Disturbance Observer Based Sliding Mode Control for Marine Diesel Engine

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Abstract. In this work, a disturbance observer is designed to obtain the total disturbance including the unknown part, and the error system of the disturbance observer is proved to be finite-time convergence. Based on the total disturbance, a sliding mode controller with exponential reaching rule is proposed for the speed control system of a marine diesel engine. The performance of the proposed controller and the disturbance observer is tested by the simulations on a marine diesel engine model. It shows that the proposed disturbance observer can achieve satisfactory results, and both the control performance and the robustness of the DOSMC are superior to the PID controller.

1. Introduction
Owing to the advantages of high-efficient and economical power installations, the diesel engines have been widely used in daily life and industrial production. In the marine industry, diesel engine is mainly used as the main engine and electric generator. Speed control is the most typical control mode in marine diesel engine as it provides a nearly linear and expected relationship between the speed setting of engine and the ship speed, even though many researchers consider it causes unnecessary load disturbances [1]. The speed control is required to be accurate and smooth, because the speed fluctuation and overshoot will reduce the service life of propulsion system, or even paralyze the entire system [2]. However, the diesel engine is such a complex system that comprises of multiple subsystems, and all subsystems are nonlinear and difficult to model accurately. Besides, the working environment of marine main engine is very harsh. Therefore, the speed control problem for the diesel engine is troublesome, and an accurate and robust speed controller is necessary.

A few decades ago, electronic governors were first designed which make the speed control of diesel engines flexible. From then on, Proportion-Integration-Differentiation (PID) has been widely used in the speed control of diesel engine because of simplicity and reliability [3]. However, the PID controller with the same parameters cannot achieve the best control effect under all conditions for the nonlinear system like the diesel engine [4]. Thus, some researchers modify the PID parameters by combining with different algorithms, such as fuzzy control logic [4], H-infinity [5], genetic algorithm [6], etc. Although these methods can expand the scope of the applicable conditions of PID controller, they cannot fundamentally solve the limitations of PID controller and the PID would lose its advantages of simple and easy implementation. With the advent of high pressure common rail (HPCR) system, fuel injection control becomes more precise, flexible and real-time. Therefore, the application of advanced control algorithms can significantly improve the performance of the marine diesel engine.
And, some researchers have experimented with different algorithms, such as neural networks control [7], model predictive control [8], active disturbance rejection control [9], etc. Sliding mode control (SMC) is also used in diesel engine speed control by some scholars due to its good robustness. Yuan and his collaborators design discrete sliding mode variable structure controller and multi-sliding surface controller, and both controllers have good control effect [10, 11]. However, such methods rely on the accurate model. Khan and Li use the second order sliding mode controller to control engine speed [12, 13], which no longer requires accurate diesel model, but the robustness is decreased. A simple SMC is designed by Zhang for speed control of generators, but the control performance is not satisfactory [14]. Zhang uses the RBF neural network to approximate the diesel engine model parameters, based on which a sliding mode controller is developed [15]. Nevertheless, this method greatly increases the computation and the convergence time is uncertain.

Considering the above situation, this work develops a disturbance observer based sliding mode controller (DOSMC). A finite time stable disturbance observer (DO) is proposed to obtain the current diesel engine disturbances quickly and accurately. Then, a sliding mode controller is designed based on the observer to realize accurate speed control. The designed DOSMC is easy to implement and has satisfactory control performance. Meanwhile, it maintains the robustness of sliding mode control.

2. Marine Diesel Engine Model and Problem Formulation
In this study, a mean value engine model (MVEM) is introduced for studying the speed control system of marine diesel engine. The MVEM consists of engine body and subsidiary systems include turbocharger system, fuel supply system, lubrication system, etc. Figure 1 shows the subsystems of a diesel engine.

![Figure 1. Structure of the marine diesel engine in a ship.](image)

According to the desired speed \( n_d \) given by the telegraph and the actual speed \( n_e \), a suitable fuel delivery per cycle per cylinder \( m_i \) is calculated by the controller to ensure the diesel engine speed track the target speed quickly and accurately. The indicated torque \( M_i \) produced by fuel combustion can be calculated by the following equation

\[
M_i = \frac{m_i q_{LHV} \eta_i N_{cyl}}{2\pi N_s},
\]  

where \( q_{LHV} \) represents the low heating value of fuel, \( \eta_i \) denotes the gross indicated efficiency, \( N_{cyl} \) denotes the number of cylinders, \( N_s \) denotes the number of strokes with \( N_s = 1 \) for two-stroke and \( N_s = 2 \) for four-stroke.

