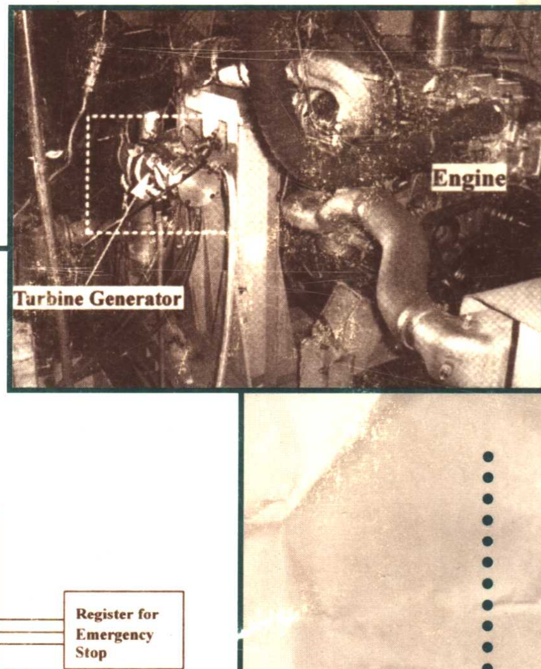
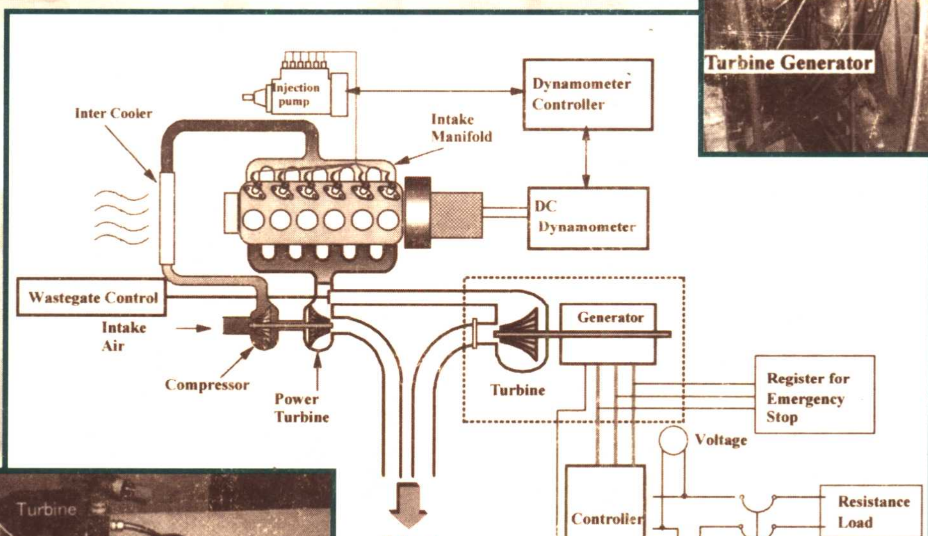
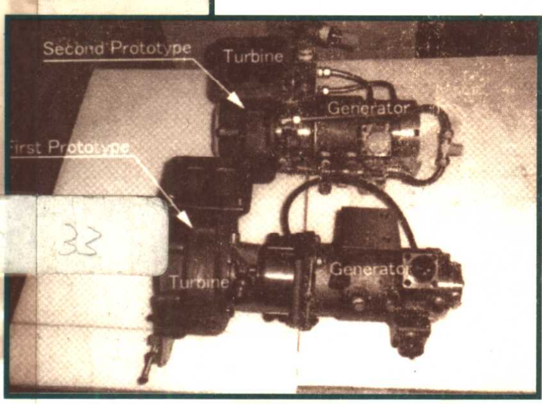
Technology for Electric and Hybrid Vehicles







SAE INTERNATIONAL

SP-1331

Technology for Electric and Hybrid Vehicles

SP-1331

Published by:
Society of Automotive Engineers, Inc.
400 Commonwealth Drive
Warrendale, PA 15096-0001
USA
Phone: (724) 776-4841
Fax: (724) 776-5760
February 1998

ISBN 0-7680-0151-X
SAE/SP-98/1331
Library of Congress Catalog Card Number: 97-81283

Printed in USA

PREFACE

This Special Publication, Technology for Electric and Hybrid Vehicles (SP-1331), is a collection of papers from the "Electric Vehicle Technology" and "Engines and Fuel Technology for Hybrid Vehicles" sessions of the 1998 SAE International Congress and Exposition.

Hybrid vehicles are now a reality in Japan, and they could soon be coming to the United States. The heart of the Toyota Prius hybrid vehicle is its fuel-efficient engine and unique transmission, coupled with a limited-range battery. The hybrid vehicle's advantage is its ability to run the engine at its "sweet spot" to minimize emissions of criteria pollutants or minimize energy consumption and CO2 production, depending on the control strategy. The key technical measure of success for a hybrid vehicle is a well designed engine--electrical-battery system that is matched to the load demand.

The papers from the "Engines and Fuel Technology for Hybrid Vehicles" session focus on leading-edge engine design, engine management, and fuel strategies for low emission, high mileage hybrid cars and commercial vehicles.

The papers from the "Electric Vehicle Technology" session focus on hybrid vehicle control technology, energy storage, and management for hybrid vehicles and simulation development.

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An Algorithm of Optimum Torque Control for Hybrid Vehicle

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Abstract

An algorithm for a fuel efficient hybrid drivetrain control system that can attain fewer exhaust emissions and higher fuel economy was investigated. The system integrates a lean burn engine with high supercharging, an exhaust gas recycle system, an electric machine for power assist, and an electronically controlled gear transmission. Smooth switching of the power source, the air-fuel ratio, pressure ratio, exhaust gas ratio as a function of the target torque were analyzed. The estimation of air mass in cylinder by using an air flow meter was investegated to control the air-fuel ratio precisely during transients.

