Application of Push-Pull Control Slipping Clutch to the Marine Propulsion System

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ABSTRACT

The use of an electronically controlled slipping clutch is gradually increasing for a marine propulsion system. This system can provide a continuous change of propeller revolution even in a lower revolution zone of prime mover. However, it hardy make the decelerating condition, because the slipping clutch itself can not make the negative torque.

In this paper, the authors have designed a push-pull control system using two slipping clutches in order to improve the above disadvantage of the conventional slipping clutch. For the evaluation of this system, the whole propulsion system including prime mover, clutches, propeller and ship are numerically simulated using the precise mathematical model described here. From several accelerating and decelerating simulations, the following results are pointed out.

1) FPP's revolution can be easily changed in the slipping zone by using this system, although the main engine is governed constant revolution.
2) Astern and ahead maneuvering is also performed by this system.
3) The mathematical model for this simulation including engine, clutch, propeller and ship is useful for the design and evaluation of the whole propulsion system.

As the result, this push-pull controlled slipping clutch system will be very useful for a marine propulsion system.

Key Words : slipping clutch, FPP, push-pull control, harbor maneuvering, astern maneuvering

1. INTRODUCTION

When a ship proceeds into a harbor, a frequent ship-speed change is ordered. A CPP (Controllable Pitch Propeller) is conveniently used for this purpose because of the easily change of the engine output as well as its direction. However, CPP has some disadvantages in the cost of construction, efficiency of propulsion plant, cavitation and noise of propeller. While a FPP (Fixed Pitch Propeller) has not such problems but has not any other control device except the revolution of a prime mover. So, the following control devices of propeller revolution are required.

(1) Electric generator-motor system
(2) Reversible hydraulic converter coupling system
(3) Mechanical transmission system with conventional clutch and gear
(4) Slipping clutch system

The characteristics of these propulsion systems are listed in Table.1 comparing with the CPP system. As for the prime mover, uni-directional engines are used such as medium speed diesel and gas turbine. The electric generator-motor system is sometimes used for special ships. This system however becomes a large scale of plant and requires a significant cost of construction. The efficiency of power plant is not also good. The hydraulic transmission system has the similar tendencies. For the mechanical transmission system, the reduction gear ratio is usually fixed, so that the change of continuous propeller revolution is not obtained particularly under the idling range of prime mover.

The slipping clutch that is electronically controlled with a hydraulic power unit becomes one solution. It easily makes an arbitrary propeller revolution to the accelerating side. It well realizes the propeller revolution by means of adjusting the hydraulic pressure of the clutch disk. However, it can hardly make the decelerating condition, because the slipping clutch itself can not provide the negative torque induced by the propeller.

In this paper, the authors propose an idea of new control system of the slipping clutch in order to improve the decelerating condition as well as the propeller-reversing zone. In this system, two slipping clutches are installed in both forward and reverse revolution gears, then they are controlled continuously in order to get the ordered propeller revolution. This system is called here a "push-pull control slipping clutch system".

2. PUSH-PULL CONTROL SLIPPING CLUTCH SYSTEM

The slipping clutch produces a lower revolution than the direct connecting system so as to slip the clutch disk by means of adjusting the connecting pressure of the disk. The pressure is usually provided by a hydraulic oil system. The clutch disk is filled by lubricating oil to make easy control. The cooling system of the lubricating oil is also provided for absorbing the thermal energy by the friction. For the actual control of the shaft revolution, it is continuously picked up and the pressure of clutch dsk is automatically adjusted in order to get the ordered revolution. This schematic diagram is shown in Fig.1