In addition to generating indicated torque, diesel engine itself also consumes some torque internally, for pumping loss and friction are not evitable. The pumping torque \( M_p \) and friction torque \( M_f \) can be calculated as follows

\[
M_p = \frac{m_i q_{LHV} \eta_i N_{cyl}}{2\pi N_s},
\]  

\[
M_f = \frac{m_i q_{LHV} \eta_i N_{cyl}}{2\pi N_s}.
\]
\[ M_a = \frac{V_a}{4\pi}(p_{em} - p_{im}), \]  
\[ M_t = \frac{V_a}{4\pi}10^3(k_1n_e^2 + k_2n_e + k_3), \]

where \( V_a \) is the total displacement, \( p_{em} \) and \( p_{im} \) respectively denote the pressures in the exhaust manifold and the intake manifold, and \( k_1, k_2, k_3 \) are experimental constants.

As shown in the figure 1, the load torque of marine diesel engine mainly comes from propeller, which consists of two parts, i.e., the propeller torque \( M_p \) and the distraction torque \( M_d \). Usually, \( M_d \) is irregular and much smaller than \( M_p \). \( M_p \) can be simulated as

\[ M_p = kn_e^2, \]

where \( k \) is a coefficient depends on the propeller torque coefficient, the density of water, and the propeller diameter.

Above all, the dynamic equation of the whole propulsion system including diesel engine, main shafting, and propeller can be described as

\[ \frac{dn_e}{dt} = \frac{30}{\pi J}(M_t - M_d - M_t - M_p - M_d), \]

where \( J \) is the total rotational inertia of propulsion system.

Combining equations (1)-(5), the dynamic equation model of the propulsion system can be described as

\[ \dot{x} = f(x) + g(t)u + d(t), \]

where system state variable \( x = n_e \), control input variable \( u = m_t \), \( f(x) \), \( g(t) \) and disturbance \( d(t) \) are given as follows

\[ f(x) = -\frac{30}{\pi J}(M_t + M_p), \quad g(t) = \frac{15q_{ival}n_eN_{cyl}}{\pi^2JN_{at}}, \quad d(t) = -\frac{30}{\pi J}(M_p + M_d) \]

In general, \( M_t \) and \( M_p \) can be measured in an experimental environment. However, in the harsh working environment of marine diesel engines, there is always a deviation between the measured value and the actual value. Moreover, the gross indicated efficiency \( \eta_i \) is affected by many factors and difficult to measure. But it usually changes in a small range when the engine is running normally. Considering all these unknown parts as disturbance, if the total interference is denoted as \( D(t) \), system (6) can be rewritten as

\[ \dot{x} = f_0(x) + g_0(t)u + D(t), \]

where \( f_0(x) \) and \( g_0(t) \) denote the known part of \( f(x) \) and \( g(t) \).

3. Controller Design

3.1. Disturbance Observer Design
For the design of the controller, a disturbance observer is designed in this subsection.

Assuming that \( D(t) = h(t) \) and \( h(t) < \bar{h} \). Treating \( D \) as an additional system state variable and letting \( z_1 = x \), \( z_2 = D \), the original system in (8) can be described as
\[
\begin{align*}
\dot{z}_1 &= f_0(x) + g_0(t)u + z_2 \\
\dot{z}_2 &= h(t)
\end{align*}
\]

(9)

which \( z_1 \) and \( z_2 \) are observable. And a disturbance observer is designed to be

\[
\begin{align*}
\dot{\hat{z}}_1 &= f_0(x) + g_0(x)u + \hat{z}_2 - \theta_2 \text{sign}(\hat{z}_1 - z_1) \\
\dot{\hat{z}}_2 &= -\theta^2 \rho_2 \text{sign}^{1/2}(\hat{z}_1 - z_1)
\end{align*}
\]

(10)

where \( \text{sign}^y(x) = \left|x\right|^y \text{sign}(x) \), \( \hat{z}_1 \) and \( \hat{z}_2 \) are observations of \( z_1 \) and \( z_2 \), respectively.

Define observation error \( e_i = \hat{z}_i - z_i \) and \( e_2 = \hat{z}_2 - z_2 \). In view of equations (9) and (10), one can obtain an observation error system as

\[
\begin{align*}
\dot{e}_1 &= e_2 - \theta_2 \text{sign}(e_1) \\
\dot{e}_2 &= -\theta^2 \rho_2 \text{sign}^{1/2}(e_1) - h(t)
\end{align*}
\]

(11)

It can be proved that the error system converges in finite time. And, the proof process is given below.

