Journal of the Korean Society of Marine Engineering, Vol.31, No.3, pp.207~214, 2007(ISSN 1226-9549)

Theoretical Analysis on Transient Torsional Vibrations of Two Stroke Low Speed Diesel Engines

Don-Chool Lee† · Sang-Hwan Kim* · Jung-Dae Yu**

(Manuscript : Received MAR 5, 2007 ; Revised APR 27, 2007)

Abstract : Theoretical analysis on transient torsional vibration was started from the early 1960s for high power synchronous motor application. Particularly, its simulation and measuring techniques in marine diesel engine field have been steadily studied by some classification societies and large marine diesel engine designers. This paper introduces the simulation method on transient torsional vibration of two stroke low speed diesel engine using the Newmark method.

Key words : Two Stroke Low speed Diesel Engine, Transient Torsional Vibration, The Newmark Method

1. Introduction

Theoretical analysis on transient torsional vibration started from the early 1960s for high power synchronous motor application. In the field of marine diesel engines, studies have been conducted by some ship classification societies (notably Det Norske Veritas) and designers of low speed diesel engine (MANB&W, SULZER)[1], [2]. Consequently, they made good progress in 1990s. Contrary to middle/high speed diesel engines, propulsion shafting system using low speed diesel engine has relatively large transfer torque and frequently sets a barred range prohibiting continuous operation at critical speed. This is in order to protect the shaft system from cumulative fatigue stresses, thus it is worth to conduct study. In this study, QPT (Quick pass technique) was applied on diesel engine for marine and land use generator where barred speed range exists. Additionally, theoretical simulation was conducted for engaging and disengaging of clutch in the propulsion system having reduction gears. The Newmark direct integral approach was used as the analysis method[3].

2. Analysis of Theoretical Transient Torsional Vibration

The complex equation of motion for torsional vibration model on diesel engine

† Corresponding Author(Mokpo Maritime University Faculty of Marine Engineering The Republic of Korea), E-mail:ldcvib@mmu.ac.kr, Tel : 061)240-7089
* Mokpo Maritime University Division of Marine Engineering The Republic of Korea
** Hyundai Heavy Industries Engine & Machinery Division The Republic of Korea

is simply expressed as shown in Equation (1). In general, calculation model is simplified to analyze transient torsional vibration. This is obtained after necessary natural frequencies and vibration modes are identified through calculation of undamped free vibration neglecting damping and exciting force. It is then substituted by equivalent mass system [4], [5]. However, there are about 70 more or less nodal points in the torsional vibration calculation model, and it is not yet focused by the recently developed PC processing speed. In this study, the analysis was conducted without substitution by the simplified equivalent mass and spring system to secure reliability and accuracy.

$$[M]\{\ddot{\theta}\}+[C]\{\dot{\theta}\}+[K]\{\theta\}=\{T\} \tag{1}$$

$[M]$: moment of inertia matrix
$\{\theta\}$: angular amplitude vector
$[C]$: torsional damping matrix
$\{T\}$: exciting torque
$[K]$: torsional stiffness matrix

Several methods can be employed for the analysis of transient vibration. In this study, the Newmark direct integral method was used so that the torsional dynamic stiffness coefficients for flexible coupling will be transferred and non-linearly changed by vibratory torques[6] and flexibility will be secured in application of damping and exciting torque. This was arranged into equations (2)~(4). Here, it was found that after change in the coefficients and comparison with other similar direct integral method, there was little difference in the results.

$$\{\dot{\theta}\}_{t+\Delta t}=\{\dot{\theta}\}_t+\left[(1-\varepsilon)\{\ddot{\theta}\}_t+\varepsilon\{\ddot{\theta}\}_{t+\Delta t}\right]\Delta t \tag{2}$$

$$\{\theta\}_{t+\Delta t}=\{\theta\}_t+\{\dot{\theta}\}_{t+\Delta t}\cdot\Delta t+\left[(0.5-\eta)\{\ddot{\theta}\}_t+\eta\{\ddot{\theta}\}_{t+\Delta t}\right]\Delta t^2 \tag{3}$$

$$[M]\{\ddot{\theta}\}_{t+\Delta t}+[C]\{\dot{\theta}\}_{t+\Delta t}+[K]\{\theta\}_{t+\Delta t}=\{T\}_{t+\Delta t} \tag{4}$$

Here, ε, η are parameters to get accuracy and stability, and Δt is the time increment. Substitution of equations (2) and (3) for vibration equation (1) is arranged into equations (5)~(7).

