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MODEL REFERENCE ROBUST CONTROL FOR MARINE PROPULSION SYSTEMS WITH MODEL UNCERTAINTY CAUSED BY HULL DEFORMATION

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Key words: marine propulsion system, model uncertainty, model reference adaptive control, ship speed regulation, hull deformation.

ABSTRACT

A marine propulsion system can be strongly coupled with the hull deformation in a large-scale ship. The interaction of this coupling dynamics increases the nonlinearity and complexity of the control model. Traditional control method based on the empirical PID (proportion-integration-differentiation) parameters has difficulty in dealing with the model uncertainty caused by the hull deflection. However, up to date, limited work has been done to address the model uncertainty problem of the marine propulsion system. It is therefore imperative to develop feasible and effective control systems that take the model uncertainty into account. In this study, a new method based on the model reference adaptive system (MRAS) has been proposed for the active and accurate control of the marine propulsion system coupling with the hull deformation. A finite element model of the ship shaft line was established to investigate the uncertainty boundaries of the propulsion system. Moreover, the sea trial was carried out on the hydraulic dredge named "Changjing 2" to measure the main engine power loss under hull deformation. The finite element analysis (FEA) and shipboard measurement results showed a fiducial interval of [0.1% 10%] for the model uncertainty of the marine propulsion system. These uncertainty boundaries were added into the ship speed control system model, and the MRAS controller was designed based on the Lyapunov theory to mitigate the adverse effects of model uncertainty. The stability of the MRAS has been proven by the Lyapunov stability criterion. Numerical evaluations using MATLAB®/Simulink® software for the "Changjing 2" ship engine parameters have showed high effectiveness of the MRAS structure for speed tracking under the hull deflection coupling. This study has demonstrated that the newly proposed control system can work stably with various ship operating conditions, and its performance is superior to the traditional PID controller.

I. INTRODUCTION

The governors of marine propulsion systems are usually designed using empirical process models without considering the influence of hull deformation [8]. These governors typically neglect the model uncertainty but work fine in practice for the speed control of small-scale ships. However, with the rapid development in the shipping industry, large vessels such as mammoth tankers/monster tankers, mining and storage ships for marine resources, aircraft carriers and dreadnaughts, etc., become bigger and bigger in size, thereby leading to an evidential increase of the interaction between the propulsion system and hull. The performance of traditional controllers in this situation is not as good as desired [3]. Nonetheless, literature review shows that little research has considered the effects of uncertain parameters on the modeling of the marine propulsion systems. It is therefore critical to conduct further research so that the model uncertainty caused by hull deformation is considered in the control of the marine propulsion systems of large-scale ships.

As well known, the ship shaft line is one of the most important components to transfer the energy from the ship engine to thrusters, and to receive the axial thrust of the propeller to drive the ship. It is the indispensable part for the normal running of the marine propulsion system that once abnormalities occur, the system propulsive efficiency may decrease greatly. This is because when a hull deformation occurs, the displacement of the support bearings of the shaft can be significant,

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leading to a misalignment of the shaft line. The torsional and transverse vibration of the shaft line caused by the misalignment may increase the model uncertainty of the marine propulsion system, which cannot be ignored in the system control. Conventional PID controllers have been proven to be effective for the ship speed control in rated conditions and have been widely used because of their reliability, simplicity and universality [3]. However, if the operation conditions deviate the system rated conditions or the system encounters disturbances, their performance may be unsatisfactory and even out of function because the PID parameters are fixed in advance. To solve these problems, advanced controllers have been introduced to control the marine propulsion system adaptively. Brzózka [3] presented a new combination of the model following control (MFC) and internal model control (IMC) to control a ship engine speed. The numerical simulation showed the high effectiveness of MFC/IMC structure for speed-keeping and speed-changing in the ship regulation. Xiros [9] proposed an improved PID tuning method based on sensitivity H∞-norm specification for marine diesel engine governors to serve against severe propeller load fluctuation. In order to improve the stability of ship speed and main engine (ME) load, Zhang and Ren [12-17] designed a ship speed regulating simulation system, which consisted of a computer, PLC, frequency control motor, servo motor and centrifugal pump. The adaptive neural networks were used as the error models to compensate the system model uncertainties and enhance the ship speed control versus the PID controller. The simulation results showed that the intelligent control process of ship speed was smooth and stable when the system model had some uncertainties. Although these recently developed methods can overcome the disadvantages of traditional PID controllers, they also suffer some drawbacks including complex and time-consuming [11]. Furthermore, the uncertainties and perturbations introduced into the systems were selected randomly, and the uncertainty cause and boundaries were not explained and discussed in their work. Due to the lack of the insight into the impact of the parameter uncertainty on the system design, ,the influence of the model uncertainty of the marine propulsion system caused by hull deflection has not been investigated yet.

This paper presents a new system to control the marine propulsion system taking the hull deformation into account. In the previous work [10] the adaptive controllers were investigated in the ship speed regulation with model uncertainties and have been proven efficient for dealing with the system disturbances. Based on the above development, a model reference adaptive robust control scheme is proposed for the marine propulsion system with model uncertainty caused by hull deformation in this paper. The boundaries of the system dynamical uncertainty about the interaction of propulsion system and hull deflection is determined using both finite element analysis (FEA) simulation and sea trial measurements, and the quantification of the propulsion system model uncertainty is hence obtained. To accommodate the model uncertainty, the model reference adaptive system (MRAS) is presented to track the propulsion system speed in the closed control loop. By doing so, the disturbances can be compressed. Numerical studies are carried out in this paper to evaluate the efficiency of the proposed control system.

