DYNAMIC COMPUTER SIMULATION OF AN ENGINE INTAKE AND CARBURETION SYSTEM

Mont Hubbard
University of California, Davis, California

ABSTRACT

Automatic control of spark advance and fuel-air ratio in production automobiles may prove to be a key to the solution of gasoline engine economy and exhaust emissions. Steady-state models exist which predict how economy and emissions are affected by engine operating variables, but dynamical models, which describe how fuel-air ratio changes with sudden changes in engine speed and throttle, are needed before such control systems can become a reality.

This paper describes the computer simulation of an engine intake and carburetion system and describes how the transient fuel-air ratio variable can be predicted. The intake system is modelled as a one-dimensional compressible fluid flow with sonic conditions possible at the carburetor throat and the intake valve. Fuel flow through the carburetor is taken to be incompressible. The program is flexible enough to incorporate different types of carburetors and hence allows comparison of dynamic characteristics of different types. Time histories of intake manifold pressure and fuel-air ratio are outputs of the program.

NOMENCLATURE

A area

C_c area contraction coefficient

C_b velocity correction coefficient

c local speed of sound

c_p specific heat

f friction factor

f/a fuel-air ratio

g acceleration due to gravity

γ ratio of specific heats

k loss coefficient

L length

p pressure

Q volumetric flow rate

ρ density

T temperature

t time

Subscripts

a atmospheric

d downstream

dc computed downstream

f fuel

i i^{th} intake station

j jet

ℓ loss

o orifice

t venturi throat

u upstream

V valve

INTRODUCTION

A very critical problem which daily affects most people in the United States is that of toxic exhaust emissions. The major mobile source of these emissions is the four-stroke, gasoline-powered, internal combustion engine. During the last fifteen years, much research effort has been expended to understand the emission formation processes and to develop methods of control. In addition, significant legislative progress has been made (e.g., the Clean Air Act) and the effects of emission control programs on air quality are beginning to be felt.

Most of the research done to date in the field of exhaust emissions has been for steady-state engine operation. In addition, most of the adjustment of variable parameters (e.g., carburetor idle adjustment) affecting engine operation and emissions is done under steady-state conditions. The combustion process, although difficult to simulate exactly, is, generally speaking well understood and the basic physical factors in the formation of exhaust emissions are well known. Models exist (1,2) which predict exhaust emissions during steady-state operation as a function of speed, load, equivalence ratio and spark advance.

Measurements of engine loads during actual operation of passenger car engines, however, have shown that, except for idling, steady-state operation occurs very rarely. Instead, the engine operates almost exclusively under transient conditions experiencing more or less continuous changes in engine speed, load, or both (3). As a result, most of the California and Federal Driving Cycles specified for emission testing new automobiles are periods of transient operation (acceleration or deceleration), because the automobile population typically operates under these condition.

Unfortunately, it is not possible to use the steady-state models mentioned above to predict emissions during transient engine operation. During a sudden change in throttle, such as would occur at the beginning of a period of vehicle acceleration, the intake manifold pressure can change by a factor of two or three in several milliseconds. Steady-state models are unable to account for the effects of such fast changes on engine operation. Beachley and Frank (4) have found that even though their steady-state emissions models suffice for prediction of NO_x, they are incapable of accurately predicting levels of HC and CO during transient operation.

It is widely believed that one of the major effects of transient engine operation on emissions is that of dynamic variations in fuel-air ratio entering the cylinder. Shown in Fig. 1 is the general effect of equivalence ratio(fuel-air ratio normalized to stoichiometric) on emissions. For example, the figure shows clearly that a 10% shift in fuel-air ratio from lean to rich operation near stoichiometric (ϕ=1.0) can double or triple CO emissions. Thus it seems that, in order to predict emissions accurately during transient engine operation, one must first be able to predict the effects of transient engine operation on fuel-air ratio.

This paper is a first attempt to explain the effect of dynamic engine operation on the fueling and intake system. A model is presented which predicts dynamic variations in fuel-air ratio through real time simulation of the fluid (air and fuel) flows in the carburetor, throttle and intake manifold. Such a model is useful not only for its predictive capability, but also because a fundamental understanding of the reasons for transient degradation might allow design changes in present engine systems or new emission prevention techniques to be developed.