Define auxiliary variable \( e_i = e_j \theta^{j+\beta} \), \( j = 1, 2 \), \( 0 < \beta < 1 \), the observation error system (11) can be rewritten as

\[
\begin{align*}
\dot{\hat{e}}_1 &= -\theta^\beta \rho_1 \text{sign}(e_1) + \theta \hat{e}_2 \\
\dot{\hat{e}}_2 &= -\theta^\beta \rho_2 \text{sign}^{1/2}(e_1) - \theta^{-\beta/2}h(t)
\end{align*}
\]

(12)

Consider a Lyapunov function candidate as

\[ V_e = \frac{1}{2} \mathbf{g}^T \mathbf{P} \mathbf{g} \]

where \( \mathbf{P} \) is a positive definite symmetric matrix, and auxiliary variable \( \mathbf{g} = [\text{sign}^{1/2}(e_1) \ e_2]^T \).

Differentiating the above Lyapunov function, one can get

\[
V_e = \mathbf{g}^T \mathbf{P} \begin{bmatrix} \frac{1}{2} |e_1|^{1/2} (-\theta^{-\beta} \rho_1 \text{sign}(e_1) + \theta \hat{e}_2) \\
\theta \rho_1 \text{sign}^{1/2}(e_1) - \theta^{-\beta/2}h(t) \end{bmatrix} = 2 \mathbf{g}^T \mathbf{P} \begin{bmatrix} 0 \\
-h(t) \theta^{-\beta} \end{bmatrix} - |e_1|^{1/2} \mathbf{g}^T \mathbf{Q} \mathbf{g}
\]

where \( \mathbf{Q} = A_\beta \mathbf{P} + \mathbf{P} A_\beta \) and \( A_\beta = \begin{bmatrix} -\theta^{-\beta} \rho_1 / 2 & \theta \\
-\theta^{-\beta/2} \rho_2 / 2 & 0 \end{bmatrix} \) which is obviously a Hurwitz matrix.

In view of the definition of \( V_e \), one can get 

\[ |e_1|^{1/2} \leq \| \mathbf{g} \| \leq \lambda_{\min}^{-1/2}(\mathbf{P}) V_e^{1/2}. \]

Therefore,

\[ V_e \leq 2 \mathbf{g}^T \mathbf{P} \begin{bmatrix} 0 \\
-h(t) \theta^{-\beta} \end{bmatrix} - \lambda_{\min}^{-1/2}(\mathbf{P}) \lambda_{\min}^{-1/2}(\mathbf{Q}) V_e^{1/2} \leq -(c_1 - c_2) V_e^{1/2}, \]

where \( c_1 = \lambda_{\min}^{-1/2}(\mathbf{P}) \lambda_{\max}^{-1/2}(\mathbf{Q}) \) and \( c_2 = 2 \bar{h} \lambda_{\max}(\mathbf{P}) \lambda_{\min}(\mathbf{P}) \). There is always a \( \theta \) that makes \( \theta^{1+\beta} > 2 \bar{h} \lambda_{\max}(\mathbf{P}) \lambda_{\min}(\mathbf{P}) \). According to literature [16], it can be known that the observation error converges to zero in a finite time, and the convergence time \( T_e \) is given by

\[ T_e = 2 V_e(c_1 - c_2)^{-1}. \]

So, the finite-time stability of the system (12) is proved.

3.2. Sliding Mode Controller Design

Based on the proposed disturbance observer, a sliding mode controller is designed.
Firstly, a control law is designed as

$$u = g_0^{-1}(t) \left[ v - f_0(x) - \hat{D}(t) \right],$$

where $\hat{D}(t) = \xi_2$ is the observed value of the total disturbance $D(t)$. One can get from the previous proof that in a finite time there will be $D(t) - \hat{D}(t) \approx 0$. Then, substituting equation (13) for system (8), a simple first order linear system can get as

$$\dot{x} = v$$

And, the sliding mode controller is proposed for this system. The structure of the entire controller is shown in figure 2.

![Figure 2. Structure of the proposed DOSMC.](image)

As shown, the error between the actual engine speed $x$ and the reference value $x_d$ is defined as $e = x - x_d$. The sliding surface is chosen as $s = ce + \dot{e}$, $c > 0$. The speed control of diesel engine is required to follow the target value quickly and avoid large overshoot. Thus, an exponential reaching rule is selected, which is

$$\dot{s} = -\varepsilon \sign(s) - ks,$$

where $\varepsilon, k > 0$.