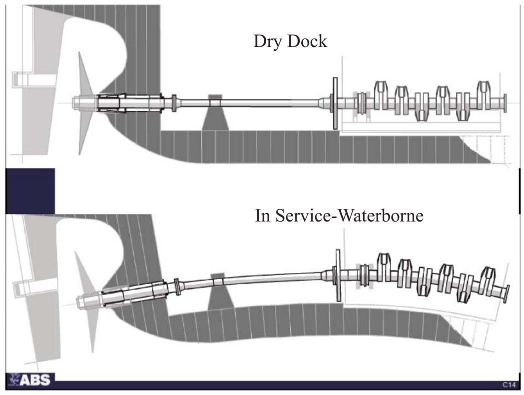

Fig. 1. The shaft line misalignment caused by hull deformation [4].

This paper is organized as follows. In Section 2, the mean value model of the marine propulsion system with model uncertainty caused by hull deflection is established. The investigation on the model uncertainty is conducted by FEA simulation and shipboard measurement in Section 3. The uncertainty boundaries are estimated according to the simulation and experiment results. In Section 4 the robust controller design is presented based on the MRAS. The stability of the control system is analyzed via the Lyapunov stability criterion. The effectiveness of the proposed method is assessed by numerical studies in Section 5. Finally, conclusions are drawn in Section 6.

II. MEAN VALUE MODEL OF MARINE PROPULSION SYSTEM

The dynamics characteristics of the marine propulsion system present strong nonlinearity and time variance, especially for the ship diesel engine. Thus it is very difficult to model the system precisely [1, 2]. Mean value model is one of the important and feasible means to designing and simulating the marine propulsion system [8]. The existing methods for model design and simulation of the marine propulsion system usually consider the impact factors of the waves and loads on the model performance [14, 15], but seldom take into account the coupling effect of the hull deformation on the propulsion system. However, the accuracy of shaft line alignment may be influenced significantly by the hull deflection. Chris [4] reported in his research that the shaft line alignment deviates from the reference line in service. Fig. 1 shows the comparison of shaft line alignment in dry dock and service [4]. It is noticeable that the shaft line deflects severely with the hull deflection. Dong *et al*. [5] stated that the maximum offset of the shaft line could be up to 180 mm, and the minimum offset of the stern bearings was 2.2 mm for a large

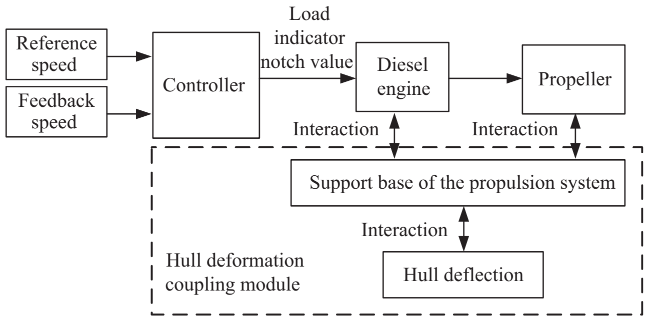

Fig. 2. Ship speed control structure with considering hull deformation.

container ship. Furthermore, Zhou [18] pointed out that a 2.8 mm lift of the intermediate bearing of the large scale ships may lead to the bearing void, which may increase the shaft vibration and energy loss, affect the system performance and even cause a system breakdown. Hence, it is imperative to investigate the interaction of hull and shaft line in the modeling of the propulsion system for large scale ships.

In this Section, the fusion of hull deformation into the modeling of the marine propulsion system is discussed. The ship speed control structure is shown in Fig. 2. The system consists of the controller module, diesel engine module, propeller module and hull deformation coupling module, etc. The mean value model of the marine propulsion system is briefly described as follows.

1. Diesel Engine Module

Normally, the output torque of the ship diesel engine can be approximately regarded as a function involved with the oil injection Q and engine speed ω, i.e. $M_e = f(Q, \omega)$. The differential equation is given by [13]

$$\Delta M_e = \frac{\partial \Delta M_e}{\partial \Delta Q}\Delta Q + \frac{\partial \Delta M_e}{\partial \Delta \omega}\Delta \omega \tag{1}$$

where Δ denotes the increments of the variables. The oil injection Q is derived by partially differentiating the engine speed ω and the stroke of the fuel pump rack L:

$$\Delta Q = \frac{\partial \Delta Q}{\partial \Delta \omega}\Delta \omega + \frac{\partial \Delta Q}{\partial \Delta L}\Delta L \tag{2}$$

2. Shaft Line Module

According to Newton's second law for rotational motion, the dynamics equation of the shaft line module can be expressed as

$$2\pi J \frac{d\Delta\omega}{dt} = \Delta M_e - \Delta M_s - \Delta M_p \tag{3}$$

where ΔM_s and ΔM_p denote the additional engine torque loss and propeller load torque, respectively. J denotes the rotational inertia of the shaft line. The additional torque is caused by the hull deflection model and is calculated as

$$\Delta M_s = k_t \Delta M_e \tag{4}$$

$$\Delta M_p = \frac{\partial \Delta M_p}{\partial \Delta \omega}\Delta \omega + \frac{\partial \Delta M_p}{\partial \Delta \zeta}\Delta \zeta \tag{5}$$

where k_t is the engine output torque loss index because of the coupling effect of the hull deflection, and ζ denotes the propeller load fluctuation.