Very little has been done in the area of modeling the intake and carburetion system in order to better understand and predict the transient fuel-air ratio variations mentioned above. A comprehensive analysis of steady-state carburetor metering was done by Harrington (5). Fundamental fluid mechanics and thermodynamics were the basis for a computer simulation program and experimental data were used to evaluate the predictions. However, his results add little insight into the causes of transient fuel-air ratio variations in a carburetor-fueled engine. Tanaka (6) attempts to model the engine fuel-air transient response purely by liquid fuel flowing along the inlet pipe. He obtains reasonable experimental correlation between his predicted and measured time responses but his model neglects the dynamics of the carburetor entirely.

ELEMENTS OF MODEL

Figure 2 is a schematic diagram of the induction system for a representative single-cylinder, four-stroke engine including the cylinder, intake port, valve and manifold, a throttle and a simplified carburetor. Although the system obviously becomes more complicated for the multi-cylinder case, the model for the induction system above can be generalized easily and represent the first step in a more general theory of intake modeling.

When the piston is near top dead center (TDC) after the exhaust stroke, the intake valve opens and the piston begins to move down, decreasing cylinder pressure and drawing the fresh charge into the cylinder. Normally, cylinder pressure is close to atmospheric pressure at the end of the exhaust stroke (7). A complete model including combustion, expansion and exhaust fluid flow would generate the cylinder pressure as an output. For simplicity, however, the pressure in the cylinder is simply initialized to atmospheric when the intake valve opens.

The area of the intake valve is normally significantly less than the cross-sectional area of the intake manifold and thus restricts the flow into the cylinder. Near the bottom of the intake stroke, the intake valve closes and the cylinder charge is isolated from the intake system. The momentum of the flow, however, causes pressures just upstream of the closed intake valve to rise momentarily and oscillations in pressures and flows occur throughout the intake system until the intake valve is opened for the next cycle. Intake manifold pressure is not exactly equal to the cylinder pressure when the intake valve closes as there can be signigicant charging due to the velocity head, especially when the intake valve is closed late in the cycle. This fluid inertia effect increases as the diameter of the manifold decreases and the average velocity of the manifold gases increases. The average airflow is regulated by changing throttle area.

Shown in Fig. 3 is an extremely simple version of a carburetor. At the cross section of minimum area in the venturi is the tip of a fuel nozzle which comes from a gasoline float bowl whose level is maintained at some constant height. As the air flow rate through the venturi increases the pressure at the throat of the venturi decreases. Since this pressure is less than atmospheric which exists at the top of the float bowl, gasoline flows through the jet into the airstream. The base of the fuel jet contains a small orifice the size of which is a controlling factor in fuel flow rate. For given nozzle design, however, the fuel flow rate is determined solely by the pressure variations in the throat of the venturi.

At this point it should be mentioned that more complicated carburetors could be coupled to the intake system and they still would be driven only by the intake pressure variations mentioned above. In particular, idling circuits for low loads and other compensating fuel jets which operate under two-phase flow conditions could be included. The above simple carburetor, however, was chosen because it illustrates clearly the effects of transients in the venturi flows and pressures on the fuel flow rate.

INTAKE MODEL

The entire intake system is modeled as a one-dimensional compressible fluid flow. Although the mass of the intake system is actually distributed, Margolis (8) has proposed a method of approximation (using a finite number of lumped control volumes) in the solution of a similar one-dimensional distributed system, the exhaust pipe of a two-stroke engine. The details of the intake model presented below follow closely the methods outlined in (8), and therefore a complete rederivation of the details of the model is not given here. The main elements are summarized for completeness.

A discretized version of the continuity equation for a representative control volume such as is shown in Fig. 4 is given by

$$\rho_i \, Q_i - \rho_{i+1} \, Q_{i+1} = AL \frac{d\rho}{dt} \qquad [1]$$

Assuming isentropic flow and local Mach numbers much less than 1.0, [1] reduces to

$$Q_i - Q_{i+1} = \frac{AL}{c^2} \frac{dp}{dt} \qquad [2]$$

The above equation neglects convective transport terms. This can be corrected by the inclusion of a "Bernoulli resistor" (9) which allows the steady-state pressure drops to be predicted as well as the wave effects.