For the subsystem (14), the first time derivative of $s$ is obtained

$$\dot{s} = c\dot{e} + \dot{\varepsilon} = cv + \dot{v} - c\dot{x}_d - \ddot{x}_d.$$

For marine diesel engine, a slope limiter for the desired speed $x_d$ is employed to avert the rapid change of engine speed which could cause damage to the propulsion system [17]. Thus $\dot{x}_d$ is zero or a constant, and $\ddot{x}_d = 0$. Combining equation (15) and (16), one can get

$$\dot{v} = c\dot{x}_d - cv - \varepsilon \sign(s) - ks.$$

Substitute controller law (17) into system (12), it definitely satisfies

$$ss = -\varepsilon |s| - ks^2 \leq 0.$$ (18)

In conclusion, the designed control law for the speed control system (8) can be described as

$$\begin{cases}
    u = g_0^{-1}(t) \left[ v - f_0(x) - \hat{D}(t) \right] \\
    \dot{v} = c\dot{x}_d - cv - \varepsilon \sign(s) - ks
\end{cases}$$ (19)

4. Simulation and Analysis

In order to get an objective comparison, a classical PID controller is designed to compare with the proposed DOSMC. The simulation processes include starting, accelerating, decelerating, suddenly loading and unloading. The slope limit of the desired speed is 400 rpm/s.

4.1. Disturbance Observer Performance

In order to avoid the early stage of the starting process, the DO begins to calculate at about 0.7s when
the speed reaches the switching value of closed-loop control. Figure 3 shows the overall tracking performance of the DO. It can be observed that the DO converges rapidly although the disturbance drastically changes because of the large deviation between the fixed $g_o(t)$ and the actual $g(t)$. The DO keeps a good tracking performance in the following conditions as well. Moreover, the rate of convergence is satisfactory during loading and unloading. In addition, faster tracking performance or smoother effects can be implemented by adjusting the parameters of observer.

![Figure 3](image)

**Figure 3.** The overall tracking performance of the disturbance observer.

### 4.2. Sliding Mode Controller Performance

The comparative results between the PID and DOSMC are illustrated in figure 4.

![Figure 4](image)

**Figure 4.** The control performance comparison between PID and DOSMC.

![Figure 5](image)

**Figure 5.** Steady state speed comparison

![Figure 6](image)

**Figure 6.** Speed comparison during loading and unloading.
In the initial stage, the diesel engine speed rises rapidly under the action of starting torque, and the controllers start work since the engine speed reaches 300 rpm. The parameters of the two controllers are adjusted for acceleration and deceleration conditions. Hence, the control performances of the both controllers are satisfying during acceleration and decelerate conditions. However, the control effect of PID controller is not as good as that of the DOSMC. The DOSMC controller also achieves good result under the starting condition, but the PID controller performs poorly.

In this study, a Gaussian noise is employed to simulate the unknown disturbance $M_d$, so there are speed fluctuations in the steady state. Figure 5 gives the steady state speed comparison of the two controllers. It shows that the speed fluctuation in the DOSMC is smaller than that in the PID. The control performance of two controllers in the face of 50% sudden unloading and loading is shown in figure 6. Both controllers achieve a similar effect when loading, but the performance of the PID controller is unsatisfactory during unloading.

As can be seen above, the DOSMC controller can keep good control performance during all working conditions, and its control performance is better than the PID.

It is well known that the same diesel engine may be used for different propellers, and the moment of inertia of the propulsion system may change with time or working conditions. This is a challenge for the robustness of the controllers. Here, the simulations with different moment of inertia are carried out, and the results are given in figure 7 and figure 8. PID1 and DOSMC1 are the simulation results with the original moment of inertia and PID2 and DOSMC2 are the results under a changed moment of inertia. Obviously, the control effect of the PID controller becomes worse when the rotary inertia is changed. Nevertheless, the DOSMC can always maintain good control performance. Therefore, it can be indicated that the DOSMC is more robust than the PID controller.

**Figure 7.** Speed response with different moment of inertia in acceleration process.  
**Figure 8.** Speed response with different moment of inertia in deceleration process.

5. Conclusion
In this paper, a disturbance observer with finite time convergence is designed, and a DOSMC based on the disturbance observer is proposed for the speed control of a marine diesel engine. In order to assess the control performance of the proposed method, extensive comparatively simulations are carried out on a MVEM by comparing with a conventional PID controller. The tested operation conditions include starting, acceleration, deceleration, loading and unloading. Besides, the robustness of the two controllers is compared by changing the model parameter. The simulation results demonstrate that the designed disturbance observer has good performance, and the proposed DOSMC can effectively improve the control effect and has better robustness than the PID.
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