressure control because the phase lag of the governing pressure is very small. Most of the past studies on hunting [3]~[7] dealt with the stability of a linear oscillation system. There has never been investigated such a transient process that a small oscillation develops into a sustained oscillation with a large amplitude. Further the conventional linear theory of the engine-governor loop does not explain the actual phenomenon, giving

a hunting frequency estimation different from that of the actual system[1][6][7]. In the present report, a numerical simulation of the low speed hunting is shown on the basis of equations of elements of the engine-governor system.

2. Outline of the Simulation

A pneumatic governor controls the fuel delivery displacing the fuel control rack with a reduced pressure taken at the subventuri beside a throttle valve, whose opening determines the mean engine speed. Reduced pressure caused by an increased engine speed is applied to a diaphragm combined with a spring and the control rack so as to decrease fuel delivery, where the rack dynamics[8] affects the response. Fuel is sprayed during the injection period determined by the cam angle of the fuel pump, and the quantity of fuel in each injection depends not only on the rack displacement but also on the engine speed. The increment of engine speed as a result of one injection of fuel is determined by its combusion torque, engine inertia and friction torque. In Fig.1 is shown a block diagram of the closed engine-governor loop.

The following are the main points taken into consideration from the results of the experimental studies[1][2][8]. 1) There exists a phase lag of the subventuri pressure responding to the fluctuation of engine speed. 2) The fuel delivery or its effective torque is a nonlinear function of the displacement of the fuel control rack and also depends on the engine speed. 3) Both the subventuri pressure and the displacement of the fuel control rack comprise a component of the hunting frequency and a component of the higher

** Associate Professor, Department of Mechanical Engineering, Saitama Institute of Technology, Okabe, Saitama, 369-02 Japan.

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frequency caused by the suction stroke of each piston. 4) Fuel delivery is controlled during each fuel injection period only.

Simulations are carried out as follows: for an engine speed to be examined, the value of each variable at the equilibrium state such as throttle valve opening and subventuri pressure etc. is determined as the initial value for calculation. The transient behavior of each element of the closed loop responding to a small step increment of the throttle opening is computed.

3. Theory and Method of Calculations

3.1 Crank shaft system

Let Ω denote the engine angular velocity. Then the rate of angular velocity increase at an idling condition is given by the expression

$$I_e\frac{d\Omega}{dt}=M_f-M_R=M_e \qquad (1)$$

where I_e is the moment of inertia of the crank shaft system, M_f the indicated torque or combustion torque, M_R the friction torque, and M_e the accelerating torque. Since the object of simulation is the mean engine speed, M_e is assumed to be constant throughout each 180°crank angle for the case of a four-cylinder engine, without consideration of speed and torque variations of the other kinds. If N_e rpm is used in place of Ω, Eq.(1) becomes

$$J_e\frac{dN_e}{dt}=M_e \qquad (2)$$

where $J_e=(2\pi I_e)/60$.

The indicated torque M_f is generally a function of fuel rack displacement and engine speed, while the friction torque depends on engine speed, so that the accelerating torque M_e 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 and the rack displacement at the moment when the corresponding piston reaches its top dead center. The injected fuel is supposed to generate a constant torque over 1/2 crank revolution. Let N_{ei-1}, N_{ei}, M_{ei-1} and T_{i-1} denote the engine speed at the moment of i-1th injection t_{i-1}, the one at t_i, the constant torque between t_{i-1} and t_i, and the time necessary for acceleration from N_{ei-1} up to N_{ei} respectively. Then Eq.(2) can be expressed as

$$N_{ei}=N_{ei-1}+\frac{M_{ei-1}}{J_e}T_{i-1} \qquad (3)$$

$$T_{i-1}=\frac{60}{N_{ei-1}+N_{ei}} \qquad (4)$$

From Eq.(3) and Eq.(4), we have

$$N_{ei}=\sqrt{N_{ei-1}^2+60\frac{M_{ei-1}}{J_e}} \qquad (5)$$

3.2 Approximating the wave form of governing pressure

At 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, the mean governing pressure P_x for each 180° crank angle at each throttle opening requires the following equation, so as to satisfy a lot of experimental data, 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.