3. Marine Propulsion System Module

Substituting Eqs. (1), (2), (4), and (5) for ΔM_e, ΔM_s and ΔM_p in Eq. (3), the marine propulsion system model is derived as:

$$\begin{aligned} &2\pi J \frac{d\Delta\omega}{dt} + \frac{\partial M_p}{\partial \Delta\omega}\Delta\omega - 2(1-k_t)\frac{\partial M_e}{\partial \Delta\omega}\Delta\omega \\ &= (1-k_t)\frac{\partial M_e}{\partial \Delta L}\Delta L - \frac{\partial M_p}{\partial \Delta\zeta}\Delta\zeta \end{aligned} \tag{6}$$

when the diesel engine working in the stable operation process, Eq. (6) can be rewritten as

$$2\pi J \frac{M_R}{\omega_R \cdot K}\frac{d\omega_r}{dt} + \omega_r = \frac{(1-k_t)L_R}{\omega_R \cdot K}\frac{\partial M_e}{\partial \Delta L}L_r - \frac{1}{\omega_R \cdot K}\frac{\partial M_p}{\partial \Delta\zeta}\Delta\zeta \tag{7}$$

where ω_R, M_R and L_R denote the rated engine rotational speed, rated engine torque and maximum rack stroke, respectively. ω_r and L_r denote the relative variations of engine speed and rack stroke, respectively. K equals $\frac{\partial M_p}{\partial \Delta\omega} - 2(1-k_t)\frac{\partial M_e}{\partial \Delta\omega}$.

4. Ship Speed Regulator Model

In the stable operation process, the terms of K, $\frac{\partial M_e}{\partial \Delta L}$ and $\frac{\partial M_p}{\partial \Delta\zeta}$ in Eq. (7) can be approximated according to the engine operation characteristic curves [13]. However, the torque loss index k_s is unknown. It is only feasible to determine its upper and lower boundaries via observing the contribution of the shaft line vibration to the engine torque loss. Thus, the mathematical model of the transfer function of the marine propulsion system in the stable operation process can be expressed as [11]:

$$G_1(s) = \frac{f(k_t)K_1}{f(k_t)T_1 s + 1} \tag{8}$$

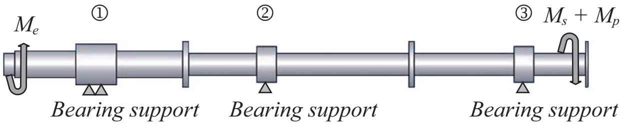

Fig. 3. Ship propulsion shaft line model consisting of 3 shafts labeled as ①, ② and ③.

where $f(k_t)$ denotes the model uncertainty of the torque loss index caused by hull deformation, K_1 denotes the magnification coefficient of the regulator channel, and T_1 denotes the engine time constant.

The transfer function of the actuator for the ship engine control is given by [11]

$$G_2(s) = \frac{K_2}{T_2 s + 1} \tag{9}$$

where K_2 denotes the magnification coefficient, and T_2 denotes the actuator time constant.

Considering the hysteresis problem of the diesel engine speed signal in the transferring process, a hysteretic link $e^{-\tau s}$ should be added into the ship speed control system. In addition, a speed detecting link $G_3(s) = 1$ is also needed. Hence, the mathematical model of the ship speed control system is

$$G(s) = G_1(s)G_2(s)G_3(s)e^{-\tau s} = \frac{f(k_t)K_1K_2}{(f(k_t)T_1 s + 1)(T_2 s + 1)}e^{-\tau s} \tag{10}$$

It can be noticed from Eq. (10) that the model uncertainty term $f(k_t)$ caused by the hull deflection increases the system complexity. For a robust design of the control system, it is therefore essential to investigate the degree of the model uncertainty and determine its boundaries.

III. INVESTIGATION ON THE MODEL UNCERTAINTY

The modeling uncertainty may influence the accuracy of the possible responses of the system. To well compensate the system perturbations, it needs to determine the probability distributions of the uncertainty in advance. Hence, prior to the design of robust control for the marine propulsion system with a consideration of the hull deformation coupling effect, the finite element analysis (FEA) and the shipboard measurement have been implemented to assess the uncertainty degree of the model.

1. Finite Element Analysis (FEA)

In this study, we have simulated the deformation of the shaft base which is caused by the hull deformation to investigate its influence on the engine power transfer. A ship shaft line transferring the thrust force to the thrust bearing (see Fig. 3) is established using FEA software.

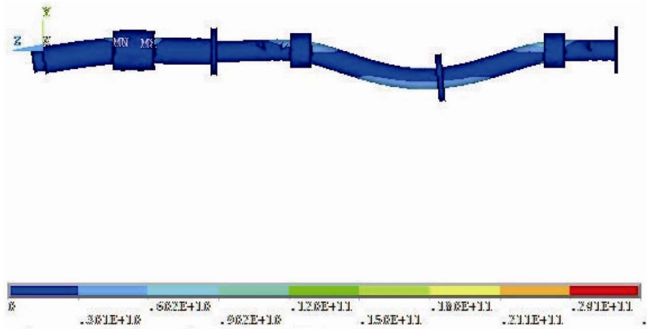

Fig. 4. The FEA analysis result of the ship shaft line.

Fig. 5. The "Changjing 2" ship applied to the sea trial.

In Fig. 3, the shaft line model consists of the first shaft ①, intermediate shaft ②, stern shaft ③ and support bearings. The length of the first shaft is 1245 mm with a diameter of 100 mm. The length of the intermediate shaft is 1600 mm with a diameter of 80 mm, and stern shaft is 1225 mm in length with and 80 mm in diameter. The entire shaft is hollow with an internal diameter of 25 mm. In the analysis, the engine torque of 600 N · m is applied to the shaft left end and the load torque of 600 N · m is applied to the other end. A distributed load of 1450 N/m acting as the shaft line gravity is distributed along the shaft. In addition, a comparison has been done to investigate the hull deflection effect on the engine torque loss via two different simulation states. One is the dry dock state where three concentrated loads with the same value of 15 kN are added on the support bearings of shaft ①, ② and ③, respectively. The other one is to simulate the severe hull deflection situation where three concentrated loads of 35 kN, 55 kN and 75 kN are applied to the support bearings of shaft ①, ② and ③, respectively. The FEA analysis result of state 2 is shown in Fig. 4. It can be seen from Fig. 4 that the shaft line is deflected obviously when introducing the hull coupling effect on it, and the engine power loss could reach to 4.5% of the input power according to the calculation result using the state 1 as the baseline. If double the three concentrated loads in state 2, the engine power loss could reach to 9.8%.

2. Shipboard Measurement

The shipboard measurement is carried out in the ship of "Changjing 2" (see Fig. 5), which is an 8000 m^3 hydraulic dredge. The normal rated power of the main engine is 6300 kW, and the rated rotation speed is 600 rpm. The temperature

Table 1. Shaft power test results for "Changjing 2".

Load	Shaft Speed (rpm)	Shaft Power A (kW)	Shaft Power B (kW)	Relative Error (%)
25%	105.8	707	675	3.87
50%	133.9	1401	1396	0.30
75%	151.4	2049	2027	0.89
85%	158.6	2388	2311	2.74
100%	167.6	2920	2817	3.11

of sea water is 21°C in the sea trial and the wind speed is 4~7 MPH. The torque meter is installed on the shaft to measure the shaft power.

In the actual measurement process, the shaft line may suffer from the bending moment and axial force etc., which arise mainly from the hull deflection. Table 1 shows the sea trial results of the shaft power. In Table 1, the shaft power A denotes that the measurement has neglected the hull deflection coupling effect, while the shaft power B has considered this effect. It can be seen that an uncertainty ranging from 0.30% to 3.87% is presented for shaft power B with respect to the value of shaft power A under the different load conditions.

Thus, the finite element analysis (FEA) and the shipboard measurement results show that there is a fiducial interval of [0.1% 10%] for the model uncertainty of the marine propulsion system. This uncertainty region should be incorporated in the controller design of the ship speed regulator to ensure the effectiveness of the control system.

IV. ROBUST DESIGN AND TEST FOR THE MARINE PROPULSION SYSTEM

As mentioned in Section 3, the control design of the ship speed system encounters mismatch between the actual plant dynamics and the dynamics of the model due to hull deformation. Fortunately, the closed loop control technique can ensure the robustness of the control system within certain bounds of the model uncertainties. The model reference adaptive system (MRAS) is one of the most promising closed loop approaches [6, 7]. The typical parallel scheme of the MRAS is given in Fig. 6, which consists of the reference model, the adjustable model and the adaptive mechanism. The general idea behind the MRAS is to develop a closed loop controller with parameters that can be updated to incorporate the model uncertainty of the system. The control parameters are modified based on an error vector that is derived from the difference between the outputs of the two dynamic models. The MRAS aims to eliminate the error and to match the response of the reference model.

1. Model Reference Adaptive Control of Ship Speed

The nonlinear model of marine propulsion system in Eq. (10) can be reproduced in the differential form:

$$\ddot{y} + f_1(k_t)\dot{y} + f_2(k_t)y = u - f(\dot{y}, u) \tag{11}$$

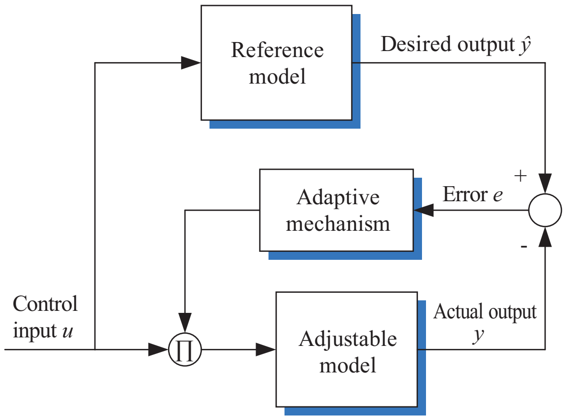

Fig. 6. The typical parallel scheme of the MRAS.

where symbol y denotes the engine speed and symbol u is the control input of the rack stroke. $f_1(k_t)$, $f_2(k_t)$ and $f(\dot{y}, u)$ denote the model uncertainties. $f_1(k_t)$ and $f_2(k_t)$ are unknown parameters involved with k_t but satisfies

$$\begin{cases} 0 \le a \le f_1(k_t) \le b \\ 0 \le c \le f_2(k_t) \le d \end{cases} \tag{12}$$

where a, b, c and d are known constants. And $f(\dot{y}, u)$ is the nonlinear function involved with the engine torque loss.

$$f(\dot{y}, u) = \kappa(\alpha)\operatorname{sgn}(\dot{y}) + \kappa(\beta)\exp(\gamma|\dot{y}|)\operatorname{sgn}(\dot{y}) \tag{13}$$

where $0 \le \kappa(\alpha) \le \alpha$ and $0 \le \kappa(\beta) \le \beta$; α, β and $\gamma(-1 < \gamma < 0)$ need to be specified. So Eq. (11) presents the actual marine propulsion model with the hull deflection coupling effect.

The reference model is defined as:

$$\ddot{\hat{y}} + p\dot{\hat{y}} + q\hat{y} = r \tag{14}$$

where p ($p > 0$) and q ($q > 0$) are the desired constants to guarantee the tracking of desired rack stroke r.

Subtract Eq. (11) from Eq. (14) to yield the error

$$\ddot{e} + p\dot{e} + qe = r - u + f(\dot{y}, u) + (f_1(k_t) - p)\dot{y} + (f_2(k_t) - q)y \tag{15}$$

where $e = \hat{y} - y$.

Rewrite Eq. (15) as

$$\begin{cases} \dot{e} = \dot{e} \\ \ddot{e} = -p\dot{e} - qe + r - u + f(\dot{y}, u) \\ \quad + (f_1(k_t) - p)\dot{y} + (f_2(k_t) - q)y \end{cases} \tag{16}$$

Define the state variable vector $x^T = [e \quad \dot{e}]$. Eq. (16) can be expressed as

$$\dot{x} = Ax + \begin{bmatrix} 0 \\ \Delta \end{bmatrix} - \begin{bmatrix} 0 \\ 1 \end{bmatrix}(u - r) \tag{17}$$

where $\Delta = f(\dot{y}, u) + (f_1(k_t) - p)\dot{y} + (f_2(k_t) - q)y$, and $A = \begin{bmatrix} 0 & 1 \\ -q & -p \end{bmatrix}$. Since A is a stable matrix, according to Lyapunov stability criterion, there exists the positive definite matrix P for any positive definite matrix Q to satisfy the Lyapunov equation:

$$A^T P + PA = -Q \tag{18}$$

Define the auxiliary signal as

$$\hat{e} = [0 \quad 1]Px = [0 \quad 1]\begin{bmatrix} p_1 & p_2 \\ p_2 & p_3 \end{bmatrix}\begin{bmatrix} e \\ \dot{e} \end{bmatrix} \tag{19}$$

Thus the control law is given by

$$u = r + k_1 y + k_2 \dot{y} + k_3 \tag{20}$$

Where the gain functions are defined as

$$\begin{cases} k_1 = \dfrac{1}{2}\{k_p[1 - \mathrm{sgn}(\hat{e}y)] + [1 + g_p\,\mathrm{sgn}(\hat{e}y)]\} \\ k_2 = \dfrac{1}{2}\{k_d[1 - \mathrm{sgn}(\hat{e}\dot{y})] + g_d[1 + \mathrm{sgn}\,\mathrm{sgn}(\hat{e}\dot{y})]\} \\ k_3 = \alpha\,\mathrm{sgn}(\hat{e}) + \beta\,\mathrm{sgn}(\hat{e}) \end{cases} \tag{21}$$

where k_p, k_d, g_p and g_d are the gain constants. Hence, the control law given by Eq. (20) uses the gain functions to compensate the model uncertainties, i.e. uses the items $k_1 y + k_2 \dot{y} + k_3$ to estimate the model uncertainty item Δ in Eq. (17). By doing so, the error e can be eliminated effectively.

Since the controller is derived via the Lyapunov stability criterion, once A is stable and the positive definite matrixes P and Q are obtained, the control system is global asymptotically stable. The proof is given below.

2. The Stability Proof of the Controller

Theorem 1. As to the uncertain nonlinear system (11) and the reference system (14), if the gains of the model reference adaptive robust control law u (20) and (21) are given by $k_p = c - q$, $g_p = d - q$, $k_d = a - p$ and $g_d = b - p$, then for any uncertain $f(\dot{y}, u)$ and uncertain parameters $f_1(k_t)$ and $f_2(k_t)$, the error can be made to $\lim_{t\to\infty}(\bar{y} - y) = 0$.

Proof of Theorem 1. Substitute Eq. (20) for u in Eq. (17) to yield the closed control loop:

$$\dot{x} = Ax + \begin{bmatrix} 0 \\ \Delta - B \end{bmatrix} \tag{22}$$

where $B = k_1 y + k_2 \dot{y} + k_3$. Choose Lyapunov candidate function

$$V(t, x) = \frac{1}{2}x^T P x \tag{23}$$

Define $C = \begin{bmatrix} 0 \\ \Delta - B \end{bmatrix}$, then $\dot{x} = Ax + C$, which yields

$$\begin{aligned} \dot{V}(t, x) &= \frac{1}{2}\dot{x}^T P x + \frac{1}{2}x^T P \dot{x} \\ &= \frac{1}{2}(Ax + C)^T P x + \frac{1}{2}x^T P(Ax + C) \\ &= \frac{1}{2}x^T(A^T P + PA)x + \frac{1}{2}(C^T P x + x^T P C) \end{aligned} \tag{24}$$

where

$$\begin{aligned} C^T P x &= [0 \quad \Delta - B]\begin{bmatrix} p_1 & p_2 \\ p_3 & p_4 \end{bmatrix}\begin{bmatrix} e \\ \dot{e} \end{bmatrix} \\ &= (\Delta - B)\hat{e} \end{aligned} \tag{25}$$

$$\begin{aligned} x^T P C &= [e \quad \dot{e}]\begin{bmatrix} p_1 & p_2 \\ p_2 & p_3 \end{bmatrix}\begin{bmatrix} 0 \\ \Delta - B \end{bmatrix} \\ &= [p_1 e + p_2 \dot{e} \quad p_2 e + p_3 \dot{e}]\begin{bmatrix} 0 \\ \Delta - B \end{bmatrix} \\ &= (\Delta - B)\hat{e} \end{aligned} \tag{26}$$

Substitute (18), (25) and (26) in (23) to obtain

$$\dot{V}(t, x) = -\frac{1}{2}x^T Q x + (\Delta - B)\hat{e} \le -\frac{\lambda_m}{2}\|x(t)\|^2 + (\Delta - B)\hat{e} \tag{27}$$

where λ_m denotes the minimum eigenvalue of Q.

As to the item of $(\Delta - B)\hat{e}$, unfold it as

$$\begin{aligned} (\Delta - B)\hat{e} &= [f(\dot{y}, u) + (f_1(k_t) - p)\dot{y} \\ &\quad + (f_2(k_t) - q)y - k_1 y - k_2 \dot{y} - k_3]\hat{e} \\ &= [f(\dot{y}, u) - k_3]\hat{e} + [(f_1(k_t) - p) - k_2]\hat{e}\dot{y} \\ &\quad + [(f_2(k_t) - q) - k_1]\hat{e}y \end{aligned} \tag{28}$$

where,

$$[f(\dot{y},u)-k_3]=$$

$$\hat{e}>0\ \begin{cases}(\kappa(\alpha)+\kappa(\beta)\exp(\gamma|\dot{y}|))-(\alpha+\beta)\le 0 & \dot{y}>0\\ -(\kappa(\alpha)+\kappa(\beta)\exp(\gamma|\dot{y}|))+(\alpha+\beta)\le 0 & \dot{y}<0\end{cases}$$