A similar discretized version of the momentum equation yields

$$A(p_i - p_{i+1} - k\,Q^2) = L\rho \frac{dQ}{dt} \qquad [3]$$

where the third term represents the loss associated with flow through a pipe and

$$k = \frac{\rho f L}{2A^2 D} \qquad [4]$$

Thus [2] and [3] are first-order differential equations for the two-state variables inside the i^{th} lumped mass; volume flow rate and pressure.

The throttle and intake valve are modeled as isentropic nozzles and thus allow for choked flow to occur. The steady-state flow through such a nozzle can be derived to be

$$Q_V = \left[2A_V^{\,2} \, c_p \, T_u \left(1 - \left(\frac{p_d}{p_u}\right)^{\frac{\gamma-1}{\gamma}} \right) \right]^{1/2} \qquad [5]$$

Since no flow can change instantaneously, an inertia was added to the nozzles to allow the flow to accelerate gradually to the steady-state flow given by [5]. This acceleration was computed by

$$\frac{dQ_V}{dt} = \frac{A_V}{\rho_V L_V} (p_{dc} - p_d) \qquad [6]$$

where p_{dc} is the downstream pressure which would exist at the present values of flow and upstream pressure and is given by

$$p_{dc} = p_d \left[1 - \frac{Q_V^{\,2}}{2A_V^{\,2} c_p T_u} \right]^{\frac{\gamma}{\gamma-1}} \qquad [7]$$

When the critical pressure ratio (0.53) is reached, the choked flow depends only on upstream conditions and the nozzle area until the downstream pressure again becomes large enough to unchoke the flow.

CARBURETOR MODEL

Since gasoline is essentially incompressible the entire jet is modeled as a lumped control volume and the momentum equation can be written for the fluid inside the jet. Figure 4 is a diagram of the jet. The momentum equation for the jet is written as

$$\rho_f L_j \frac{dQ_f}{dt} = A_{eff} \left[p_a + \rho_f g(h_1 - h_2) - p_{\ell_1} - p_{\ell_2} - p_t \right] \quad [8]$$

where p_{ℓ_1} and p_{ℓ_2} are the pressure losses along the jet and across the nozzle and are given by

$$p_{\ell_1} = f \frac{L_j}{D_j} \rho_f \frac{Q_f^2}{2A_j^2} \quad [9]$$

and

$$p_{\ell_2} = \frac{\rho_f Q_f^2}{2C_c^2 C_V^2 A_0^2} \quad [10]$$

Here A_{eff} is the effective area over which the pressures act and is in the range: $A_0 < A_{eff} < A_j$.

The instantaneous fuel-air ratio (f/a) at the throat of the venturi is given by

$$f/a = \frac{\rho_f Q_f}{\rho_t Q_t} \quad [11]$$

NUMERICAL RESULTS

The model presented above was exercised using engine parameters representative of a single-cylinder CFR engine together with the simplified carburetor model described above. Figure 5 is a plot of cylinder pressure and mass of the cylinder charge versus crank angle. Note that the mass flow rate into the cylinder had not reached a maximum when the valve was closed; in other words, at this particular speed and throttle setting, the valve was closed too early for optimum volumetric efficiency. It is obvious that similar simulation results over a set of speeds and throttle settings representative of the engine operating regime could be used to determine the effect of valve timing on volumetric efficiency and therefore to choose optimum valve timing. In addition the simulation can be used to determine the effects on volumetric efficiency of such other engine parameters as valve lift, intake length and diameter, compression ratio and cylinder volume.

Pressures at different points in the intake are plotted in Fig. 6 for three values of crank angle. The propagation and reflection of the pressure wave is clearly shown by the three succeeding pressure traces (here spaced at time intervals $\Delta t = 0.001$ sec).

Shown in Fig. 7 is a plot of the dynamic variation in fuel flow rate at the venturi versus crank angle. Note that there is significant variation in time around some average value. If these variations persist into the cylinder they can produce variations in fuel-air ratio from cycle to cycle and even spatial gradients in fuel-air ratio inside the cylinder.