$$T_P\frac{dP_x}{dt}+P_x=-\alpha N_e \qquad (6)$$

Provided P_{xi-1} denotes the mean value of reduced pressure for governing between t_{i-1} and t_i, Eq.(6) may be approximated as

$$T_P\frac{P_{xi-1}-P_{xi-2}}{\Delta T_{i-1}}+P_{xi-1}=-\alpha N_e \qquad (7)$$

while the time length ΔT_{i-1} necessary for the pressure to increase from P_{xi-1} to P_{xi} can be expressed as

$$\Delta T_{i-1}=\frac{30}{N_{ei-1}} \qquad (8)$$

| Desired speed | Depression | Control rack displacement | Fuel flow | Torque | Controlled speed |

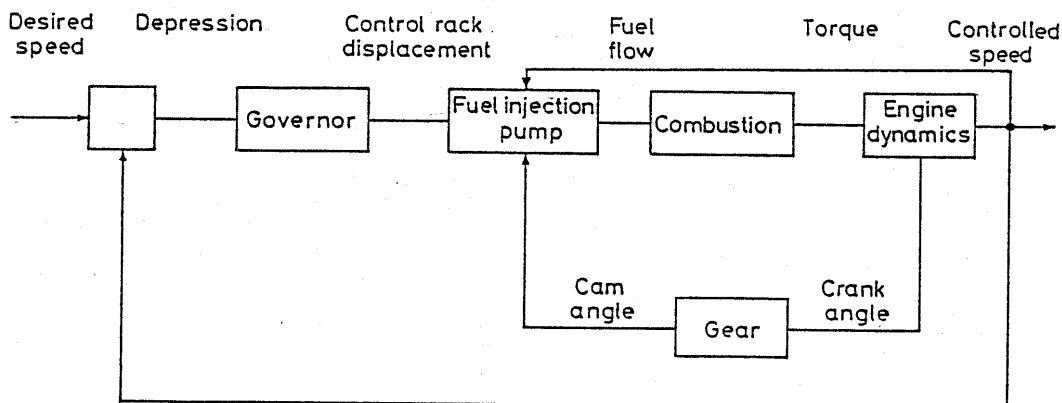Governor — Fuel injection pump — Combustion — Engine dynamics

Cam angle — Gear — Crank angle

Fig.1 Block diagram of the closed engine-governor loop.

From Eq.(7) and Eq.(8), we have

$$P_{xi-1}=\left[P_{xi-2}\frac{T_P}{\Delta T_{i-1}}-\alpha N_e\right]\bigg/\left[\frac{T_P}{\Delta T_{i-1}}+1\right] \quad (9)$$

where the following mean speed expression should be substituted for N_e:

$$N_{mi-1}=(N_{ei-1}+N_{ei})/2 \quad (10)$$

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

$$P=P_{xi-1}(1-\cos\omega_{mi-1}t) \quad (11)$$

where

$$\omega_{mi-1}=\frac{2\pi N_{mi-1}}{60}\times 2 \quad (12)$$

and the origin of t is set at the instant when the piston leaves its top dead center. Equation (11) is based on the experimental confirmation that the maximum

value of the governing pressure taken at the subventuri is nearly equal to the atmospheric one.