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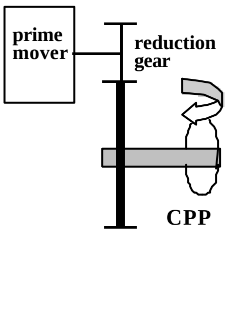
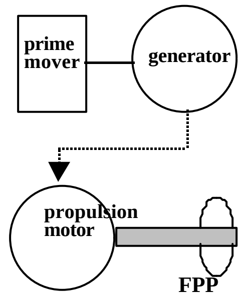
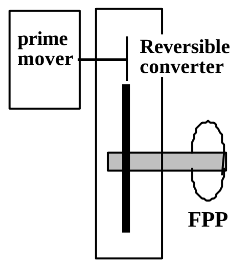
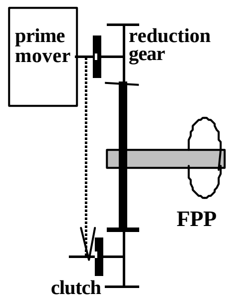
Table 1 Comparison of Propulsion System

Case	Controllable Pitch Propeller	Electric Generator Motor System	Reversible Converter Coupling System	Mechanical Reduction Gear and Clutches
Configuration				
Merits	● provide very low ship speed ● easy to astern/ahead	● provide very low ship speed ● easy to astern and ahead	● provide very low ship speed ● easy to astern and ahead ● small and light weight ● low cost	● small and light weight ● low cost ● high efficiency
Demerits	● heavy weight ● high cost ● dry docking to repair the CPP mechanism ● low efficiency ● cavitation noise	● heavy weight ● high cost ● a lot of ancillary equipment ● low efficiency	● very low efficiency	● do not provide low ship speed

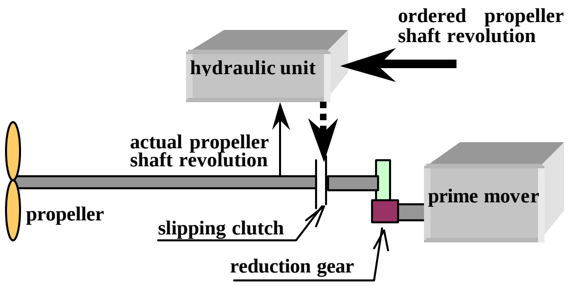


Fig.1 Schematic diagram of conventional slipping clutch

In the accelerating condition as well as steady sailing of ship, the propeller torque is obviously positive, this system is working well by means of slipping the clutch disk. However, in case of the decelerating condition, it is ordered to reduce the propeller revolution, then the pressure of clutch disk goes down until the propeller shaft revolution reaches the ordered one. The propeller torque is also reduced. Soon after, the torque turns to negative. At this moment, the propeller revolution can not reduce any more and just idling released from the prime mover. Although such coasting may be useful for the open sea as saving the fuel energy, it is dangerous particularly in congested harbor and restricted channel.

In order to improve such decelerating condition of ship, the following slipping clutch system is designed as shown in Fig.2. In this system, two slipping clutches are installed in both forward and reverse revolution gears. No.1 clutch system works when the transmitting torque is positive. No.2 clutch system does when it is negative by the lower propeller revolution. Once the propeller revolution is ordered to reduce, the pressure of No.1 clutch disk goes down until the propeller torque turns to negative. After that, the No.2 clutch system begins to produce the negative torque instead of No.1 system.

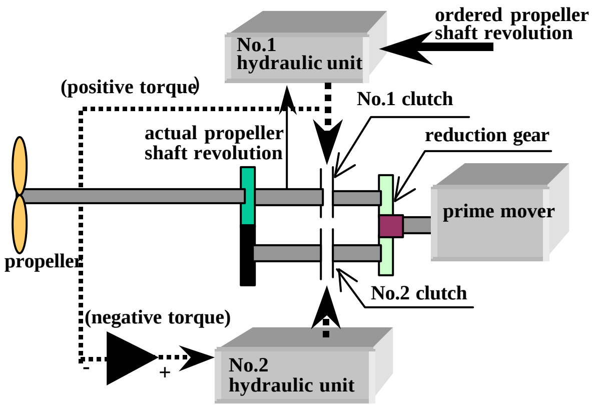


Fig.2 Push-Pull control slipping clutch

Then, the propeller revolution can reduce and reach the ordered one. If the propeller torque comes back to the positive side, the propeller revolution becomes lower than the ordered one. Then, the pressure turns to positive and No.1 clutch works instead of No.2 clutch. Since No.1 and No.2 clutch systems do not work simultaneously, the capacity of hydraulic units and cooling systems are not necessary the twice capacity.

