

Fuzzy-PID Speed Control of Diesel Engine Based on Load Estimation

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Abstract

In order to improve the anti-disturbance performance of engine-load and the effect on speed control for the diesel engine, the paper presents the fuzzy-PID speed control strategy in the architecture of torque-based control. The engine-load estimation algorithm is designed based on the mean-value-model and crankshaft dynamics model, and the estimation precision is validated by engine test in both steady and dynamic conditions.

Through the experiment verification of the diesel engine, the fuzzy-PID control strategy based on load estimation can significantly improve the anti-disturbance performance of engine-load in the speed control. Through the vehicle-simulation in the acceleration-deceleration condition and the circle condition, this paper verifies that the fuzzy-PID control strategy based on load estimation can keep the rapid adjustment of engine-speed control and the good pedal-speed following performance; this method can also avoid repeated calibration of control parameters and reduce the occurrence of the unexpected shift.

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Introduction

The engine dynamic characteristics are changed with the variation of engine speed and load, showing the non-linear characteristic, therefore, variable parameters in PID speed control strategy has been widely used [1,2,3]. In order to reduce the fluctuation of the engine speed, many papers designed the correction algorithm of dynamic gain based on the variable PID control [2-3] when the road resistance and the power consumption of accessories changed. However, since its trigger condition was the difference value of the target and actual speeds without considering the load change, it increased the quantity of calibration in the test bench.

The engine load of the heavy vehicle in the process of acceleration increases with the pump wheel speed of the torque converter increasing, which also increases the adjustment time of engine speed, resulting in the worse control effect on the calibrated PID parameters. Therefore, while driving in bad road conditions, drivers need to repeatedly adjust the accelerator pedal to achieve the desired speed, but the frequent change of the accelerator pedal causes the misjudgment of shift logic, contrary to the anticipated shift of the driver [4-5].

Therefore, this paper considered the control idea of variable PID parameters with load changing, and designed the full-range speed control algorithm of the fuzzy-PID specific for the diesel engine of heavy vehicles. Due to the high cost of the direct measurement of engine load, engine load needs to be estimated, and there are two main methods: First, the use of vehicles and road information estimates reversely the engine load [6], however, the road resistance of the heavy vehicle changes violently so that this method is difficult

to apply; Second, it builds the load estimation model constituted of engine measurable parameters. The model meets the demand for the simple structure and easy calculation [7-8], and the heavy vehicle requirement of maintaining high estimation precision in violent load changing.

Taking a heavy vehicle powertrain as the research object, this paper built the engine load estimation model which was based on mean-value-model and crankshaft dynamics model, and the estimation precision was studied in both steady and dynamic conditions. Given by experimental verification in fixed speed and variable load conditions, this paper analyzed the anti-disturbance ability of engine load in traditional PID speed control and fuzzy-PID speed control. Through the vehicle simulation in the acceleration condition and the circle condition, this paper verified the more rapid effect of pedal-speed following performance in fuzzy-PID control method based on torque estimation, and this method can avoid repeated calibration of control parameters and reduce the occurrence of an unexpected shift.

1. The Method Study of Engine-Load Estimation

1.1. Load Estimation Method and Parameter Calibration

Considering the crank dynamics model, the estimated load of engine can be calculated by the following equation:

$$\hat{T}_l = \hat{T}_i - \hat{T}_f - I_e \dot{n}_e$$



Where $\hat{T}_l$, $\hat{T}_i$ and $\hat{T}_f$ respectively represent the load torque, the average indicated torque and the friction torque; I_e is the inertia of the engine; $\dot{n}_e$ is the average acceleration of engine.

The average indicated torque can be calculated by the Equation (2), Where $\dot{m}_f$ is the circulatory fuel-injection-quantity of the engine, H_u represents the low calorific value of the diesel, which is 43.2 MJ/kg; η_{it} is the indicated thermal efficiency of the engine.

$$T_i = 9550 \dot{m}_f H_u \eta_{it} / 120000 \quad (2)$$

In this paper, the object is an in-line pump turbocharged diesel engine, so the circulatory fuel-injection-quantity can be calculated by pump characteristics in Equation (3), where $\dot{m}_f$ is the circulatory fuel-injection-quantity, P_r is the oil-gear-lever position of the in-line pump, and n_e is the engine speed.

$$\dot{m}_f = f_Q(n_e, P_r) \quad (3)$$

The indicated thermal efficiency in the steady state of the diesel engine is closely related to the load and speed, and the indicated thermal efficiency in the dynamic condition is also affected by other parameters, such as the air-fuel ratio [9], which is difficult to calibrate respectively due to the relationship of these parameters. Therefore, this paper used Equation (4) (where η_{it} is the indicated thermal efficiency and α is the air-fuel) to calculate the indicated thermal efficiency, on account of this method: the diesel engines is required by changing the load in the fixed speed to obtain a corresponding relationship between the indicated thermal efficiency and the air-fuel ratio [9-10]. So this relationship can reflect the impact of load on the indicated thermal efficiency in the steady condition, and approximatively reflect the impact of air-fuel ratio on the indicated thermal efficiency in the dynamic condition. The Accuracy of the method is verified by the experiment in Section 1.2.