3.3 Motion of the fuel control rack
Let X denote the rack displacement from the position for no fuel delivery. Then the response of rack displacement to the governing pressure P can be written as
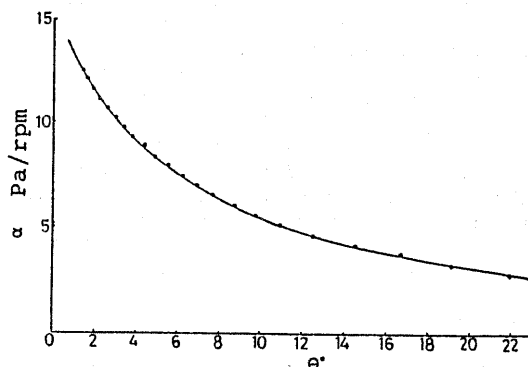
Fig.2 Subventuri pressure sensitivity versus the throttle valve opening.
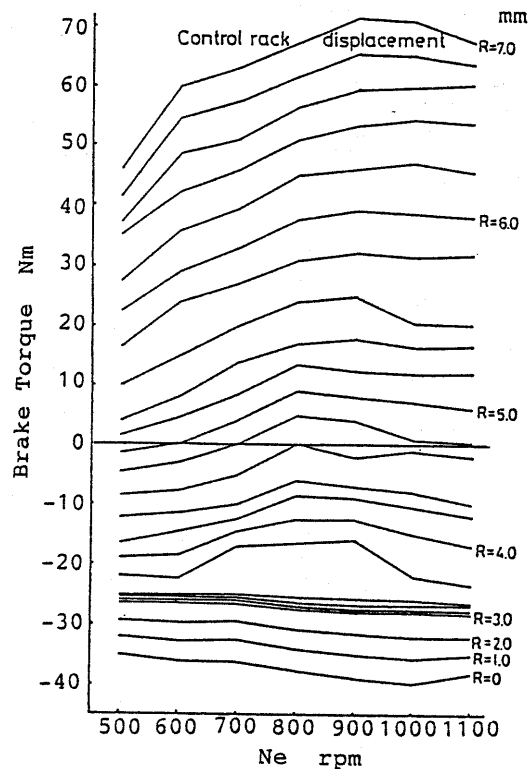
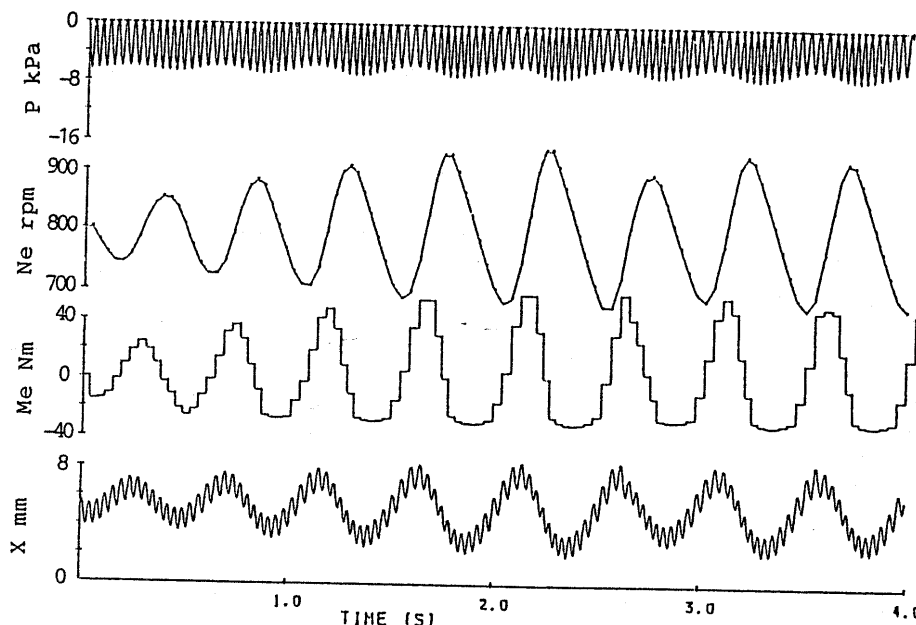Fig.3 Brake torque versus the engine speed relative to the rack displacement as a parameter.

Fig.4 Calculated transient behavior of the closed engine-governor loop (800 rpm).

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$$m_e\frac{d^2X}{dt^2}+C_e\frac{dX}{dt}+k(X+L_0-L)=A_d\cdot P \qquad (13)$$

where m_e is the equivalent mass of the 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 is the effective diaphragm area[8]. The equivalent viscous damping includes the effects of a liaison pipe and a chamber of reduced pressure. The rack displacement X_i at $t=t_{i-1}$ is derived by numerical integration of Eq.(13) combined with Eq.(11) and also with the initial values of X and dX/dt at $t=t_{i-1}$.

3.4 Torque and the rack displacement at the moment of injection

In the subroutine program is given a map of the measured torque as a function of the rack displacement and engine speed for finding out the torque M_{ei} corresponding to the values of X_i and N_{ei} by interpolation.
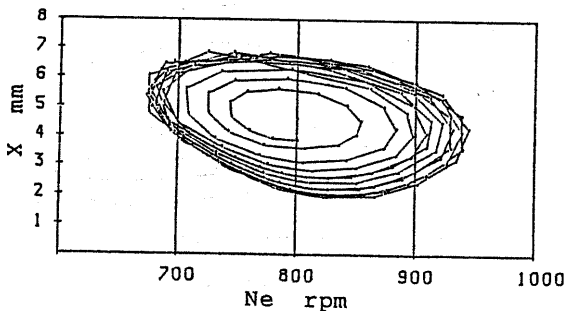
Fig.5 Limit cycle in a phase plane (800 rpm).