$$[\overline{K}]\{\theta\}_{t+\Delta t}=\{\overline{T}\}_{t+\Delta t} \tag{5}$$

$$\{\ddot{\theta}\}_{t+\Delta t}=a_0\left[\{\theta\}_{t+\Delta t}-\{\theta\}_t\right]-a_2\{\dot{\theta}\}_t-a_3\{\ddot{\theta}\}_t \tag{6}$$

$$\{\dot{\theta}\}_{t+\Delta t}=\{\dot{\theta}\}_t-a_6\{\ddot{\theta}\}_t+a_7\{\ddot{\theta}\}_{t+\Delta t} \tag{7}$$

$$[\overline{K}]=[K]+a_0[M]+a_1[C]$$

$$\{\overline{T}\}_{t+\Delta t}=\{T\}_{t+\Delta t}+[M]\left[a_0\{\theta\}_t+a_2\{\dot{\theta}\}_t+a_3\{\ddot{\theta}\}_t\right]+[C]\left[a_1\{\theta\}_t+a_4\{\dot{\theta}\}_t+a_5\{\ddot{\theta}\}_t\right]$$

$\varepsilon \geq 0.5,\ \eta \geq 0.25(0.5+\varepsilon)^2$

$a_0 : 1/(\eta\Delta t^2),\ a_1 : \varepsilon/(\eta\Delta t),\ a_2 : 1/(\eta\Delta t),\ a_3 : 1/(2\eta)-1$

$a_4 : \varepsilon/\eta-1,\ a_5 : 0.5\Delta t(\varepsilon/\eta-2),\ a_6 : \Delta t(1-\varepsilon)$

$a_7 : \varepsilon\Delta t$

3. Analysis of Diesel Engine Transient Torsional Vibration on Three Different Applications

The study conducted theoretical simulation on three different diesel engine application that is; 1) marine diesel engine with fixed-pitch propeller, 2) generator engine for diesel power plant and 3) marine diesel engine with reduction gear and clutch.

3.1 Marine propulsion engine with fixed pitch propeller application Ⅰ

In torsional vibration calculation application Ⅰ, the marine shaft system with fixed-pitch propeller was used. The specification of engine and propeller is shown in Table 1. The 6th order critical torsional vibration speed is 51.2 rpm and the barred speed ranges from 46~56 rpm. Fig. 1 shows the result of converting gas pressure in the cylinder and the inertia force of the piston into torque at critical speed, which acts as major exciting torque to torsional vibration. Fig. 2 shows the result of torsional vibration calculation in the quasi-steady state at 51.2 rpm of critical speed as almost similar with the result of analysis in the frequency domain. Advanced technology is required for 'Quick-passing (QP)' to rapidly pass through a barred speed range, and it depends mainly on external conditions[7]. The time taken for QP is influenced by the load of the engine, type and design margin of the propeller, performance of the turbocharger attached to the engine, vessel type, and sea conditions. Thus, it can be hardly estimated and depends upon the authors' experience in this study. Three (3) seconds 'run-up' was set as a basis to reduce QP time to the maximum as shown in Fig. 3, and theoretical simulation was conducted on the assumption that the longest QP time within normal design criteria was about 20 sec as shown in Fig. 4. Maximum and minimum vibratory torques were described for each case in Table 2. Here, it is found that about 20% of the vibratory torque was reduced during the 3 sec QP compared in the quasi-steady state, while a 10% reduction in vibratory torque was attained in a 20 sec QP. This shows that the shorter the QP time, the higher is the improvement effect; however, it still has limitation on relying upon external conditions. In this engine model, the vibratory torque was reduced by approximately 15% from that of the quasi-steady state [8] as per measurement result during sea trial.

Table 1 Specification of 6S60MC propulsion engine

Engine	Type	6S60MC
	Cyl.bore×stroke	600 × 2,292 mm
	Power at MCR	15,400 bhp× 97 rpm
	Pmi at MCR	19.3 bar
	Recipro. mass	5,559 kg/cyl
	Firing order	1-5-3-4-2-6
	Conn. ratio(r/l)	0.436
	M.O.I(total)	69,538 kg·m^2 (49 %)
	Minimum speed	20 rpm
	Weight	500 ton
Propeller	Type	Fixed pitch propeller
	Dia	7,800 mm
	Dia of shaft	550 mm
	No. of blade	4 ea
	M.O.I(in water)	71,482 kg·m^2 (51 %)
	Weight	45 ton