Furthermore, this system can be used in the propeller reversing side without any modification. When ASTERN is ordered, the hydraulic pressure of No.1 clutch disk becomes negative and the clutch is released soon. Then the No.2 clutch system begins to work until the propeller revolution gets steady in reversing side.

The characteristics of the steady propeller-shaft revolution are schematically shown in Fig.3 comparing from the mechanical clutching system.

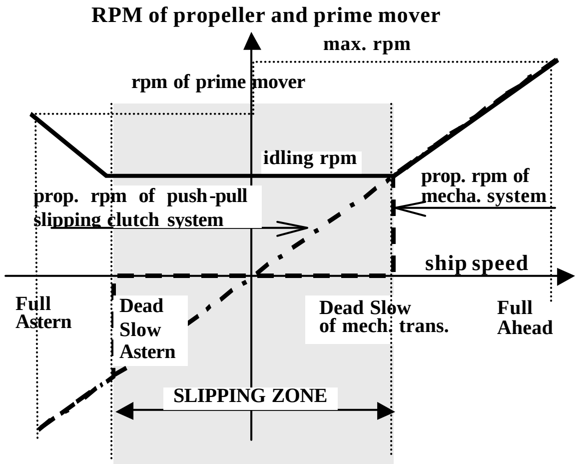


Fig.3 Comparison of steady speed-rpm characteristics between push-pull slipping clutch system and conventional mechanical clutch system.

3. NUMERICAL SIMULATIONS

In order to evaluate how well works the above mentioned push-pull slipping clutch system, numerical simulations are performed with several accelerating and decelerating conditions.

3.1 Mathematical Model for Simulations

(1) Torque of Slipping Clutch

The transmitting torque of the slipping clutch (Qc) is assumed linear with the pressure of clutch disk (p), disk area (A_d) and mean radius of clutch plate (R_m) [1],[2]

$$Qc=\mu A_d R_m\ p \qquad \text{-------------------- (1)}$$

where, μ: coefficient of friction

p : pressure of clutch disk

Although μ varies with slip speed, pressure of disk, temperature of disk and another conditions of clutch disk, it is assumed to be constant for the simulation.

The pressure is controlled by the following simple P-I equation with the difference between ordered propeller shaft revolution (n_p*) and actual one (n_p).

$$p= c_p(n_p{*}-n_p)+c_i \int_0^t (n_p{*}-n_p)dt \qquad \text{------------------- (2)}$$

where, c_p and c_i are the feed back constants.

From eq.(1) and (2), the transmitting torque of these two clutches ($Qc1$),($Qc2$) are expressed as the following function, taking account of some non-linear factors of the friction and dead zone to prevent system flutter around the zero crossing point of torque.

$$Qc = \mu A_d R_m\{c_p(n_p{*}-n_p)+c_i \int_0^t (n_p{*}-n_p)dt\} \qquad \text{-------------- (3)}$$

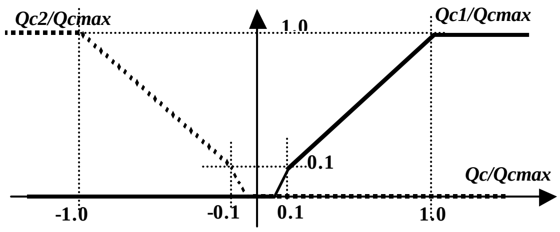


Fig.4 Characteristics of No.1 and No.2 clutch torque

(2) Response of Prime Mover

As for the prime mover, a conventional diesel engine is considered and the slipping clutch is connected to this shaft. The following simple mathematical models are used including the characteristic of turbo charger.

$$2\pi Ie\ \dot{n}_e=Qe\ -Qef\ -(Qc1+Qc2) \qquad \text{------------------ (4)}$$

where, Qf : frictional torque of main engine ($=f_1(n_e)$)

Qe : genarated torque of main engine ($=f_2(R)$)

$$R= c_1(n_e{*}-n_e)+c_2 \int_0^t (n_e{*}-n_e)dt \qquad \text{------------- (5)}$$

c_p and c_i are the feed back constants.