4. Simulation of the Low Speed Hunting

4.1 Numerical data for calculations

Given data are the moment of inertia of the crank shaft system I_e =0.261 kgm^2, the equivalent mass of the pneumatic governor system m_e=0.273 kg, the stiffness of the rack spring k=211 N/m, the equivalent damping coefficient C_e=24.9 Ns/m in the case of the liaison pipe 40 cm long , L=6.2 cm, L_0=2.59 cm, and A_d=21.0 cm^2 (8). A four-cylinder four-stroke engine of the swirl-chamber type with a total stroke volume amounting to 1986 cm^3 is used in combination with a Bosch-type individual fuel injection pump having plungers 6.5 mm in diameter, cam lift 8 mm high, and a diaphragm 60 mm in its outer diameter. The time constant T_P of the governing pressure response to the engine speed is derived from the experimental data[2] as

$$T_P=\frac{\tan \varphi}{2\pi f} \qquad (14)$$

where φ and f denote phase lag and hunting frequency respectively. Under the subventuri pressure control, the value of the phase lag φ of the reduced pressure relative to the speed fluctuation with a frequency of 2 Hz is 25~30°, Provided f=2 Hz and φ =30°, Eq.(14) gives T_P =0.05 s . The pressure sensitivity α versus the throttle valve opening is shown in Fig.2, and the brake torque versus the engine speed relative to the rack displacement as a parameter is shown in Fig.3.

4.2 Results of the simulation

Figure 4 shows the calculated transient behavior of the closed engine-governor loop responding to a small step increment of the throttle opening at an
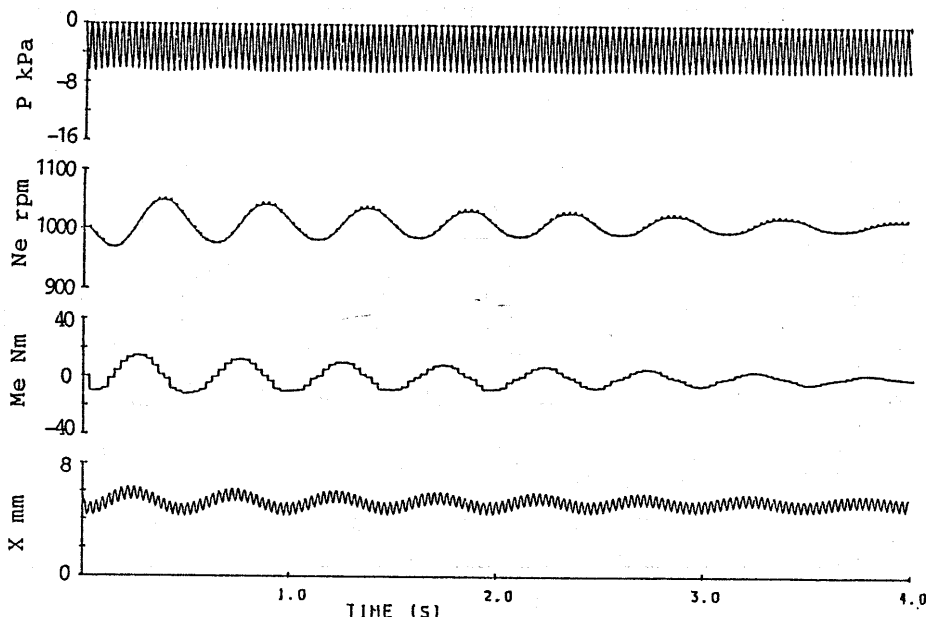

