**H∞ Controller and Bumpless Transfer Design for Marine Propulsion System**

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**Abstract:** A method for controller design and switching controller without bump effect has been proposed for a marine propulsion system with diesel engine used as propeller prime-mover. Due to different regimens operation of ship propulsion, it is common practice to design more than one linear controller, each at a different operating point, and to switch between them; which enables the system to be controlled satisfactorily within the whole of its operating range. Each design consists of a feedback H-controller (FB-HC) and an associated bump-less transfer H-controller (BLT-HC). The method is implemented in an auto-tuning procedure and satisfactory results are obtained using real time hardware in the loop simulations (HILS).

**Key-Words:** ship propulsion system, H controller, bumpless transfer

1 **Introduction**  
Controller design for marine systems are mainly based on PID technology [1,2,3,4,5,6], nevertheless, if advanced control strategies are used, some improved results can be obtained. Diesel engine is used as propeller prime-mover for the majority of modern merchant ships. This is due to three major reasons: 1) the superior (thermal) efficiency of Diesel engines, b) large Diesel engines can burn heavy fuel oil (HFO), c) slow-speed Diesel engines can be directly connected to the propeller without the need of gearbox and/or clutch and are reversible. As shortcoming, Diesel engines require a large engine room compared to gas turbines, which can be a problem when extremely large power outputs are required for large high-speed vessels [1,2].

In this work, robust control theory results are applied to design the propulsion control system of a merchant ship. We employ a mathematical model which is a synthesis of different models given in literature [1,2,3,4,5], a nonlinear model which captures the essential characteristics of ship propulsion dynamics and it is used in order to carry out real time hardware in the loop simulations (HILS) [7,8,9]. Linearized models are used to design PID and H∞ controllers [1,2,3,4,10,11] for different operation conditions. To change controller parameters without bump effect we propose a bump-less procedure, which is applicable for controller switching in gain scheduling method used for adapting controller to changes in plant dynamics.

The rest of this paper is organized as follows: in section two the system propulsion model is described, the controller design methodology is outlined in section three, simulation results are depicted in section four, and finally concluding remarks are given in section five.

2 **Propulsion system model**  
The propulsion system consists of two basic control loops, one for propeller pitch (pitch controller) and one for shaft speed (shaft speed controller). The propulsion set-point is performed by a lever, named “the telegraph”, placed on the bridge. Each lever position corresponds, via the combinatory curve, to a pitch setting and to the required rotational speed of the engine. The reference signals are then transmitted to the controller (named governor). The governor inputs are the requested (set-point) and the actual engine speeds, as well as the propeller pitch and its set-point are used. The governor controls the fuel flow to the cylinders in order to maintain the required engine speed.

Diesel engine of the prime-mover is modelled taking into account the following components: 1) the input signal produced by the controller is first converted to an equivalent current signal that drives the actuator. The injection actuator has a time constant $\tau_a$ (as nominal value $\tau_a = 0.1$ sec. is used in simulations) that is dependent on the oil temperature. The output of this unit is the fuel-flow, which is a direct input to the engine. The injection process is
characterised by a dead-time (injection delay) \( \tau_d \), which is a non-linear function of the engine speed \([13,1,2,3]\)

\[
\tau_d = (AN^2 + BN + C)/N^2
\]

which has been estimated to lie within the range:

\[
15/N < \tau_d < 15/N + 60/(Nz)
\]

where \( z \) is the number of engine cylinders.

To model the engine thermodynamic process that determines engine brake torque \( Q_{eng} \), a first order transfer function with thermodynamic gain \( K_{TC} \) and time constant \( \tau_{TC} \) is considered \([2,3,4]\)

\[
\tau_{TC} \dot{Q}_{eng}(t) + Q_{eng}(t) = K_{TC} F_R(t - \tau_d)
\]

where \( F_R \) is the named fuel index (rack) position or fuel-rack position, in this equation \( \tau_{TC} \) is mainly due to the effect of turbo-charging on the power generation process. As nominal value it is used \( \tau_{TC} = 0.25 \) seconds in simulations. Propeller thrust \( T_p \) and torque \( Q_p \) are modelled by means of

\[
T_p = K_T N^2 |\theta| \theta/\theta
\]

\[
Q_p = K_Q N^2 |\theta| \theta
\]

where \( \theta \) is the propeller pitch ratio, \( K_T \) and \( K_Q \) vary with propeller shaft-speed \( N \), and advance velocity \( V_a \), and can be approximated as a n-th order polynomial in advance number \( J_a \). Usually, a first order polynomial

\[
K_Q = K_{Q0} + K_{Q1} J_a, \quad J_a = V_a/(N D)
\]

\[
K_T = K_{T0} + K_{T1} J_a
\]

or second order polynomial are employed \([14,15,16,17]\), where \( D \) is the propeller diameter, and advance velocity \( V_a \) of the water-flow over the propeller disk is related to the ship’s surge velocity \( U \) via wake coefficient \( w \), as \( V_a = (1-w) U \).