1.INTRODUCTION

Consumers are inceasing their demands for vehicles that are more fuel efficient, environmentally friendly, and affordable. Some form of electric and hybrid vehicle is increasingly being viewed as one answer to user demands. Thus, introduction of a future car system integrating an internal combustion engine and an electric machine seems inevitable [1]. While electric vehicles and hybrid-electric vehicles are still dominant [2-4], conventional hybrid systems, with their complicated energy management and storage systems, may not be the final answer to the ultimate high-mileage, low-emissions passenger vehicle. Many of today's hybrid and electric designs are simply too complex, heavy and costly to be considered a viable supercar-type vehicle.

Minimal hybridization will present the best solution to the low-emitting, high-economy passenger vehicle of the future [1]. The internal combustion engine will continue to dominate the world passenger-vehicle market for at least the next 25 years [5]. The engine will require better efficiency and lowered emissions output which means that fuels will similarly require refinement, so that the engines can eventually be refined to the point that they produce almost no harmful emissions. Internal combustion engines using synthetic fuel made of natural gas, similar to light quality gasoline, seem to be the most promising advancement in the near future [5].

To reduce the system's cost and increase its efficiency, the engine is driven at the lowest possible speed at the maximum gear ratio of the transmission at low vehicle speed. Thus, the capacity of the electric machine and battery can be kept small. Systems that combine an integrated interactive hybrid drivetrain control system, such as to give lean burn, with an electronically controlled transmission, and electric machine control systems mentioned above, have been partially examined [1]. The optimum combination of two power sources-a hybrid drivetrain with an internal combustion engine and a small electric machine-would make it possible to get significant reductions in fuel consumption and exhaust emissions. The control system without the electric machine has been already investigated [6], as well as the control system with the electric machine and a continuously variable transmission [7]. A control system with the electric machine and simple gear transmission was presented [8]. A concept for an advanced hybrid control system was investegated that combined a high supercharging engine and a small electric machine [9]. In this paper, an algorithm for an advanced hybrid drivetrain control system that combines the engine drivetrain control system and electric machine systems is investegated.

2. SYSTEM CONCEPT

2.1 Outline

Idealized, the concept would include an engine and electric machine drivetrain, such as in Figure 1 and Table 1. The electric machine is usually functioned as an electric motor. On one hand this provides fuel saving and lower exhaust emissions while using the engine system such as direct injection stratified charge sytem [10], or rapid combustion system with high dispersed fuel-air mixture [11] and high supercharging, on the other hand, it allows for short-distance driving and low load driving with the electric machine such as electrically exited synchronous drive with power inverter. The engine brake, wheel brake, and regeneration by the electric machine are controlled optimally during deceleration and downhill travel. A transmission with electronically controlled synchromesh gear sets is used for this purpose.

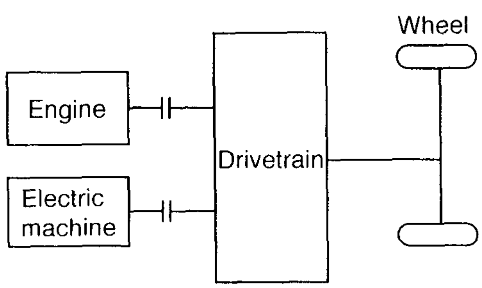


Fig. 1 Hybrid drivetrain

Table 1 Hybrid drivetrain

(1) Engine
(a) Rapid combustion with high dispersed mixture
(b) High supercharging
→ lower nitrogen oxides emissions

(2) Electric machine
Electrically exited synchronous drive with power inverter
→ short distance and low load driving

(3) Transmission
Electronically controlled synchromesh gear set
→ lower power loss

2.2 Basic control technique

The aim of the control system is to obtain a smooth drivetrain force change relative to the torque set point, which is given by the accelerator position, over a wide range of vehicle speeds and loads. The system should be able to cope with large changes in engine load and drivetrain switches from the electric machine to the engine and from the engine to the electric machine without increasing nitrogen oxides emissions, and without degrading driveability.

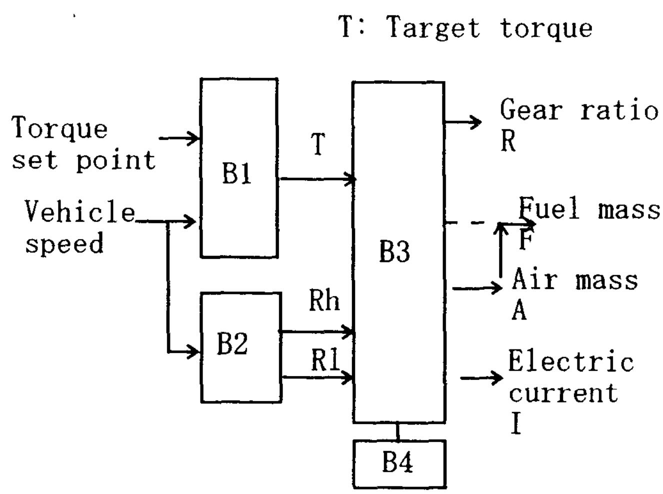


Fig. 2 Control system

As shown in Figure 2, the target drivetrain torque T is calculated as a function of the torque set point and the vehicle speed in block1 (B1). The upper and lower limits of the equivalent gear ratio Rh and Rl are calculated as a function of the vehicle speed in B2. The equivalent gear ratio R, fuel mass F, and air mass A are simultaneously calculated as a function of the target torque T, taking the limit Rh and Rl in consideration in B3. Some control strategies such as the dynamic compensation described in section 3.7 are executed in B4. Fuel mass F is delivered with an electronically controlled fuel injector as shown elsewhere [10]. Fuel is injected directly into the cylinders. Therefore, the system is free from transient fuel compensation which is commonly in port injection systems [8]. The air mass A is controlled with an electronically controlled throttle valve and air bypass valve as described later. The fuel mass may be set by the target torque directly, as in diesel engines. But the estimation of the air mass by the accelerator position is not accurate. Therefore,the fuel mass is controlled by the air mass which is generally measured by the air flow meter, to control the air-fuel ratio precisely [12].