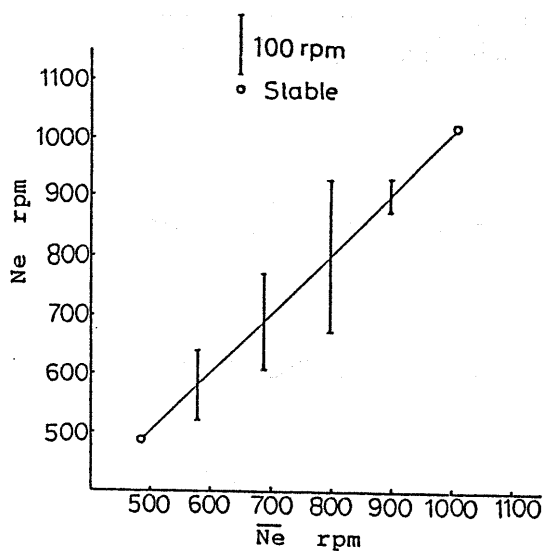Fig.6 Calculated transient behavior of the closed engine-governor loop (1000 rpm).

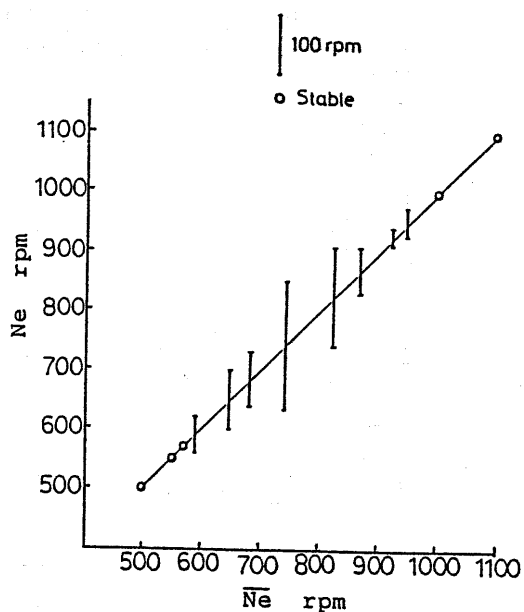Fig.7 Calculated amplitude of the engine speed variation.

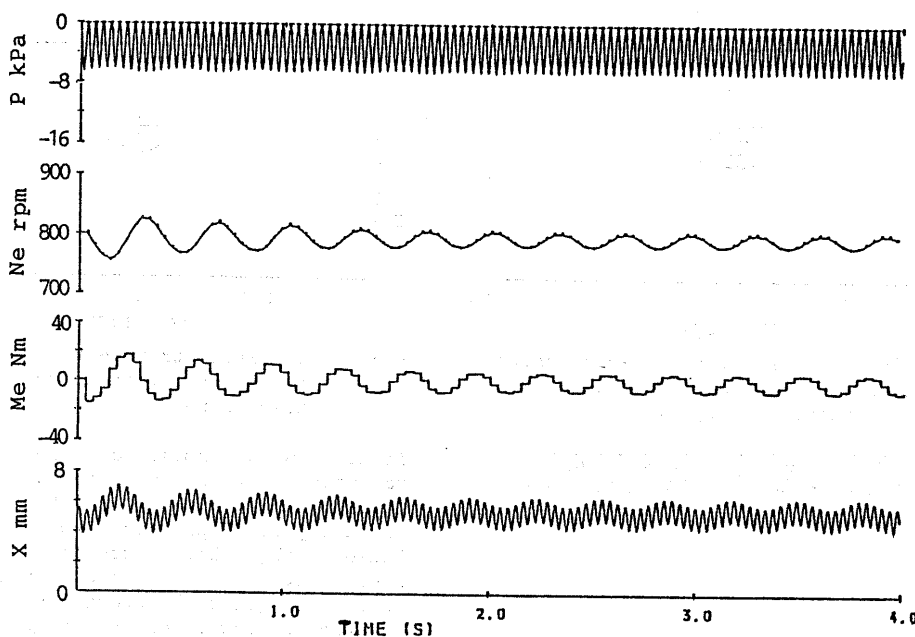Fig.8 Measured amplitude of the engine speed variation.