$$\hat{e}<0\ \begin{cases}(\alpha+\beta)-(\kappa(\alpha)+\kappa(\beta)\exp(\gamma|\dot{y}|))\ge 0 & \dot{y}>0\\ (\alpha+\beta)+(\kappa(\alpha)+\kappa(\beta)\exp(\gamma|\dot{y}|))\ge 0 & \dot{y}>0\end{cases}$$

$$[f_1(k_t)-p-k_2]=\begin{cases}f_1(k_t)-p-g_d=f_1(k_t)-b\le 0 & \hat{e}\dot{y}>0\\ f_1(k_t)-p-k_d=f_1(k_t)-a\ge 0 & \hat{e}\dot{y}<0\end{cases}$$

$$[f_2(k_t)-q-k_1]=\begin{cases}f_2(k_t)-q-g_p=f_2(kt)-d\le 0 & \hat{e}y>0\\ f_2(k_t)-q-k_p=f_2(kt)-c\ge 0 & \hat{e}y<0\end{cases}$$

So $[f(\dot{y},\ u)-k_3]\,\hat{e}\le 0$, $[(1/f(k_t)-p)-k_2]\,\hat{e}\dot{y}\le 0$ and $[(1/f(k_t))^2-q)-k_1]\,\hat{e}y\le 0$.

Hence, $(\Delta-B)\hat{e}\le 0$, and (27) satisfies

$$\dot{V}(t,x)\le-\frac{\lambda_m}{2}\|x(t)\|^2\quad \forall x \tag{29}$$

Eq. (29) indicates that for any given $f(\dot{y},\ u)$, $f_1(k_t)$ and $f_2(k_t)$, the control system is asymptotically stable. Hence, this completes the proof.

V. NUMERICAL EXPERIMENTS AND RESULTS

The propulsion system of "Changjing 2" ship is taken as the control objective in this work. The "DAIHATSU 12DKM-36F" heavy slow diesel engine is selected in the numerical experiments. The engine time constant T_1 is 5.2, the magnification coefficient K_1 is 46.8, the hysteretic time τ is 0.03, the magnification coefficient K_2 and time constant T_2 of the actuator are 1.05 and 0.3, respectively. Thus, the propulsion model is

$$G(s)=\frac{49.14f(k_t)}{(f(k_tt)5.2s+1)(0.3s+1)}e^{-0.03s} \tag{30}$$

According to the uncertainty investigation in Section 3, the variation interval of $f(k_t)$ is between 0.9 and 0.999. Hence, in the differential form (11), we can deduce that

$$\begin{cases}0\le a=3.525\le f_1(k_t)\le b=3.547\\ 0\le c=0.642\le f_2(k_t)\le d=0.712\end{cases} \tag{31}$$

Choose $\kappa(\alpha)=0.5$, $\kappa(\beta)=0.04$ and $\gamma=-0.1$. The reference model is set as:

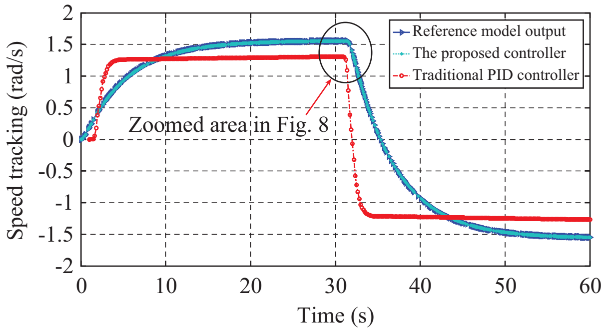

Fig. 7. The engine speed tracking performance of the two methods.

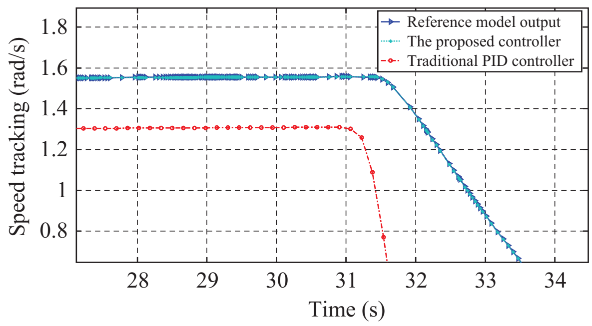

Fig. 8. The zoomed picture of the marked area in Fig. 7.

$$\dddot{y}+3.526\ddot{y}+0.641\dot{y}=31.5r \tag{32}$$

where $p = 3.526$ and $q = 0.641$.

Thus $A=\begin{bmatrix}0 & 1\\ -0.641 & -3.526\end{bmatrix}$. Set the positive definite matrix be $Q=\begin{bmatrix}1 & 0\\ 0 & 1\end{bmatrix}$. Then use the Lyapunov equation to calculate matrix P, and $P=\begin{bmatrix}2.983 & 0.780\\ 0.780 & 0.363\end{bmatrix}$.

Two kinds of time-varying reference of the rack stroke r are used to evaluate the robustness and speed tracking ability of the proposed control system in the numerical tests. One is step reference order, and the other is sinusoidal reference order.

1. Case Study 1

In this case study, we choose $r = \mathrm{sgn}(\sin(0.1t))$ to simulate the ahead and astern operations. The proposed MRAS method is compared with the traditional PID controller in the experiments. The comparison of the engine speed tracking ability of the two controllers is shown in Fig. 7. Fig. 8 is the local zoomed area of Fig. 7. The efficiency of the speed tracking of the proposed controller is evaluated by the tracking error in Fig. 9, and Fig. 10 gives the controller output.