Ie: moment of inertia of main engine

n_e: actual revolution of main engine (rpm)

n_e*: ordered revolution of main engine (rpm)

(3) Response of Propeller shaft

The equation of motion with propeller and propeller shaft can be described as the following form..

$$2\pi Ip\dot{n}_p=(kr1Qc1-kr2Qc2)\ -Qpf\ -Qp(n_p,\ u) \qquad \text{------ (6)}$$

where, Ip : moment of inertia of propeller and propeller shaft

$kr1$,$kr2$: gear ratios of No.1 and No.2 reduction gear

Qp: propeller torque

$= (\rho/2)C_Q\{(1-w)^2u^2+(0.7\pi n_pD_p)^2\}(\pi/4)D_p^3$

$$C_Q= b\sum_{k=0}^{20}\{C_k\ cos(k\beta)+D_k\ sin(k\beta)\}$$

$$\beta= tan^{-1}\{(1-w)u/(0.7\pi n_pD_p)\}$$

u : ship speed

D_p : diameter of propeller

$(1-w)$: effective wake fraction of propeller

ρ : density of sea water

Qpf: frictional torque of propeller shaft

$=Cpf|n_p|n_p$

(4) Response of Ship speed

The equation of forward motion of ship can be described as the following mathematical model.

$$(m+m_x)\ \dot{u}= -Cs|u|u\ +(1-t)kpT \qquad \text{--------------------- (7)}$$

where, $(m+m_x)$:virtual mass of ship in longitudinal direction

Cs : coefficient of ship resistance

kp : number of propeller

T : thrust of propeller

$= (\rho/2)C_T\{(1-w)^2u^2+(0.7\pi n_pD_p)^2\}(\pi/4)D_p^2$

$$C_T= b\sum_{k=0}^{20}\{A_k\ cos(k\beta)+B_k\ sin(k\beta)\}$$

$(1-t)$: thrust deduction factor

3.2 Dimensions of Simulated Ship

The simulated ship is a ferryboat with twin propeller and twin diesel engine. This is not an actual ship but just designed one. The principal dimensions are listed in Table 2. As for the prime mover, two sets of conventional middle speed Diesel engines are utilized. The characteristics of C_T and C_Q are obtained from NSMB's report [3]. They are displayed in Fig.5. The Robinson's curves provided by C_T and C_Q for various ship speed and propeller rpm are shown in Fig.6.

Table 2. Principal dimension of the simulated ship

Hull (ferryboat)	
Length of ship	95.0 m
Breadth molded	17.0 m
Draught molded	4.0 m
Displacement	3,100 t
Propeller (FPP ×2sets)	
Diameter	2.8 m
Main engine (Diesel ×2sets)	
Output (max.)	2,350 kW
rpm (max.)	750 rpm
reduction gear ratio	2.86 (fore & aft.)
Ships speed (max.)	18.0 knots