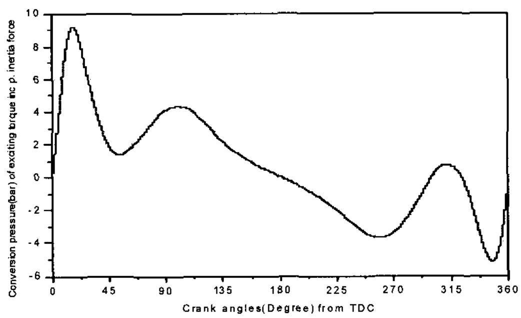


Fig. 1 Exciting torques during one revolution at critical speed of 6S60MC propulsion engine

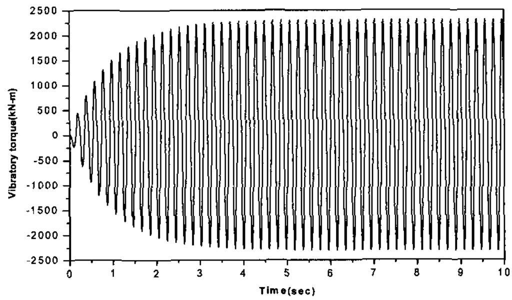


Fig. 2 Simulated vibratory torques for propeller shaft of 6S60MC engine at critical speed

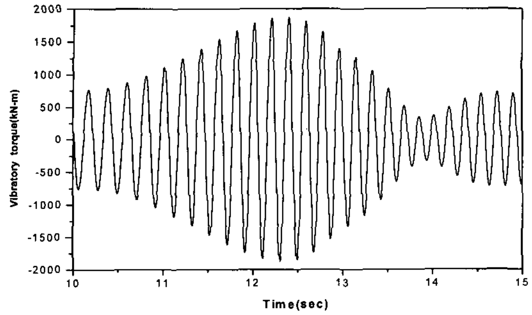


Fig. 3 Simulated vibratory torques during quick passing (3 sec run-up) of critical speed for 6S60MC engine propeller shaft

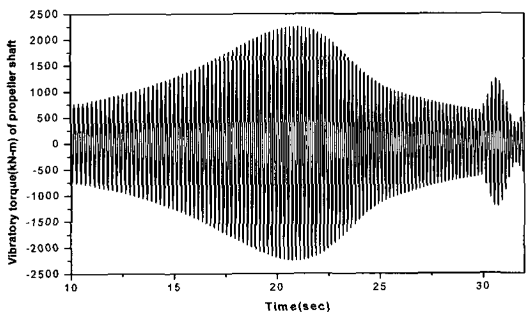


Fig. 4 Simulated vibratory torques during quick passing (20 sec run-up) of critical speed for 6S60MC engine propeller shaft

Table 2 Vibratory torques during Quick-passing for 6S60MC propulsion engine

QP time	Max. torque (kN-m)	Min. torque (kN-m)
Steady	2312.9 (100 %)	-2312.9 (100 %)
3 sec	1872.2 (80.9 %)	-1874.9 (81.1 %)
20 sec	2249.9 (97.3 %)	-2248.0 (97.2 %)

3.2 Generator engine for diesel power plant application II

In torsional vibration calculation application II, the generator engine for land-use large diesel power plant was used. The specification of engine and generator is shown on Table 3. In this engine, uneven crank angles were designed[4] to reduce the 6th order X-guide force moment and 4th order torsional vibration at synchronized speed, which results in sharp increase of torsional vibration at about 77.4 rpm of the 5th order (resonance point). Fig. 5 shows the result of vibratory torques calculation in the quasi-steady state at critical speed. A barred speed range of 72~83 rpm is set to protect the shafting system and reduce structural vibration of the engine body for QP. For generator engine, engine speed is slightly influenced by turbocharger under no-load conditions but moment of inertia for the generator rotor is relatively larger compared to the propeller in model 1, thus more time is taken for QP at critical speed. Fig. 6 shows the result of theoretical calculation on the assumption that the time taken for 'run-up' at a barred speed range is 10 sec on the basis of data obtained from initial field test.