2. Case Study 2

In this case study, we choose $r = \sin(0.1t)$ to simulate a

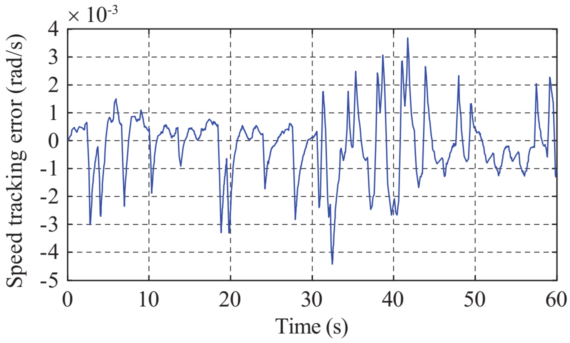

Fig. 9. The engine speed tracking error of the MRAS.

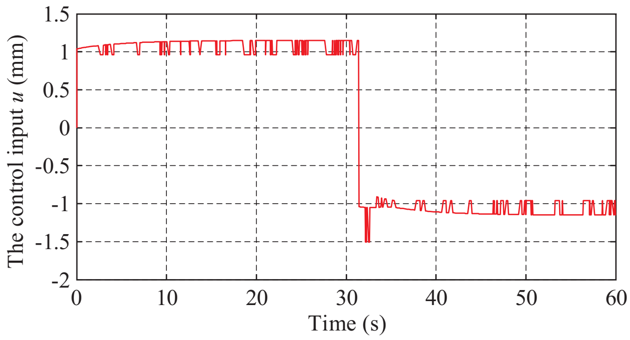

Fig. 10. The control input *u* of the MRAS.

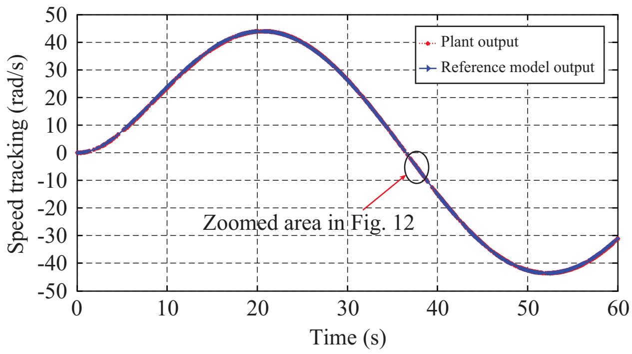

Fig. 11. The engine speed tracking performance of the MRAS.

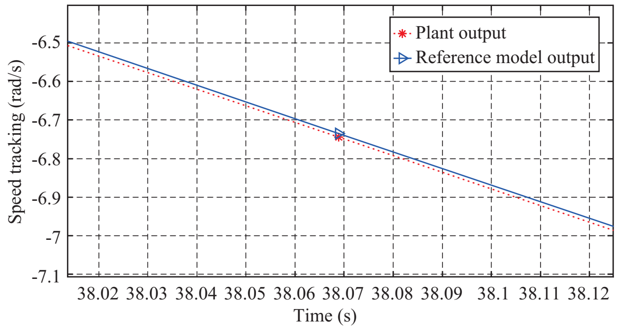

Fig. 12. The zoomed picture of the marked area in Fig. 11.

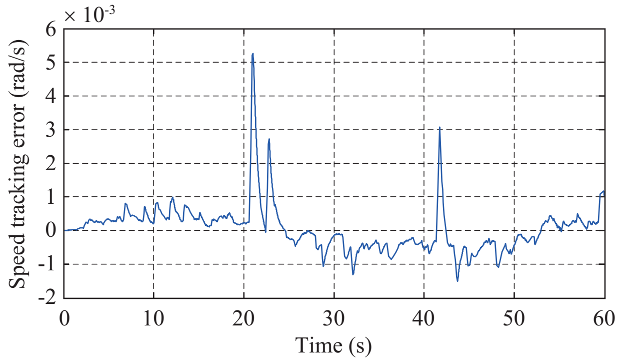

Fig. 13. The engine speed tracking error of the MRAS.

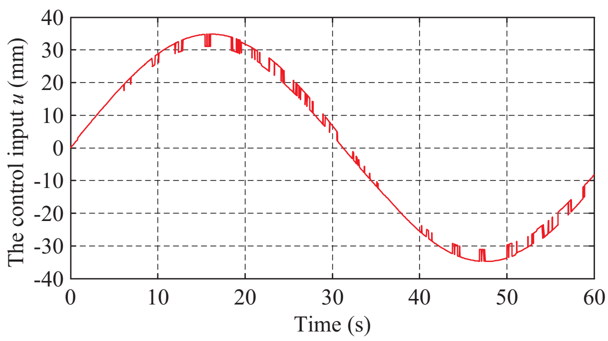

Fig. 14. The control input *u* of the MRAS.

severe nonlinear disturbance in the marine environment. The engine speed tracking ability of the proposed controller is shown in Figs. 11-12. The efficiency of the speed tracking of the proposed controller is evaluated by the tracking error in Fig. 13 while Fig. 14 gives the controller output.

In order to highlight the effectiveness of the proposed controller, we have increased the model uncertainty up to

$$\begin{cases} a = 3.025, b = 4.047 \\ c = 0.042, d = 1.212 \end{cases} \tag{33}$$

The performances of the MRAS in this condition are shown in Figs. 15-18. The output data has demonstrated that the controller works well even with a high level uncertainty.

3. Discussion

The numerical experiment results of the two case studies demonstrate that the proposed controller works stably and can track the given reference order precisely with model uncertainty caused by the hull deformation.

We can see from Figs. 7-8 that the MRAS outperforms the conventional PID controller with respect to the speed tracking ability. The PID controller deviates from the reference speed seriously, and its tracking accuracy is low when encountering the situations with hull deformation. The mean square error is 2.5 rad/s in case study 1 using the PID controller. It can infer that the PID controller cannot adapt the model uncertainty due to that the PID parameters have been predetermined and cannot be changed in the control process.