In practice, \( K_Q \) and \( K_T \) undergo fluctuations due to variations on shaft rpm and ship surge speed, and this is taken into account in the design procedure for robust controller. In this context, variations on \( K_Q \) can be treated as a disturbance acting on the process instead of considering an uncertain parameter. Fluctuations on \( K_Q \) is one of the major disturbance to marine plant operation and the source of perturbations for the engine rpm \( N(t) \) and fuel index \( F\eta(t) \) signals.

The propulsion plant and surge ship dynamics can be represented by two dynamical equations, the first equation represents the surge motion of the ship,

\[
M \ddot{U} = T_p - T_R
\]

where \( T_R \) is the ship resistance and \( M \) is the effective mass. The second equation represents the rotational motion of the shaft line

\[
J \dot{\phi} = Q_{eng} - Q_p - Q_{ext}
\]

where \( Q_{eng} \) is the engine torque, \( Q_p \) is the propeller demand torque and \( Q_{ext} \) is used to take into account external disturbances.

To take into account waves induced disturbance, which produces a variation in propeller torque, it is used an approximation of the Pierson and Moscovitz spectrum, what is common in the field of marine control engineering. For that, it is used a second-order model driven by white noise \([11,12,13,14]\)

\[
D_w(s) = \frac{\omega_w S}{s^2 + 2\omega_w \xi s + \omega_w^2}
\]

where \( \omega_w \) is the central wave frequency, and \( \xi = 0.15 \) typically. The output \( d(t) \) of the filter \( D_w(s) \) is used as the wave induced propeller torque disturbance, and its magnitude depends on the variance of the white noise employed.

In order to controller design and linear analysis, linearization of the system equations are employed for the vessel sailing under service speed conditions, i.e., the nominal (steady-state) operating points. For a concrete ship, it is necessary to carry out towing tank tests and sea trial experiments, and with the acquired data to adjust parameters and improve equations of the mathematical model. Nevertheless, that approach is not objective in this paper and will be carried out in next works with real marine systems in collaboration with a naval construction company.

In real time simulation studies with hardware in the loop (RT-HILS) the differential equations are solved using fourth order Runge-Kutta method with fixed step size of 1.0 ms. The dead-time is implemented using a circular buffer using the same step time. For RT-HILS we employ EPESC hardware/software environment \([9]\).

### 3 Controller design methodology

The success and widespread use of linear design techniques in control system design can, in part, be attributed to the relative ease of synthesis and implementation of linear controllers, and to the
powerful, intuitive and convenient mathematics associated with linear systems theory [15,16,17]. However, the strengths of these techniques have to be balanced against the fact that all real-world systems are, to some varying degrees, inherently non-linear. This has the consequence that most linear controllers have to be designed around a specific operating point. Variation around this operating point can cause degradation of the performance of the controlled system, even when the engineer employs robust methods of design.

Due to different regimens operation of ship propulsion, it is common practice to design more than one linear controller, each at a different operating point, and to switch between them; which enables the system to be controlled satisfactorily within the whole of its operating range.

The problem of smooth real-time switching between controllers, in the closed-loop control applications, is referred as bump-less transfer (BLT). In general, BLT arises in many cases of practical interest. One of such cases is on-line performance assessment of advanced control laws against the industry standard, typically PID-based. Another case is the attainment of an improved closed-loop system performance via switching between the controllers with the complementary properties, such as the ones separately optimized for tracking and disturbance rejection and/or for the specific set-points to cover the entire operating range of interest. In practice, due to controllers are implemented in software, all their states are available, and bump-less transfer is performed in the steady state to meet safety requirements.

In this paper we propose a design methodology for: 1) feedback H-controller (FB-HC) design, 2) bump-less transfer H-controller (BLT-HC) design. For each FB-HC design, it is obtained its corresponding BLT-HC as it is shown in Fig. 1; where: GcA represents the active FB-HC, GcL corresponds to the latent controller and GcBLT represents the BLT-HC.