Under stratified charge conditions, the accelerator pedal opening angle, rather than the intake manifold pressure, is the most important information for determining the quantity of injected fuel. But, information about the amount of intake air has also importance in actual engine operation to control the air-fuel ratio A/F precisely.

The gear ratio R is controlled with an electronically controlled transmission [13-16]. The command electronics for an electrically excited synchronous drive can be easily accomplished. In the case of a synchronous drive, an inverter provides optimal control of rotor excitation, stator current amplitude I, and stator current phase. A power inverter with insulated gate bipolar transistors transforms the battery voltage into the rotating voltage system for the motor driving of the electric machine. An additional chopper controls the DC current for the rotor. Then, the drivetrain output torque is obtained, which is equal to the target torque T if there are neither calculation nor control errors.

2.3 Air-fuel ratio control

As the target torque T increases, the drivetrain switched from the electric machine to the engine. The fuel mass F, air mass A of the engine and the electric current I are changed stepwise.

The air mass and fuel mass of the engine are changed frequently as the target torque changes. The air-fuel ratio must be controlled during the transient conditions precisely to reduce exhaust emissions and improve driveability. It was determined that the volumetric efficiency during and immediately following a transient, at any engine temperature, was not equal to the staedy-state value. The transient volumetric efficiency was found to be as large as 10% different from the steady-state value. The volumetric efficiency is dependent upon instantaneous cylinder wall and valve temperature. To control the the air-fuel ratio A/F during transients accurately, the engine controller needs precise predictions or measurements of the amount of intake air, and the amount of fuel injected that will go directly in-cylinder [12].

The intake system of the engine is equipped with a compressor for supercharging and an exhaust gas recycle system, as shown in Figure 3. Wt, Wc, Wb, Wh, Wr, and We are the air or gas mass flow rate at the upstream throttle valve, at the outlet of the compressor, at the bypass valve, at the downstream throttle valve, at the exhaust gas recycle valve, and at the intake port of the engine, respectively. In conventional engine control systems, the air flow into the cylinders should be predicted based on the movement of the throttle plate [12]. The air intake process is modeled through the manifold abusolute pressure observer model. The observer is based on the estimated throttle opening [12]. With stratified

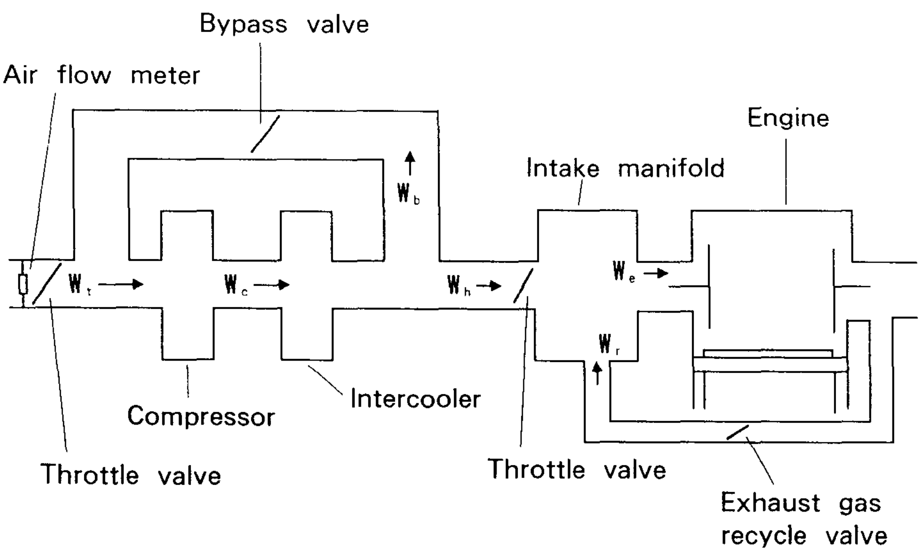


Fig. 3 Intake system

Table 2 Calculation conditions

Type	4-stroke 4 cylinder
Cylinder volume per cylinder	4×10^{-4} m^3
Maximum air mass per cylinder at atmospheric pressure A0	4.8×10^{-4} kg
Maximum exhaust gas recycle ratio	40 %
Maximum exhaust recycle mass per cylinder G0	1.9×10^{-4} kg
Sum of A0 and G0 Gv0	6.7×10^{-4} kg
Maximum fuel mass per cylinder at atmospheric pressure F0	4.8×10^{-5} kg
Maximum air-fuel ratio	40
Atmospheric pressure	9.8×10^{4} Pa
Maximum pressure ratio of compressor	2
Torque T	$1.92\times10^{6}\times F-19.2$ Nm
Cylinder volume per cylinder	4×10^{-4} m^3
Thermal efficiency of engine	30 %
Maximum output power of electric machine	10.5 kW

charge engines, nearly unthrottled operation is realized. Under these conditions, the estimation of the air flow based on movement of the throttle plate is not accurate due to the small pressure differential accross the throttle plate. Therefore, the model based on the air flow meter was investigated in this paper.