$$\eta_{it} = f_{\eta i}(n_e, \alpha) \quad (4)$$

In order to calculate the air-fuel ratio, it can calculate the intake air-flow by Equation (5):

$$\dot{m}_\alpha = \frac{\eta_v V_d n_e P_{im}}{120 R T_{im}} \quad (5)$$

Where $\dot{m}_\alpha$ is the intake air-flow; η_v is the charging efficiency; V_d is the engine displacement; P_{im} is the intake pressure; R is the thermodynamic parameter, and values 287 J/kg/K; T_{im} is the intake air-temperature.

The charging efficiency η_v introduced by Equation (7) can be calculated by the engine speed and the intake pressure in Equation (6):

$$\eta_v = f_{\eta v}(P_{im}, n_e) \quad (6)$$

For the turbocharged diesel engine, the friction torque is associated with not only the engine speed, but also the average effective pressure in cylinder and the pressure difference in the intake and the exhaust pipe. To consider the influence of the gas path on the mechanical efficiency, this paper calculated the friction torque by Equation (7) which is based on the engine speed and the intake pressure:

$$T_f = f_f(n_e, P_{im}) \quad (7)$$

With Equation (1), (2), (3), (4), (5), (6) and (7), the load can be estimated based on the accelerated speed is given by Equation (8):

$$\hat{T}_l = H_u f_Q(n_e, P_r) f_{\eta i} \left[\frac{f_{\eta v}(n_e, P_{im}) P_{im} V_d}{f_Q(n_e, P_r) R T_{im}}, n_e \right] - f_f(n_e, P_{im}) - I_e \dot{n}_e \quad (8)$$

Where H_u , R , V_d and I_e are constants, and the oil-gear-lever position P_r , the intake air-pressure P_{im} and the engine speed n_e can be measured in real-time. The pump characteristic $f_Q(n_e, P_r)$ and the volumetric efficiency characteristic $f_{\eta v}(n_e, P_{im})$ are calibrated in the engine speed controller [11], and this paper does not repeat it, but calibrates the indicated thermal efficiency and the friction torque in each conditions.

In this paper, the selected test engine is a turbocharged diesel engine of the 6-cylinder in-line pump and 330kw maximum power value, and other equipments are selected include the electric-eddy-current dynamometer, the Kistler cylinder pressure sensor, the torque sensor, intake pressure and temperature sensors, the combustion analyzer, the encoder, the air-flow meter, and the fuel consumption instrument. By changing the load at a fixed speed to obtain correspondence between the air-fuel ratio and indicated thermal efficiency, this paper calibrated the indicated thermal efficiency in the test engine shown in Figure 1.

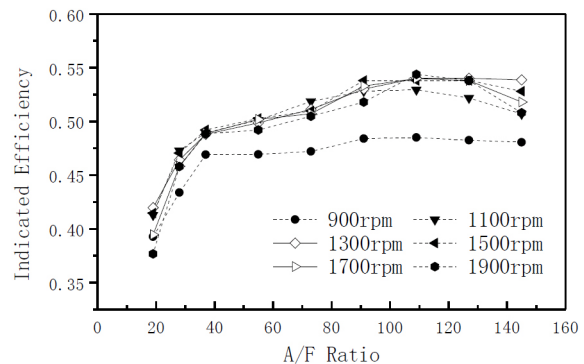


Figure 1. The relationship of indicated thermal efficiency with the engine speed and air-fuel ratio

With using the cylinder pressure sensor to measure the cylinder pressure, this paper calculates indicated torque value. With using the torque sensor to measure the effective torque value, the friction torque value can be obtained by subtracting the two corresponding values. Since the coolant and oil temperature will have a great impact on the friction torque, the calibration work of this paper is under

80-85 °C temperature of the coolant. In the condition of the heated engine, the final calibration of the relationship among friction torque, engine speed and intake pressure is shown in [Figure 2](#).

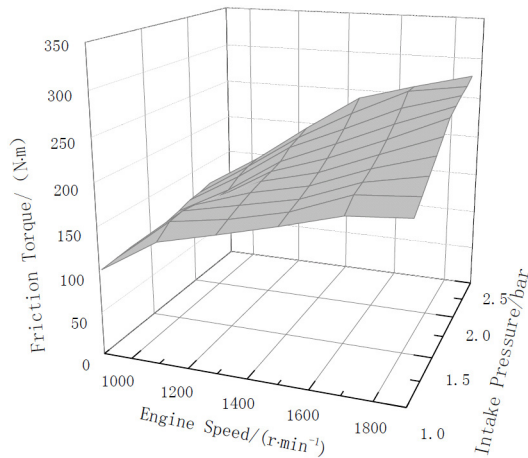


Figure 2. The relationship of friction torque with the engine speed and intake pressure

1.2. The Experiment Study of Load Estimation

After parameters calibrating, the steady load in each conditions is calculated by [Equation \(8\)](#) (in this case the acceleration $\ddot{n}_e$ is 0), and [Figure 3](#) shows the distribution of absolute error for the load estimation result in each speed, which can be seen that the absolute error is larger under the large load, but the relative error of the estimated value is still less than 5%.