Table 3 Specification of 12K90MC-S generator engine

Engine	Type	12K90MC S
	Cyl.bore×stroke	900 × 2,300 mm
	Power at MCR	70,430 bhp× 103.4 rpm
	Pmi at MCR	18.3 bar
	Recipro. mass	16,885 kg/cyl
	Firing order	1-5-12-7-2-6-10-8-3-4-11-9
	Conn. ratio(r/l)	0.364
	M.O.I(total)	435.85 ton·m^2 (9 %)
	Idling speed	50 rpm
	Weight	1,826 ton
Generator	Dia of rotor	9,425 mm
	Min. dia of shaft	1,050 mm
	M.O.I(rotor)	4,564.00 ton·m^2(91 %)
	No of poles	58 ea
	Weight(rotor)	349 ton

At 'run-down' for marine engine, speed may be rapidly dropped due to high resistance of the propeller and exciting torque is reduced since additional power is unnecessary. So vibratory torques are lower compared to 'run-up'. However, for land-use generator engine, the moment of inertia for the generator rotor is very large and damping energy can hardly be expected under no-load conditions. In addition, a longer time is taken at 'run-down' and it is just considered as a quasi-steady state. Considering this, Fig. 7 shows the result of theoretical calculation on the assumption that the initial 'run-down' time is 60 sec. Taking all of these into consideration, torsional vibration may be reduced to some degree at 'run-up' as shown in Table 4. On the other hand, it is hardly expected at 'run-down' and thus reducing the exciting torque is desirable.

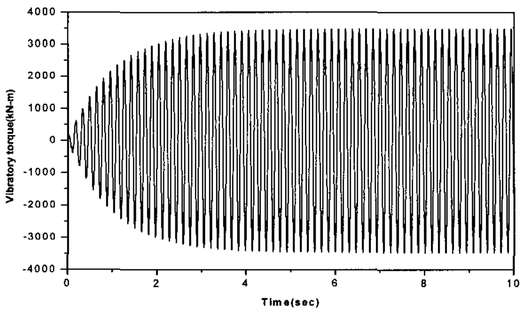


Fig. 5 Simulated vibratory torques for generator shaft of 12K90MC-S engine at critical speed

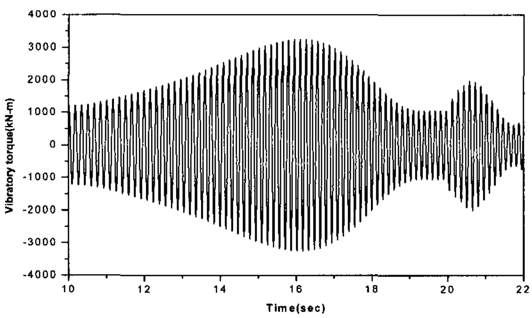


Fig. 6 Simulated vibratory torques during quick passing (10 sec run-up) of critical speed for generator shaft of 12K90MC S engine

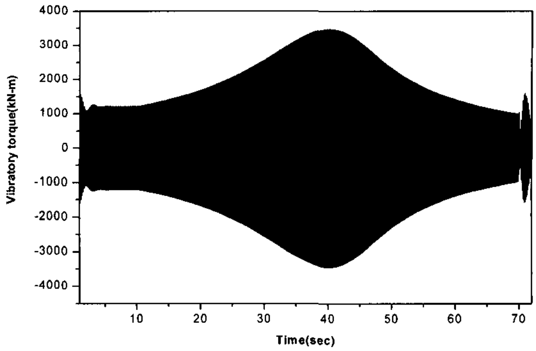


Fig. 7 Simulated vibratory torques during quick passing (60 sec run down) of critical speed for generator shaft of 12K90MC-S engine

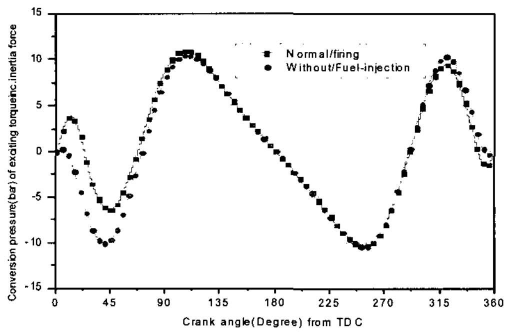


Fig. 8 Exciting torques at the critical speed of 12K90MC-S generator engine

Lately, electronic fuel injection system without camshaft has been put to practical use on two stroke low speed engine[9], [10]. As shown in Fig. 8, operation of the engine just under external pressure at 'run-down' without injecting fuel will reduce the 5th order exciting torque up to about 30% in theory.