3. ANALYSIS

3.1 Simulation conditions

A 4 cylinder, 4-stroke engine with a cylinder volume of 4×10^{-4} m^3 was used for testing. The engine was equipped with a direct injection stratified charge system [10], a supercharger and an exhaust gas recycle (EGR) system. The air-fuel ratio A/F was set between 11 and 40. The maximum ratio of the EGR was 40 %. The maximum pressure ratio of the superchager was 2. The air mass A was controlled by opening and closing of the throttle valve or the bypass valve in Figure 3. The relevant gear ratios from 1st-5th for a stepped transmission were 3.5, 2.0, 1.3, 1.0 and 0.73, respectively. Fuel mass F was controlled with electronically controlled fuel injectors. Table 2 shows the calculation conditions. The output power of the electric machine was 10.5 kW, and the torque was 50 Nm at the speed of 2000 rpm.

3.2 Smooth switching of power source

As the target torque T increases in Figure 4, the drivetrain switches from the electric machine to the engine. The switching is carried out by simultaneously decreasing the power of the electric machine and increasing the engine power. At T=50 Nm in Figure 4, the power source is switched from the electric machine to the engine. The fuel mass F, air mass A and the exhaust recylce mass G are increased stepwise simultaneously to keep the air-fuel ratio 15 and the EGR ratio 40 %. As the target torque T increases further, the supercharger starts, the air mass A is increased more than A0, and the EGR mass is also increased. At T=162 Nm, the air mass A and the EGR mass G becomes doubled,

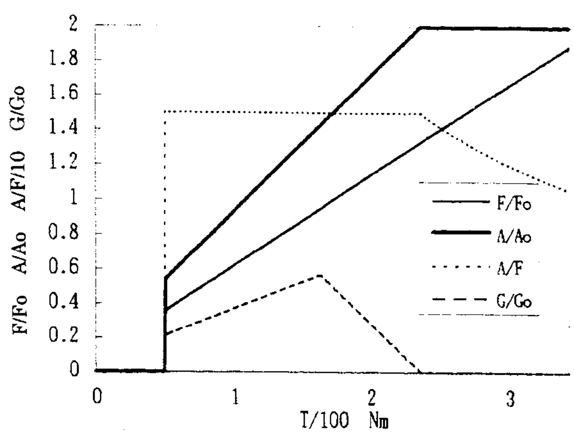


Fig.4 Fuel mass F,air mass A and EGR mass G versus target torque

which is limited by the pressure ratio of the supercharger. At T=243 Nm, the EGR mass G is decreased and the air mass A is doubled. When the target torque increases further, A/F becomes lower than 15, and the air mass must be controlled by using the throttle valve and the bypass valve.

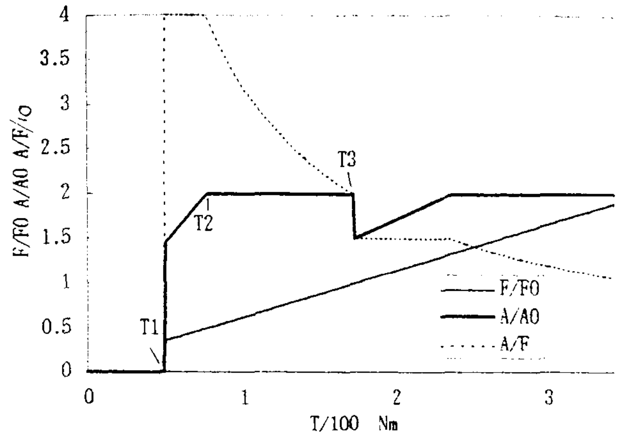
3.3 Lean burn control by supercharging

Figures 5 (a), (b) and (c) show the simulation results with high supercharging and lean burn. When the target torque is more than T1= 50 Nm, the power source switched from the electric machine to the engine. When the air mass ratio A/A0 becomes more than 1 , the supercharge starts, the air mass ratio A/A0 is finally doubled. In Figure 5(a), at T=50 Nm, the supercharger starts simultaneously with the switching to the engine. When the target torque becomes higher than T2, The air-fuel ratio A/F becomes lower than 40. In Figure 5(b), the supercharger starts at T= 78 Nm. The air-fuel ratio A/F is increased temporally from 20 to 40. When the target torque becomes T3, the air mass A is decreased by decreasing the air-fuel ratio from 20 to 15 stepwise ,without passing into the high nitorogen oxide emission region.

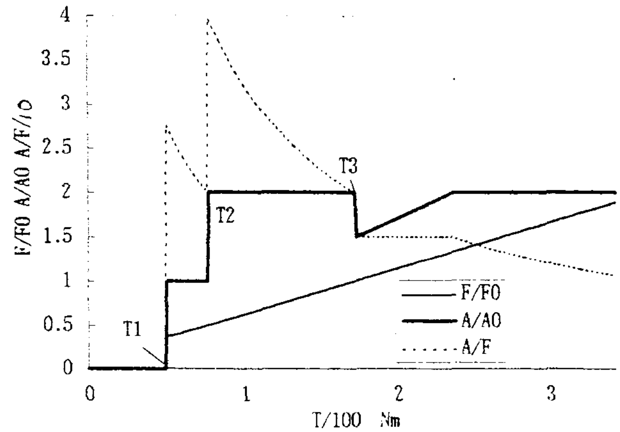
Figure 5 (c) shows the result with high supercharging when pressure is controlled by the bypass valve proportionally to keep the air-fuel ratio at 15. When the target torque becomes T2, the air mass is decreased slightly to decrease the air-fuel ratio from 20 to 15 by controlling the throttle valve. When the target torque increases further, the supercharger starts again and the pressure is controlled by the bypass valve.