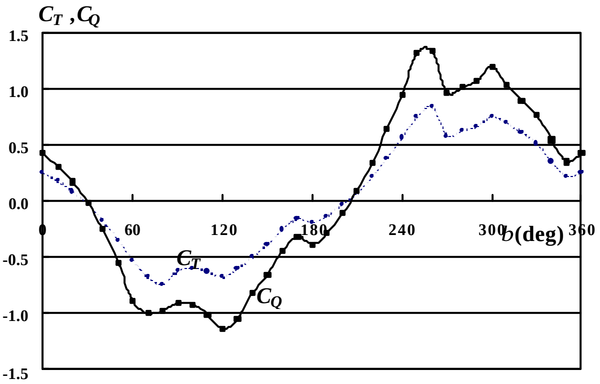


Fig.5 Characteristics of propeller force

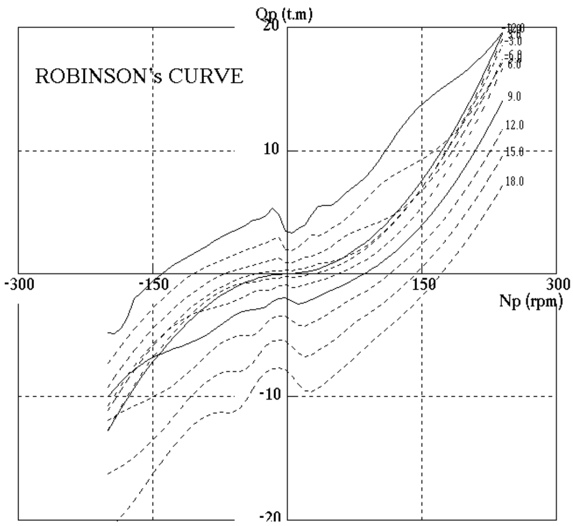


Fig.6 Robinson's curve of the simulated ship.

3.3 Simulated results

Using above mathematical model, typical simulated results are shown. In each simulation, the main engine is governed a constant revolution (=450rpm).

<u>(1) Acceleration and Deceleration of ship</u>

Fig.7 shows the simulated accelerating motion from DEAD SLOW to HALF and deceleration from HALF to STOP. The simulated propeller revolution fairly responds to the ordered one, even in the decelerating zone.

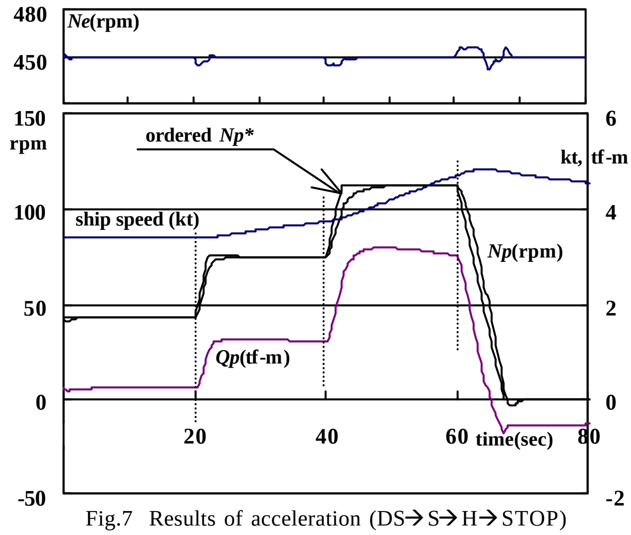


Fig.7 Results of acceleration (DS→S→H→STOP)

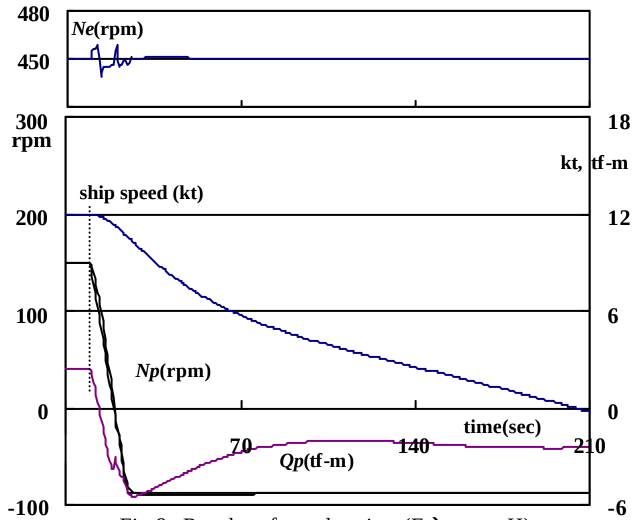


Fig.8 Results of acceleration (F→astern H)

<u>(2) Stopping and Astern of ship</u>

Fig.8 shows the simulated typical stopping maneuver from HARBOR FULL to ASTERN HALF. The propeller revolution well gets astern soon and ship can stop easily.

4. CONCLUSION

The following conclusions are obtained from this research.

1) FPP's revolution can be easily changed in the slipping zone by using this system, although the main engine is governed constant revolution.
2) Astern and ahead maneuvering can be also performed by this system.
3) The mathematical model for this simulation including engine, clutch, propeller and ship is useful for the design and evaluation of the whole propulsion system.

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