Fig.9 Calculated transient behavior of the closed engine-governor loop without consideration of time lag of the subventuri pressure (800 rpm).
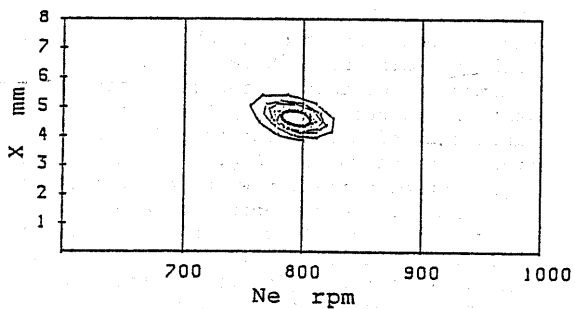
Fig.10 Calculated transient behavio without consideration of time lag of the subventuri pressure in a phase plane (800 rpm).
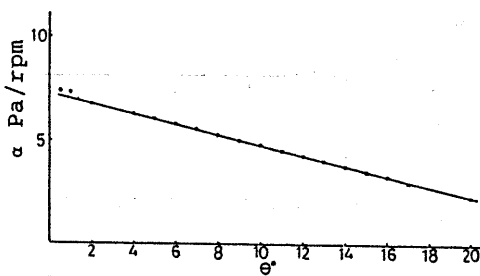
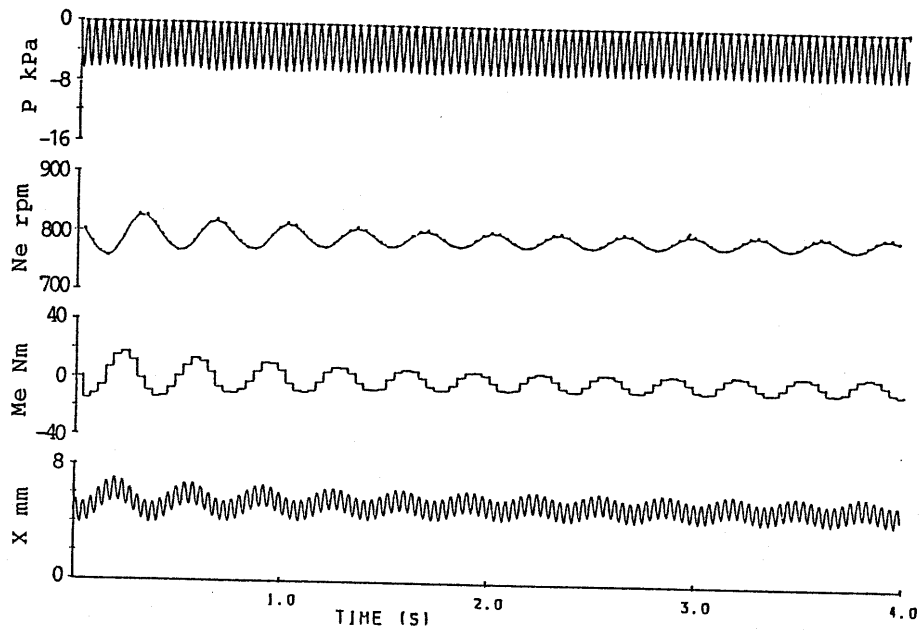Fig.11 Suction pressure sensitivity versus the throttle valve opening.

Fig.12 Calculated transient behavior of the closed engine-governor loop under the suction pressure control (800 rpm).
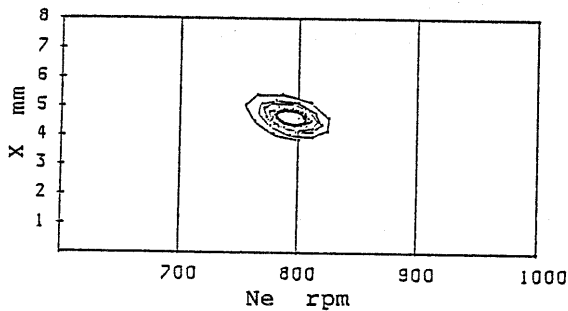

Fig.13 Calculated transient behavior under the suction pressure control in a phase plane (800 rpm).
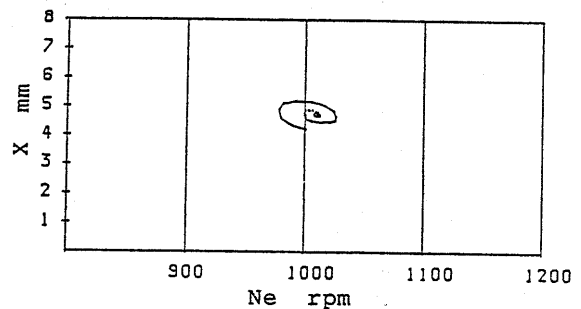