3.4 Smooth gear shift with exhaust gas recycle control

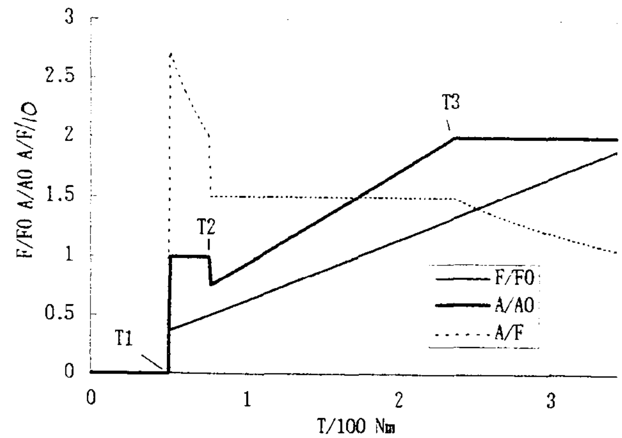
Figures 6 (a)-(d) show the results when the gear is shifted from 4th to 2nd at the target torques are Tg=100 Nm, 170 Nm, 238 Nm, and 300 Nm, respectively. The engine torque must be changed simultaneously, so that the output torque remains the same during the shift operation. The engine torque is controlled by decreasing the mass of fuel. The air mass and EGR mass are decreased simultaneously to keep the air-fuel ratio at 15 and EGR ratio at 40%. The air mass is controlled by opening and closing the



(a) Early supercharging



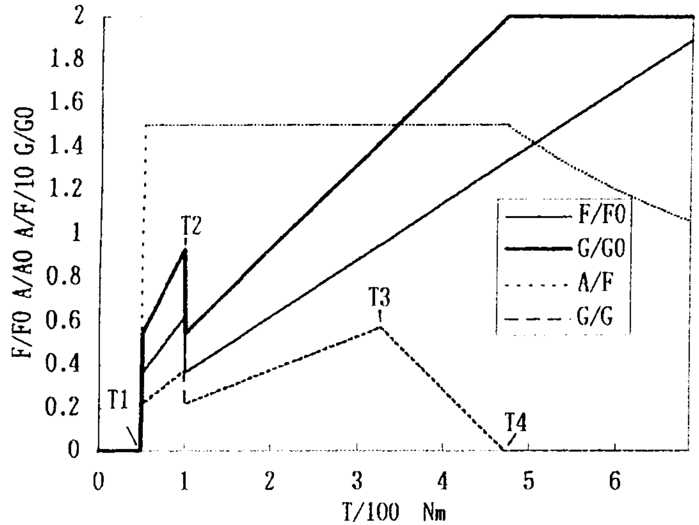
(b) Late supercharging



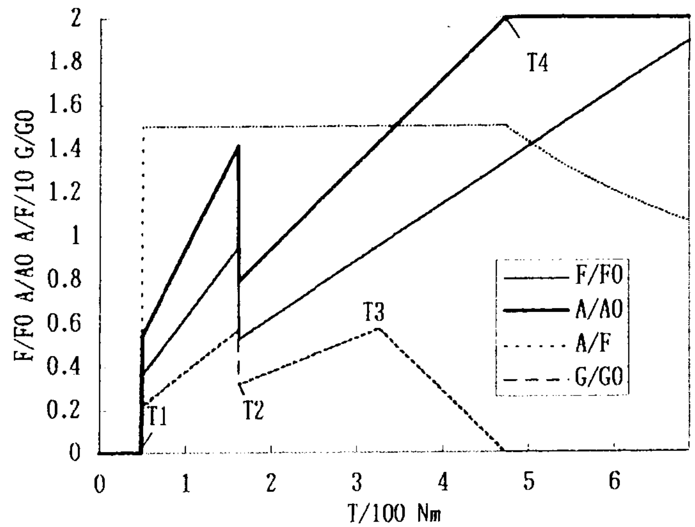
(c) Proportional supercharging

Fig. 5 Fuel mass F,air mass A and EGR mass G versus target torque T

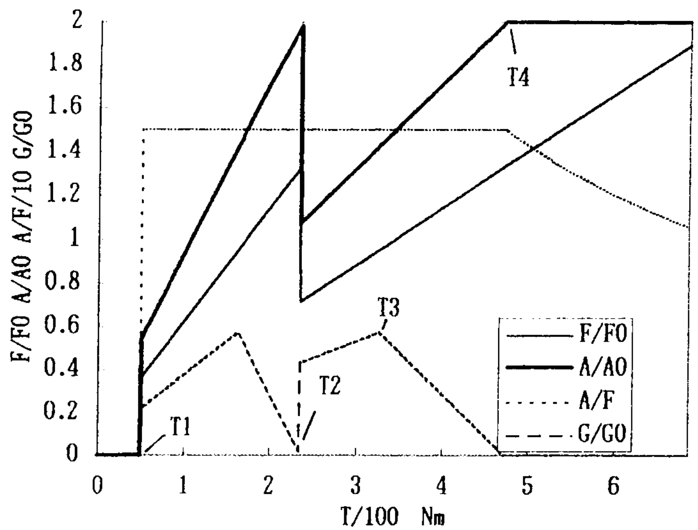
throttle valve and the bypass valve. When the target torque becomes higer than T3 (Figures 6 (a)-(c)), the EGR mass G decreases. When the target torque T