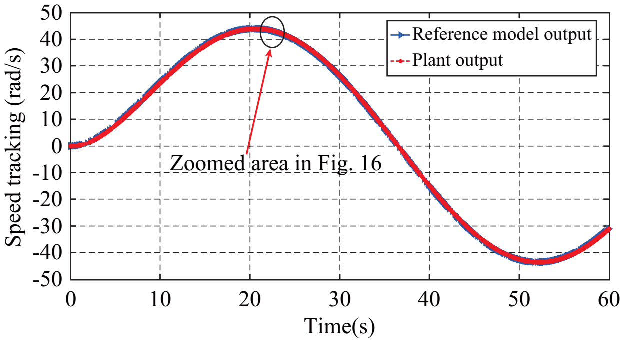

Fig. 15. The engine speed tracking performance of the MRAS.

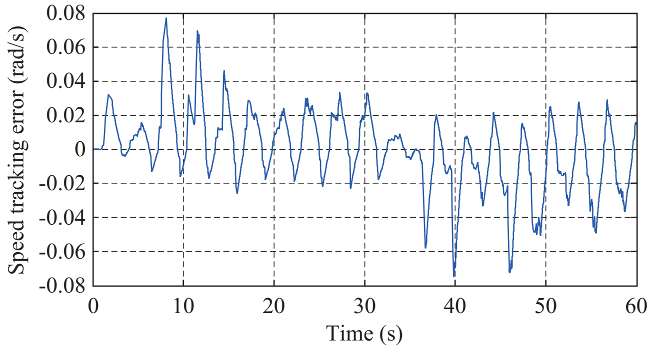

Fig. 17. The engine speed tracking error of the MRAS.

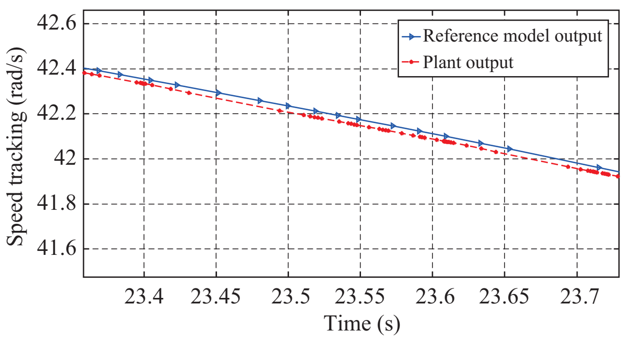

Fig. 16. The zoomed picture of the marked area in Fig. 15.

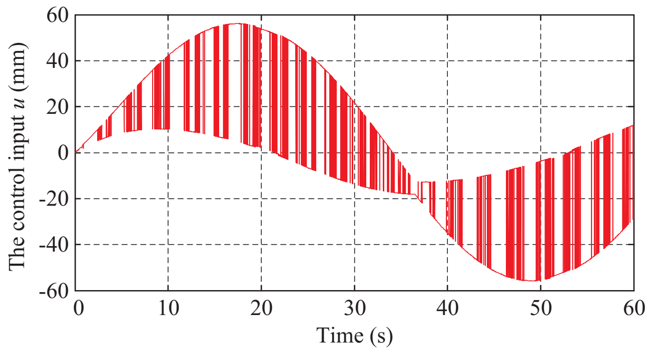

Fig. 18. The control input *u* of the MRAS.

However, taking the advantage of self-adaptation, the MRAS can follow the desired speed output accurately under varying reference orders, and its tracking error is much smaller than that of the conventional one.

In case study 2, the time-varying reference order increases the control complexity. However, Figs. 11-13 show that the proposed MRAS is suitable and stable to control the ship speed. Moreover, in the enhanced experiment, although the model uncertainty has increased significantly, the control system can still work well, and the speed tracking error is acceptable. In case 2, a comparison to the PID controller is not carried out. This is because the traditional PID cannot follow the time-varying reference order, and the control system is readily to diverge.

Hence, the robustness of the proposed controller has been verified by the numerical experiments, which shows the effectiveness of the presented control system.

VI. CONCLUSIONS

The large-scale ships enhance the efficiency of the waterborne transport, which benefits the national and international social economy significantly. However, the rapid increase of the ships in size brings the coupling problem of the propulsion system and the hull. Severe hull deformation may intensify the shaft line vibration, cost extra engine power, and even damage the propulsion system. As a result, the model uncertainty of the propulsion system which often increases with the sizes of the ships, has become an urgent issue. Traditional PID controller limits itself on the empirical parameter settings, and cannot achieve high control performance when dealing with the marine propulsion system with the model uncertainty. Based on the marine propulsion nonlinear system model, a novel control structure via model reference adaptive robust control algorithm is presented in this paper. The model uncertainty investigation is implemented through both finite element analysis and sea trial to provide insight into the interaction of the propulsion system and the hull deformation. The advantages of this investigation are to determine reliable uncertainty boundaries for the propulsion system. Thus, the Lyapunov approach can be used to obtain asymptotically stable MRAS. Finally, the marine propulsion control system based on the proposed MRAS is tested and evaluated using the "Changjing 2" ship. Two case studies show the effectiveness and the salient performance of the proposed control approach on the system uncertainty. The proposed MRAS can operate robustly on the design objective, and the comparison between the proposed method and the traditional PID demonstrates that the newly developed control system outperforms the traditional one.

In order to validate the proposed control model close to a practical environment, the influence of the ship motion will be taken into account in our future work. This is because in a real sea environment, the wave loads may cause not only the hull deformation but also the ship swing motions, including the heave, sway, roll, yaw, pitch and surge. These swing motions strongly influence the control performance of the propulsion system. To address this issue, future work will investigate the combination control method of the ship propulsion system and ship motion.

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