Table 4 Vibratory torques during Quick passing for 12K90MC-S generator engine

QP time	Max. torque(kN-m)	Min. torque(kN-m)
Steady	3475.7 (100 %)	-3475.7 (100 %)
10 sec	3256.87 (93.7 %)	-3251.8 (93.6 %)
60 sec	3461.0 (99.5 %)	-3460.6 (99.5 %)

3.3 Marine propulsion engine with reduction gear and clutch application Ⅲ

In torsional vibration calculation application the 2-unit2-shaft large propulsion diesel engine for oil tanker with reduction gear and clutch was used. The specification of the engine, reduction gear and propeller is shown in Table 5. Several operation modes exist according to engine operation and the calculation for condition of 'clutch-in' at 43 rpm of engine speed is shown in Fig. 9. Clutch working oil pressure was calculated on the assumption that it was continued for 3 seconds at 500 kN·m/s and it significantly showed severe speed variation of diesel engine and propeller. As shown in Fig. 9, it is supposed that the maximum engine speed is controlled by the engine governor under actual operating conditions, so variation will be less than the analyzed value. Furthermore, damping and dynamic characteristic of the clutch may not be exactly investigated and only a simple damping was considered. Fig. 10 shows the analysis result on the assumption that duration of engine speed from 25 rpm to engine stop is 30 sec. It can be noted that at this time, the fuel was automatically shut off and only an external pressure was allowed as shown in Fig. 11. The coupling damping was set lower than the general high peak frequency significantly. Particularly, as the critical engine speed is 15.4 rpm of the 7th resonance point and the vibratory torque transmitted to the reduction gear exceeds the 10% (144 kN·m) normal torque recommended by the gear manufacturer as shown in Fig. 10, hammering is expected to occur. It is presumed that there will be

Table 5 Specification of 7S60MC-C propulsion engine

Engine	Type	7S60MC C
	Cyl.bore×stroke	700 × 2,400 mm
	Power at MCR	21,470 bhp× 105 rpm
	Pmi at MCR	20.3 bar
	Recipro. mass	5,003 kg/cyl
	Firing order	1-7-2-5-4-3-6
	Conn. ratio(r/l)	0.436
	M.O.I(total)	95,585 kg·m^2 (54.6 %)
	Min. speed	22 rpm
	Nominal torque	1,436 kN·m
	Weight	410 ton
Reduction gear	Maker	Schelde
	Reduction ratio	1.59(105 : 66 rpm)
	Gear hammering	10 % of Nominal torque
Flexible coupling	Maker	Vulkan D631ES
	Nominal torque	2,000 kN·m
	Permissible v.t.	600 kN·m
	Weight	13,839 kg
Propeller	Dia.	9,000 mm
	Dia of shaft	700 mm
	M.O.I(in water)	201,105 kg·m^2
	M.O.I(Equivalent)	79,457 kg·m^2 (45.4 %)
	No. of blades	4
	Weight	47 ton

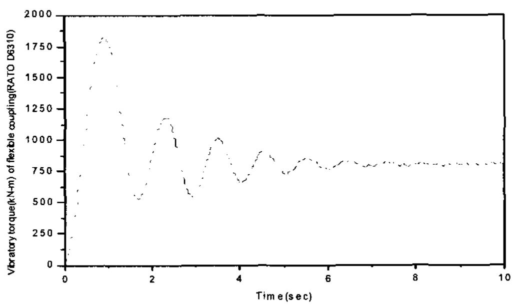


Fig. 9 Vibratory torque of flexible coupling during clutch-in for 7S60MC-C propulsion engine

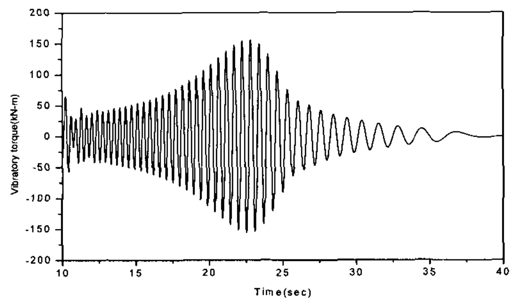


Fig. 10 Vibratory torque of flexible coupling during engine stop for 7S60MC-C propulsion engine

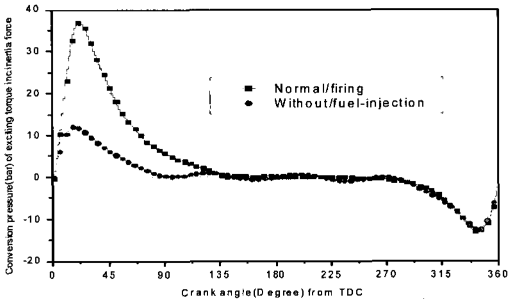


Fig. 11 Exciting torques at critical speed of 7S60MC-C propulsion engine

no significant problem as the 7th resonance point will be passed the moment the engine is operated. However, when the engine stops, a longer time is basically required by the inertial torque of the shafting and hence, reduction of torsional vibration by applying QP is actually impossible. Therefore it was expected that the clutch will be disengaged at the section exceeding 10% of normal torque when the engine stops to change the vibration mode and protect the reduction gear from hammering.