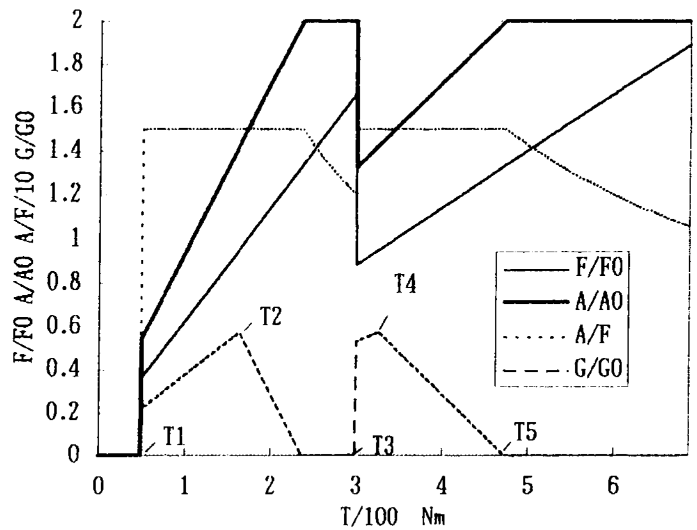
(a) Tg= 100 Nm



(b) Tg= 170 Nm



(c) Tg= 238 Nm

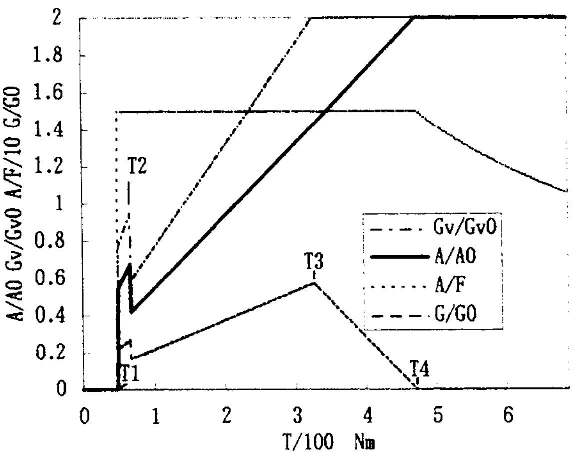


(d) Tg= 300 Nm

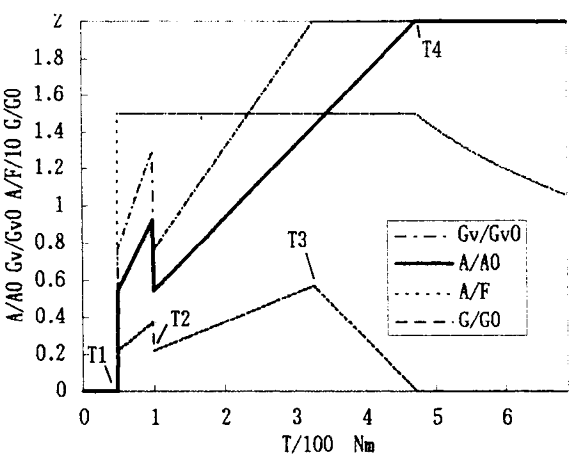
Fig. 6 Fuel mass F, air mass A, EGR mass
and air-fuel ratio A/F versus traget torque T

becomes higher than T4 (Figures 6 (a)-(c)), the air-fuel ratio becomes lower than 15. As the target torque at the gear shift Tg becomes higher, the region of supercharging increases. In Figure 6 (d),the air-fuel ratio becomes less than 15 at T=230-300 Nm, resulting in the increase of carbon monoxide emission.

Figures 7 (a) and (b) show the total mass Gv (the sum of air mass and EGR mass) as a function of the target torque T. Tg is the target torque at gear shift .The gear is shifted from 4th to 2nd. The total mass ratio Gv/Gv0 is lowered when the Tg becomes lower. Thus, the target torque T when the supercharger starts . , becomes higher. When Tg is 100 Nm, the supercharger starts at the target torque T of less than 100 Nm.



(a) Tg= 70 Nm

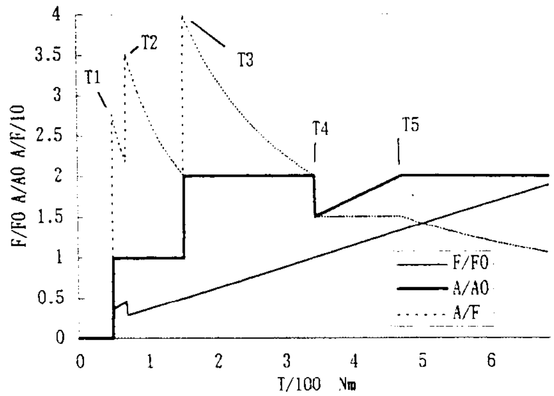


(b) Tg= 100 Nm

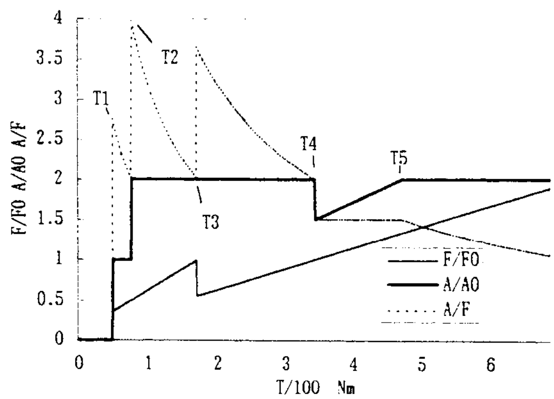
Fig. 7 Total mass Gv as a function of the target torque T

3.5 Smooth gear shift with lean burn control

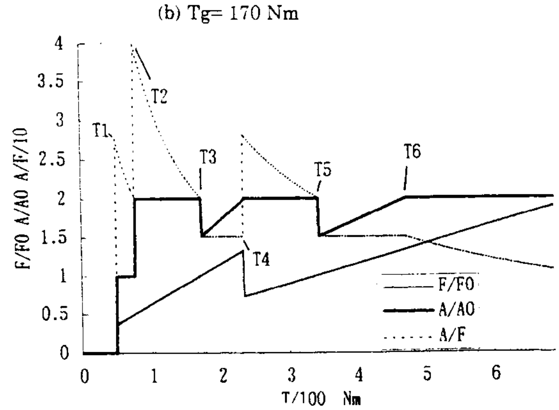
Figures 8 (a)-(c) show the results when the gear is shifted from 4th to 2nd at the target torque Tg=70 Nm, 170 Nm and 238 Nm, respectively. As Tg becomes higher, the target torque when the superchager starts becomes lower. The engine torque must be changed so that the output torque remains the same during the shift operation. The engine torque can be controlled by controlling the fuel mass only. The air mass remains the same during the shift operation.