For dynamic conditions, the minimum proportion in the load estimation is the friction torque. The result error does not exceed 6% in the dynamic condition with using the calibrated friction torque equation in the steady condition [9]. So the paper mainly studied indicated torque and inertial torque calculation method and the estimated accuracy in dynamic conditions.

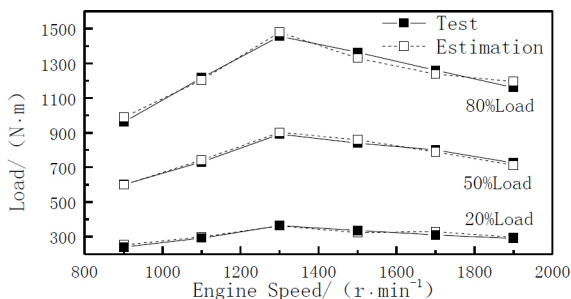


Figure 3. Load estimation results in the steady condition

This paper adequately motivates indicated torque dynamic characteristics in the designed condition of the constant speed and increasing torque, shown in [Figure 4](#), sets the engine to two-range speed control, and adjusts the dynamometer to the fixed rotation speed mode (the experimental speed is 1500r/min). Throttle opening rapidly increases from 20% to 80% with the corresponding increase in supply oil-gear-lever position and the fuel injection quantity; the dynamometer adjusts the load automatically to ensure the speed about

1500 r/min. In this process, the indicated torque is calculated by using the cylinder pressure sensor, and the engine load of the loading process is measured by the torque sensor.

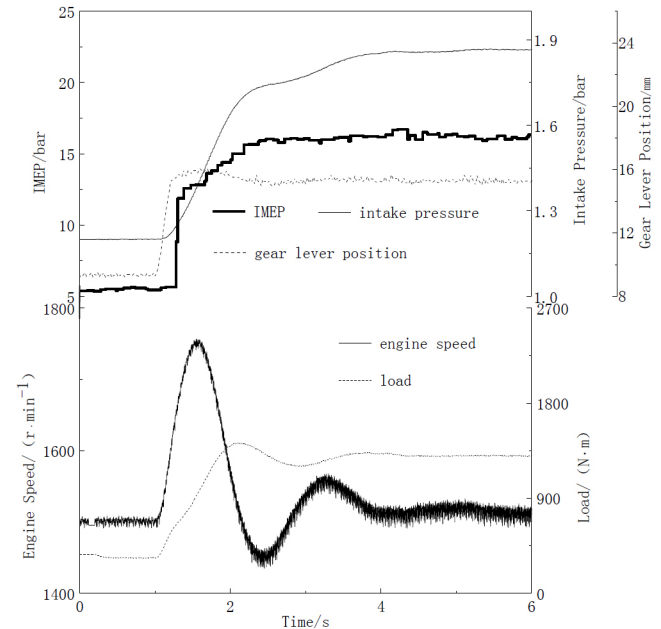


Figure 4. State parameters change in the dynamic condition

[Figure 5](#) compares the estimated value of the indicated torque and the indicated torque value calculated by the cylinder pressure, and the estimated value of the indicated torque in dynamic conditions is a low overall trend, shown in [Figure 5](#), which is due to the impact of the injection duration and in-cylinder pressure in each load points on thermal efficiency cannot be excluded, in the calibration process of the relationship between the air-fuel ratio and the indicated thermal efficiency. Overall, the maximum error in this process is less than 10%, selecting the larger area error (1s ~ 3s in [figure 5](#)) to calculate the mean error of the estimation, which is 5.88%.

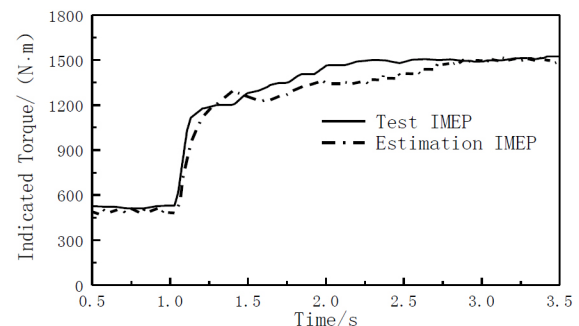


Figure 5. Indicated torque estimation in the dynamic condition

In this paper, the calculation of the inertia torque needs to obtain the average acceleration of the engine cycle, so the mean filtering process of instantaneous speed is needed firstly, and to reduce the impact on speed fluctuations within the cycle. In this paper, the average engine-speed is calculated by [Equation \(9\)](#), where n is the teeth number of the engine crankshaft. As can be seen from [Figure 6](#), the average speed removes effectively the speed fluctuation within the cycle, but also caused a delay about 0.1s.