4. Conclusion

Recently, cumulative fatigue failure of shafting system resulting from transient torsional vibration has been examined by CIMAC[11] (International Council on Combustion Engines), DNV[12] (Det Norske Veritas), et al and there are proposals to regulate it. With this trend, software to theoretically solve and examine the transient torsional vibration is expected to be developed and adopted to practical use. Results of this study are summarized as follows:

1) High reliability and accurate calculation result could be secured from a 'full-model' transient torsional vibration instead of simplified model.

2) The accuracy of transient torsional vibration analysis depends greatly on the external conditions during engine operation and the broad experience on-site than the analysis software itself. Therefore, it is expected that these aspects will be methodically obtained and arranged into the database.

3) The vibratory torque or torsional stresses for the two stroke low speed engine is relatively given much more attention and a barred speed range was emphasized in this study, it is also recommended that it will be incorporated into four stroke middle/high speed engine as well.

Reference

[1] MAN B&W, 1998, Vibration Characteristics of Two stroke Low Speed Diesel Engines, MAN B&W.

[2] H. Keller, 1996, "Simulation of Transient Torsional Vibrations in Marine Diesel Propulsion Plants", New Sulzer Diesel.

[3] Klaus-Jürgen Bathe, 1996, Finite Element Procedures, Prentice Hall, pp.768~837.

[4] D. C. Lee, 1999, "Unstable Torsional Vibrations on the Propeller Shafting with Diesel Engine Generator", Journal of the Korean Society for Noise and Vibration Engineering, Vol.9 No.5 pp. 936~942.

[5] D. C. Lee, J. D. Yu and J. K. Kim, 2000, "Control of Torsional Vibration for Diesel Power Plant using Uneven

Crank Angles ", Journal of the Korean Society for Noise and Vibration Engineering, Vol.10 No. 4 pp. 655~661.

[6] Vulkan, 2000, RATO Highly flexible Couplings, Vulkan Kupplungs. und Getriebebau

[7] S. Sasaki and M. Takagi, 1995, "Some Consideration for Behaviour of Propulsion Diesel Engine in Transient Condition", Journal of the M.E.S.J. Vol.30 No.9, pp. 651~660.

[8] Dynamics Lab. of MMU, 2002, Technical Report (Torsional Vibration measurement for Hyundai-Samho SH133), Document. No. MDL-02041

[9] MAN B&W, 2000, The Intelligent Engine : Development Status and Prospects, P.375-00.11, MAN B&W.

[10] Wärtsilä NSD, 2001, The Common Rail Low-Speed Engine, P.21.01.07.40.

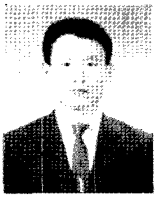
[11] CIMAC WG14, 2002, "Harmonization of the rules Regarding the Vibration Analysis of Two Stroke Diesel Engine Installations", CIMAC

[12] DNV, 2001, Calculation of Shafts in Marine Application, DNV Classification Notes No. 41.4

Author Profile

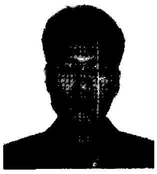
Don-Chool Lee

Date of birth : January 1959.
Bachelor of Mechanical Engineering, Chonnam National University, 1981; Ph. D of Mechanical Engineering, Korea Maritime University, 1995; Engine & Machinery Division, Hyundai Heavy Industries, 1983 to 1999; Associate Professor, Mokpo Maritime University, 1999 to present.



Sang-Hwan Kim

Date of birth : October 1967.
Bachelor of Mechanical Engineering, Chosun University, 1985 to 1992; Master of Marine Engineering, Mokpo Maritime University, 2000 to 2002; Ph. D course (ongoing), Division of Marine Engineering, Mokpo Maritime University, 2002 to present.



Jung-Dae Yu

Date of birth : February 1972.
Bachelor of Mechanical Design Engineering, Seoul National University, 1992 to 1996; Master of Mechanical Engineering, Korea Maritime University, 2002; Engine & Machinery Division, Hyundai Heavy Industries, 1996 to present.