(a) Tg= 70 Nm



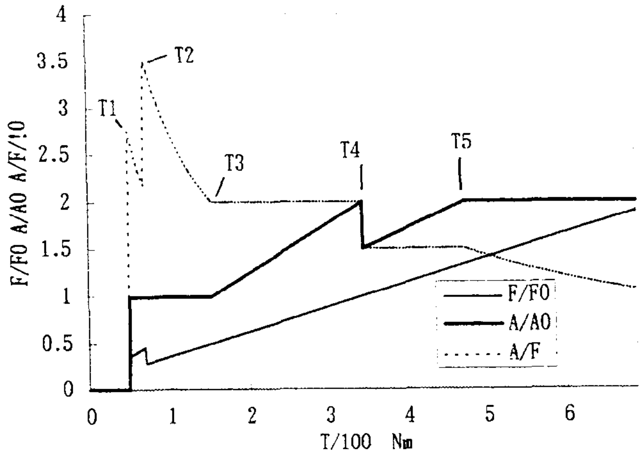
(b) Tg= 170 Nm



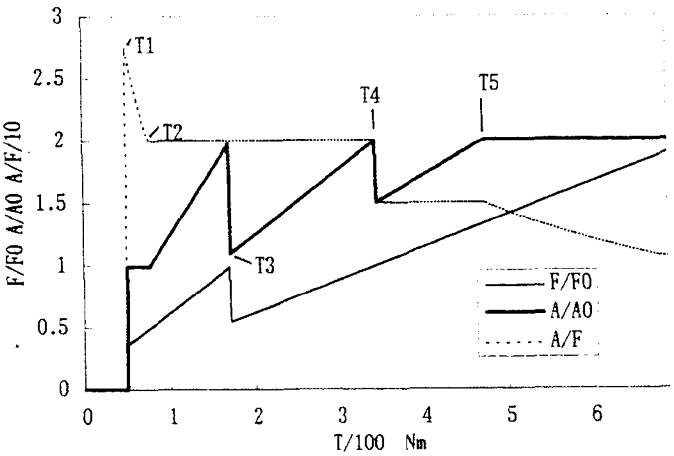
(c) Tg= 238 Nm

Fig. 8 Fuel mass F, air mass A, EGR mass as a function of the target torque T

Figures 9 (a) and (b) show the results when gear is shifted from 4th to 2nd at the target torque Tg=70 Nm, 171 Nm, respectively. As the target torque at gear shift Tg becomes higher, the region of supercharging becomes wider, the region of the air-fuel ratio A/F of more than 20 becomes narrower.



(a) Tg= 70 Nm



(b) Tg= 170 Nm

Fig. 9 Fuel mass F, air mass A, EGR mass as a function of the target torque T

3.6 Air-fuel ratio control

As mentioned before, the fuel mass is controlled by the air mass which is generally measured by the air flow meter. During switching from the electric machine to the engine, the air mass increases stepwise against the target torque T to maintain the air-fuel ratio adequately. The accuracy of maintaining the air-fuel ratio for operation during transient conditions depends on the accuracy of the estimated air mass entering the engine cylinders. It is necessary to estimate this air mass in advance of fuel injection timing and before placing the fuel in the cylinders. In case of gasoline direct injection the fuel injection is free from compensation for the fueling dynamics. When the capacity of the compressor, the surge tank and the intercooler in the intake system [17] is larger in Figure 3, the air mass going through the air flow meter increases temporarily to fill the surge tank and intercooler during throttle opening and the compressor starting. The air mass must be compensated also according to the response lag of the air flow meter and the filling lag of the EGR.

The filling spike must be compensated to reduce fluctuation of the air-fuel ratio which is apt to increase exhaust emissions. This compensation is attained by using the aerodynamic model of the intake system [13]. The air mass into the cylinder is calculated by using the intake manifold filling dynamics and the compressor dynamics. Then, the model of the intake system to predict future air mass is applied. Some simulation results, obtained by the method mentioned above, are shown in Figure 10 which has samples of the traces for air mass flow rate Wa, Wc, Wb, Wh, and We and the pressure pi (10^5 Pa) at the intercooler when the compressor is started during 0-0.2 s and the bypass valve is closed during 0.3-0.4 s. Wa is the measuring value by the air flow meter. The estimated air mass, We, is close to the air mass

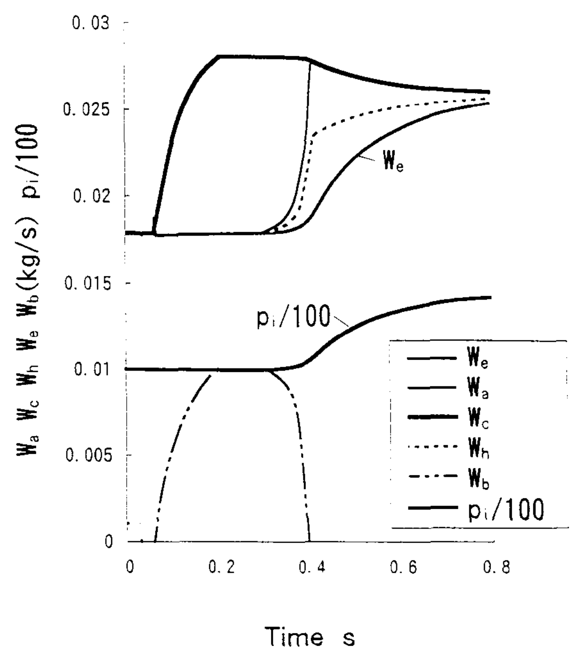


Fig. 10 Air flow during bypass valve control

entering the cylinder. It is seen that the air mass can be estimated at the beginning of the intake stroke, resulting in the fuel supply without any delay, thus a precise air-fuel ratio control.