Fig.15 Calculated transient behavior under the suction pressure control in a phase plane (1000 rpm).
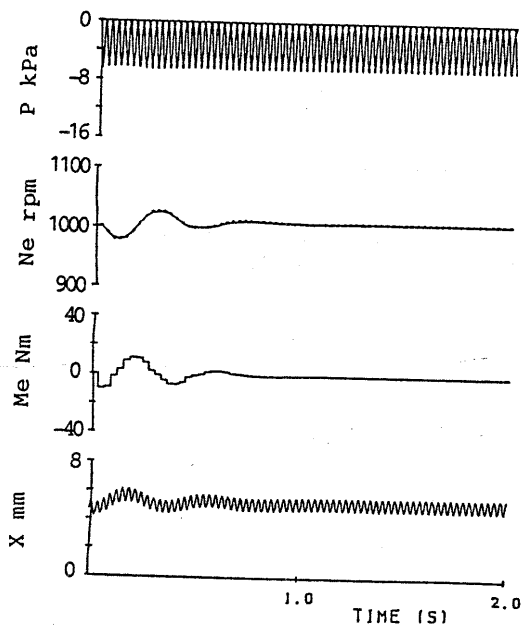

Fig.14 Calculated transient behavior under the suction pressure control (1000 rpm).

initial engine speed of 800 rpm. The variation of engine speed with an amplitude of 55 rpm at the beginning gradually develops into a sustained oscillation with a large constant amplitude, i.e. into a low speed hunting. The hunting frequency is about 2 Hz. If this behavior is rewritten on a phase plane plotting rack displacement at the moment of injection versus engine speed, the development of a limit cycle is clear as shown in Fig.5, where the amplitude of engine speed variation is about 125 rpm, while the difference between the values of maximum and minimum of the rack displacement is about 4 mm. The result of simulation of the idling at 1000 rpm is stable as shown in Fig.6 , where the engine speed variation with an amplitude of 40 rpm at the beginning converges into an amplitude of 5 rpm in about seven periods of oscillation. The calculated amplitude of the engine speed variation at each mean engine speed shown in Fig.7 fairly well agrees with the experimental one shown in Fig.8, while the calculated frequency is about 2 Hz in good agreement with the experimental values[1].

On the assumption of governing pressure without phase lag, calculations at 800 rpm are shown in Fig.9 and Fig.10. The engine speed variation with an amplitude of 35 rpm at the beginning converges into an amplitude of 10 rpm in about 2 seconds, which means no occurrence of hunting. As a result, it is necessary to take the phase lag into consideration.

5. Simulation of Idling under the Suction Pressure Control

Simulations are carried out also for the cases of suction pressure control where the phase lag is minimized by displacing the pressure source to the common intake duct just down the throttle valve and the subventuri. The data used in the calculations are the same as those of the subventuri pressure control except that $T_P=0$ and that the sensitivity of the governing pressure shown in Fig.11 is adopted. In Fig.12 and Fig.13 are shown the calculated results, in which the engine speed variation with an amplitude of 35 rpm at the beginning decreases to 10 rpm in 2 seconds without appearance of hunting. The results of the idling calculation at 1000 rpm are shown in Fig.14 and Fig.15, in which the transient behavior rapidly vanishes converging to the equilibrium state. The calculated idling speed under the suction pressure control is stable over a wide range, which agrees well with the experimental result [1].

6. Conclusions

The main results may be summarized as follows:

(1) A mathematical model which gives similar behaviors to those of the actual engine-governor loop has been formulated. With this simulation model the hunting phenomenon of a pneumatically governed compression-ignition engine has been investigated using a digital computer. Comparison between the results of simulation and the experiments proved the adequateness of the model.

(2) Hunting frequency, amplitude of shaft speed variation at hunting and shaft speed range where hunting occurs as the result of simulation for the case of subventuri pressure control were in good agreement with the experimental ones. An example of transient development of shaft speed hunting from a small disturbance was shown.

(3) Hunting does not occur under the suction pressure control, which also agrees well with the actual system.

Thus, it was confirmed that the low speed hunting is a self-excited oscillation ascribed to the phase lag of the reduced pressure at the subventuri. In the 4th report, the effect of simulated venturi diameter on phase lag of the governing pressure response will be investigated.

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