The proposed methodology in this paper for H-controller design procedure is given below. It is carried out in automatic form (auto-tuning), and the user does not need to know theoretical fundaments of H control, only needs to know how to adjust two parameters $\beta_H$ and $\hat{\beta}_H$. A third parameter $\sigma_H$ is fixed to a constant value.

**FB-HC design**

Step 1. It is used a model of the plant for controller design, $G_p(s)$. This is obtained from experimental identification, or by linearization in case of the non-linear model of the process be known.

Step 2. It is obtained two parameters associated to plant dynamics: $K_H$ and $\tau_H$. For ship propulsion control, $K_H$ coincides with stationary gain, and $\tau_H$ is the effective time constant.

Step 3. Weighting transfer functions

$$\{W_S(w, \tau_H, \rho_H), W_R(\beta_H), W_T(\sigma_H)\}$$

are calculated using pre-tuning values for adjusting parameters $\rho_H$ and $\beta_H$.

The meaning of these ponderation transfer functions is as follows: $W_S(w, \tau_H, \rho_H)$ (first order transfer function) is used in order to fix closed loop bandwidth; $W_R(\beta_H)$ (zero order transfer function) takes into account control effort, and $W_T(\sigma_H)$ (zero order transfer function) is related with relative uncertainty bound at low frequencies.

Step 4. It is used zero order hold (ZOH) transformation for $G_H(s)$ discretization, with sampling time $T_m$, $G_H(z)$.

Step 5. Inverse bilinear transform is used for w-plane transfer function $G_H(w)$.

Step 6. It is obtained the generalized plant $P_H(w)$.

Step 7. It is solved the following $H_\infty$ optimal problem

$$\|T_{zw}(w)\|_\infty < \gamma, \quad \gamma > 0$$

and H-controller is obtained $G_c(w)$, where

$$T_{zw} = \begin{bmatrix} W_SS & W_RR & W_TT \end{bmatrix}^T$$

Step 8. Bilinear transform is used for obtaining discrete version of the controller, $G_c(z)$. This controller is implemented as a recursive algorithm.

Step 9. Performance and robustness of the control system are analyzed using numerical indicators (R&PI) obtained with $G_c$ and $G_H$ in first phase, and with real process in second phase. If R&PI are satisfactory then finish the design procedure and go to step 10; in other case, modify fine tuning parameters ($\rho_H, \beta_H$) and go to step 3.

Step 10. Finish design procedure for the feedback H-controller (FB-HC).

**BLT-HC design**

For the bump-less transfer H-controller (BLT-HC) design, the same procedure is followed, but in this case the feedback H-controller (FB-HC) is used as plant. In Fig. 1, GcA is the active controller, GcL is the latent controller and GcBLT is the BLT controller designed for GcL. Therefore, GcL is used as plant for GcBLT design. The BLT-HC design procedure is applied even if the GcL is a PID type controller. In
In this case, PID would be used as plant model for the GcBLT calculation.

![Diagram](image)

**Fig. 1.** Switching between controllers with bump-less transfer controller.

## 4 Simulation results

In order to have plants simulated in real time with hardware in the loop support for testing controller designs for different marine control systems (such as heading control, rolling attenuation and propulsion control), we are using mathematical models based on different references [2,3,4,11,12,13,14]. In this paper we present results obtained with adapted data of the SE propulsion system, where additionally transducers from rpm to volt have been included and variable pitch propeller have been considered.

The propulsion power plant of the containership “Shanghai Express” (SE) is considered as a typical case of a merchant ship propulsion system [2]. In this case, illustrative data of propeller speeds are given (for fixed pitch propeller): for fuel index ($F_R$) of 25% propeller rotational speed ($N$) is 59.2 rpm, for $F_R = 50\%$, 75% and 100%, the following speeds are respectively obtained: $N = 74.6$ rpm, 85.4 rpm and 94.0 rpm.