3.7 Control strategies

(1) Dymamic compensation

Good acceleration performance will require some modification for the strategy mentioned in section 3.6. The output torque is reduced during the change in gear ratio and the power source (engine, electric machine) because part of the engine torque is used to accelerate the engine itself. The power is controlled by compensating the fuel mass and air mass through dynamic models, resulting in smooth switching and reduced drivetrain vibration.

(2) Damping of the torque oscillation

The torque of the engine is controlled during acceleration by monitoring differences between engine and wheel speeds in order to provide a measure of phase differences in the event of engine torque oscillating. The difference between the engine and wheel signals is integrated and fed to an ignition angle correction circuit in order to damp out the torque oscillations [18]. Or, the fuel mass is compensated in place of the ignition angle. In the case of the transmission without power shift, the oscillation will increase when the fuel mass is increased stepwise after the gear engagement. The fuel mass is controlled, taking the oscillation into consideration or, the control is executed by estimating the torque by engine speed signal [19].

(3)Block of the shift

The control system compares and calculates the difference between two stationary torques calculated at different times. Then it recognizes whether the vehicle is on the crest of a hill or in a valley, and whether the vehicle is traveling uphill or downhill over the crest of the hill or through a valley. The gear shift is blocked according to this information to avoid frequent gear shifts.

(4) Integration of brake control

Full utilization of regenerative braking increases electric machine operating range by 25% [20]. The engine brake, wheel brake, and regeneration by the electric machine are controlled optimally during downhill travel. However, available regenerative braking methods vary for several operating conditions such as state of battery charge and vehicle speed. Regeneration cannot be executed when the battery charge is full. Integration of brake and drivetrain controls allows maximum energy recovery with minimal friction braking. This dynamic interaction between the vehicle's brake and drivetrain systems also improves driveability.

(5) Power assist with electric machine

The electric machine is connected between the engine and transmission or between the transmission and the wheel. In the former, the electric machine can assist synchronization of the gear sets during the shift operation in the synchromesh transmission. The operating force becomes unnecessary, and the friction cones for synchronism are eliminated. Therefore, the gear shift mechanism in the synchromesh transmission becomes very simple. In the latter, the electric machine can assist power supply to the wheel during the shift operation in the synchromesh transmission, resulting in better driveability.

3.8 Future outlook

The system mentioned above is a concept, although some components and subsystems have been tested already. More detailed analysis would be conducted to demonstrate the magnitude of the efficiency gains by introducing all components and subsystems in a testing vehicle.

4. SUMMARY

A fuel efficient engine drivetrain control system was proposed which combines a lean burn engine with a supercharging, an exhaust gas recycle, an electric machine for power assist, and an electronically controlled trasmission. The smooth switching of power source, the lean burn control with supercharging, the smooth gear shift with exhaust gas recycle control and lean burn control, and the air-fuel ratio control by using an air flow meter were analyzed to attain better fuel economy and better driveability.

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Energy Regeneration of Heavy Duty Diesel Powered Vehicles

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ABSTRACT

The objective of this study is to improve fuel economy and reduce carbon dioxide emissions in diesel-electric hybrid automotive powertrains by developing an exhaust gas turbine generator system which utilizes exhaust gas energy from the turbocharger waste gate.

The design of the exhaust gas turbine generator was based on a conventional turbocharger for a direct-injection diesel engine.

Data from steady-state bench tests using air indicates about 50% of the turbine input energy can be converted to electric energy. Turbine generator output averaged 3 kW, while a maximum of about 6 kW was observed. Based on this data, we estimate that energy consumption in a vehicle could be reduced between 5% and 10%.

Engine tests were conducted under both steady-state and transient conditions. These tests revealed that optimal performance occurred under high-speed, high-load conditions, typical of highway or uphill driving, and that performance at low-speed, low-loads was relatively poor.

The efficiency at low engine speeds could be improved by controlling the inlet flow to the turbine generator.

INTRODUCTION

In the automotive field, many effort has been made to reduce carbon dioxide (CO_2) emission by reducing fuel consumption with combustion improvement of the engine or reducing power train and driving resistance.

Thermal efficiency has been pursued to almost its maximum in the case of automotive diesel engines and further improvement of fuel consumption by means of engine modification seems to be very difficult.

However, only 20 to 30% of total fuel energy consumed is used for net engine power output and the remainder was wasted. To achieve the notable fuel consumption improvement and consequent CO_2 reduction, recovering these wasted energies, such as kinetic energy at deceleration and exhaust gas heat energy etc., and converting (regenerating) them to the energy useful for vehicle driving source is required.

In this effort, an exhaust heat energy regeneration technique has been experimentally studied with an exhaust turbine generator system.

APPROACH

Today, energy regeneration systems of diesel powered vehicles where a part of the wasted energy is regenerated and used as auxiliary power source are made practicable in Japan as pressure storage type hybrid systems and a diesel-electric hybrid system (**HIMR**: Hybrid Inverter controlled Motor & Retarder System)[1].

However, these systems regenerate only decelerating energy mechanically or electrically and uses it as auxiliary power source. Therefore, these system can fulfill its function only under driving conditions with high in frequency of acceleration and deceleration and may suitable under urban driving conditions which has relatively low average speed.

This study aims at the improvement of the total energy consumption efficiency by regenerating wasted energies mainly at high speed and heavy load engine operating regions such as highway or long uphill driving and adding them to the existing hybrid system.

In this study, we consider regenerating exhaust heat energy to electric energy, which is the most convenient energy to use, and then applying them to HIMR system. A turbine generator system, in which a generator is driven by a exhaust power turbine, was developed for exhaust energy regeneration and conversion to electric energy. At first, amount of the energy which can be