The simulations shown in Figs. 5-7 were run on a Burroughs B6700 computer. At an engine speed of 3600 RPM and integration step size of 10^{-5} sec, the simulation of one complete engine cycle required approximately five minutes of processer time.

CONCLUSIONS

A model has been demonstrated which predicts dynamic changes in fuel-air ratio. The results demonstrate the effects of pressure waves in the intake on fuel-flow rate and hence on instantaneous fuel-air ratio.

ACKNOWLEDGEMENTS

The author is grateful to Donald Margolis for fruitful discussions concerning the model and to Richard Sterrett for his help in coding and debugging the computer program.

REFERENCES

1. M. Hubbard, "Applications of Automatic Control to Internal Combustion Engines", Ph.D. dissertation, Stanford University, June 1975.

2. M. Hubbard, P. D. Dobson and J. D. Powell, "Closed Loop Control of Internal Combustion Engine Efficiency and Exhaust Emissions", SUDAAR #493, Final Report, DOT Contract No. DOT-OS-3011, May 1975.

3. H. D. Koller, "Motor-Fastkollektive and Betriebszustands Kollektive von LKW, Omnibus and PKW in Fahrbetrieb", Deutsche Kraftfahrforschung und Strassenverkehrstechnik, Vol. 225, VDI-Verlag, 1972.

4. N. H. Beachley and A. A. Frank, "Increased Fuel Economy in Transportation Systems by Use of Energy Management", DOT Final Report, Contract No. DOT-OS-30112, December 1974.

5. D. L. Harrington, "Analysis and Digital Simulation of Carburetor Metering", Ph.D. thesis, University of Michigan, September 1968.

6. M. Tanaka, "Transient Response of a Carburetor Engine", presented at the DOT Automobile Engine Control Workshop, Cambridge, Massachusetts, July 8-9, 1975.

7. Jennings, B. H. and Obert, E. F., "Internal Combustion Engines Analysis and Practice", International Textbook Company, Scranton, Pennsylvania, 1944.

8. D. L. Margolis, "Modeling of Two-Stroke Internal Combustion Engine Dynamics Using the Bond Graph Technique", SAE paper 750860 presented at SAE Off-Highway Vehicle Meeting, September 1975.

9. D. C. Karnopp and R. C. Rosenberg, "System Dynamics: A Unified Approach", John Wiley and Sons, New York, 1975.

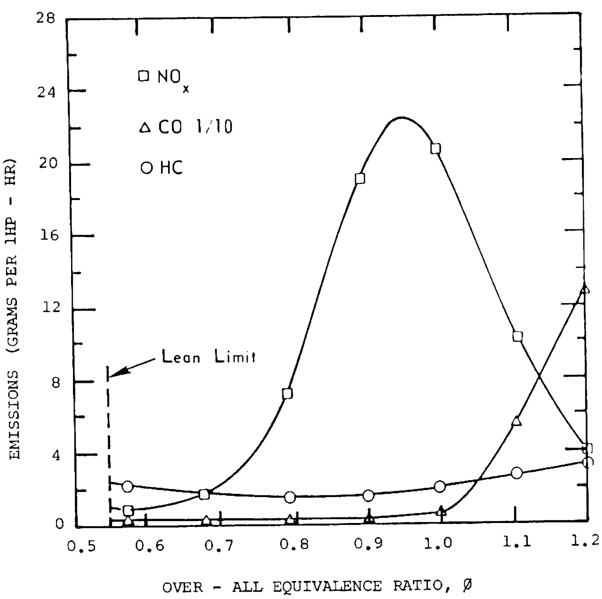

Fig. 1 Emissions vs. Equivalence Ratio

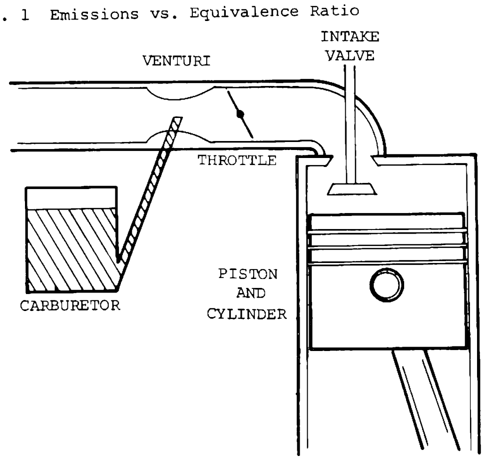

Fig. 2 Schematic Diagram of Induction System

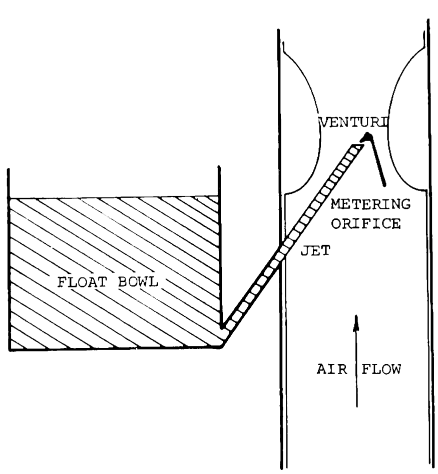

Fig. 3 Simple Carburetor

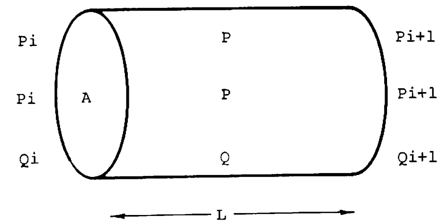

Fig. 4 Lumped Intake Element

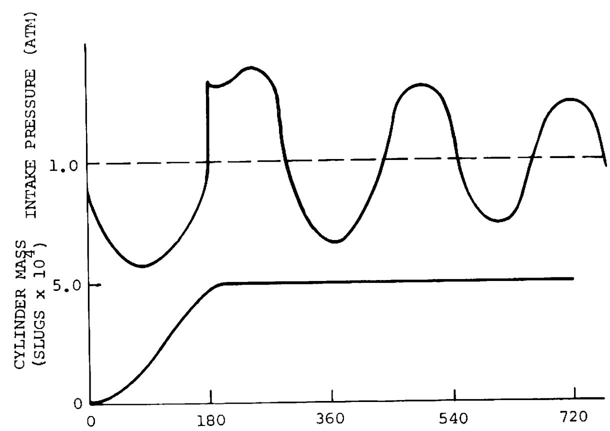

Fig. 5

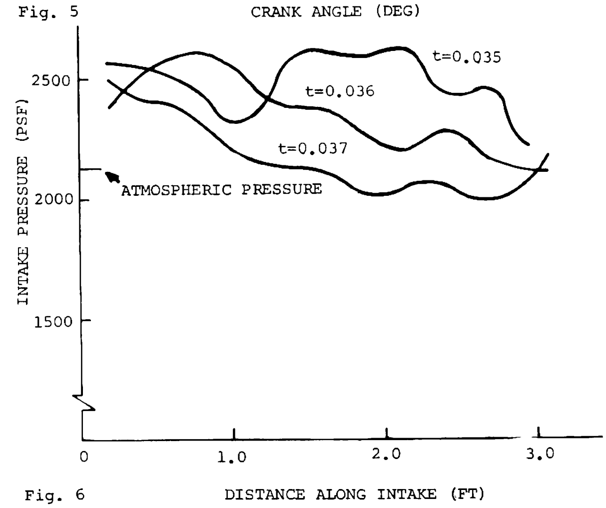

Fig. 6

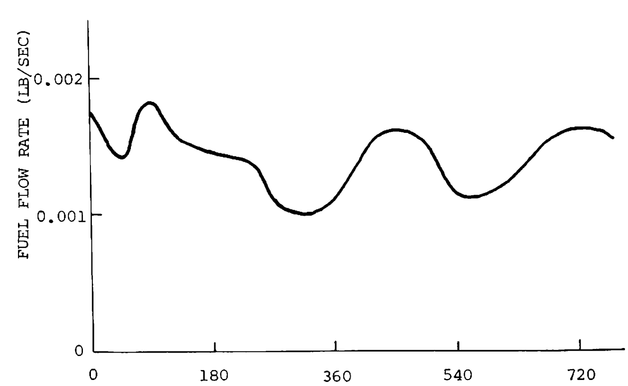

Fig. 7