For a given plant condition, the linearized model has the following form:

$$G(s) = \frac{KKg}{(\tau_{TC}s + 1)(\tau_{r}s + 1)(\tau_{p}s + 1)}\exp(-\tau_{d}s)$$

For nominal operation ($F_R = 75\%$ and $N = 85.4$ rpm, $Kg = 1$) is given by

$$G(s) = \frac{43.201}{(s + 10)(s + 4)(s + 1.025)}\exp(-0.21s)$$

$G(s)$ relates rpm of the ship propeller expressed in volts (output of rpm transducer) with the controller signal in volts (input to injection actuator). This response is used for system identification and the following estimated model (first order plus dead time, FOPDT) is obtained. This model is used in practice for PID tuning.

$$\hat{G}_{FOPDT}(s) = \frac{1.052}{1.05s + 1}\exp(-0.52s)$$

FOPDT and linearized models are used for controller design, and the obtained results with PID and $H_\infty$ controller are analyzed with the nonlinear model described before. For controller design, dead time term is approximated as a first order Padé approximation. Advantages of $H_\infty$ controller are basically: 1) easy to design using our procedure implemented in ControlAvH software [18], b) our fine tuning procedure only depends on two parameters ($\rho_H, \beta_H$) and it is based on basic rules, c) control system performance and robustness are improved with respect to PID. Classical drawbacks associated with $H_\infty$ control, such as difficult of the design procedure and high order or the controller are overcome. On one hand, this is due to the fact that a digital fourth/fifth order controller is easy to implement in hard real time for specific digital processor and also for an industrial programmable automata or PLC; and for the other side, the design procedure is transparent for the user, due to he only must take into account the relation between two design parameters ($\rho_H, \beta_H$) and its relation with the control system observed response. No theoretical knowledge about $H_\infty$, control is needed for controller fine tuning.

In Fig. 2, closed-loop system responses for three operation conditions (nominal and two others with ±50% changes in stationary gain for the plant) and a fixed H-controller (designed for $Kg=1$) are shown. These conditions corresponds to changes in stationary gain characterized by $Kg$ parameter, which takes three values: $Kg = 1, 1.5$ and 0.5. Nevertheless, as it can be seen, one response is so slow (for $Kg = 0.5$), and other has excessive overshoot (for $Kg = 1.5$). Therefore, three controllers (Gc1, Gc2 and Gc3) must be designed, one for each plant operation condition. In this case, satisfactory responses (extremely similar behavior) are also obtained for $Kg = 0.5$ and for $Kg = 1.5$. The used parameters for the three FB-HC designs are the following:

- $\rho_H = 10$, $\beta_H = 1$, $\tau_H = 1.05$, $K_H = 1.052$
In Fig. 3, the closed-loop system responses for three operation conditions (Kg = 1, 1.5, 0.5) and their respective H-controllers (designed for each operation condition) are shown. Practically the same RPM responses are obtained for the three controllers. Due to the proximity between the three RPM responses, it seems that only one curve appears in this Figure.

In order to prove BLT controller the following tests are considered: System is in stationary state with controller designed for Kg = 1 (Gc1 as active controller GcA), and it is decided to switch to controller designed for Kg = 0.5 (Gc3 as latent controller GcL). If BLT controller is not used, the bump effect happens as it is shown in Figure 4. If BLT is considered, BLT controller can be activated in different instants: a) when it is decided to change the controller from Gc1 to Gc3, b) one half second before to controller switching. If case a) is considered, BLT controller needs a settling time to reduce differences between uL and u (see Figure 1), and therefore bump effect will not be avoid completely. If case b) is tested, the previous time is used to get that (uL-u) be sufficiently small and bump-less controller switching is obtained. The following parameters have been used for BLT controller (GcBLT in Figure 1):

\[
\rho_H = 0.2, \beta_H = 0.1, \tau_H = 1.05, K_H = 1.052
\]

In this case, the controller Gc3 (designed for Kg = 0.5) is used as plant to control, and the GcBLT is its controller.

Without BLT controller, bump effect is significant as it can be seen in Fig. 4. Resultant effect from switching is an equivalent disturbance. For that, it is employed the BLT controller. In Fig. 5 it can be shown behavior when a BLT (Gc_BLT for Gc3) is used. In this case, controller for BLT (Gc_BLT) is connected 0.5 sec. before controller switching. This time is necessary for controller convergence: uL \rightarrow u (see Fig. 1).

6 Conclusion
A method for controller design and switching controller without bump effect has been proposed for a marine propulsion system with diesel engine used as propeller prime-mover.

Each design consists of a feedback H-controller (FB-HC) and a bump-less transfer H-controller (BLT-HC). The method is implemented in an auto-tuning procedure by means ControlAvH [18]. Satisfactory results are obtained using hardware in the loop simulations (HILS) with EPESC [9]. The employment of our method gives good performance and robustness properties, and bump-less transfer
when switching between controllers are carried out. In next works, experimental marine systems will employ to test our design methodology in collaboration with a marine construction company.

References: