A Prediction Method of Fatigue Life at The Rudder Stock

Ahmad S. Mujahid,[a,*] and Wibowo H Nugroho,[b]

[a)]*Research Engineer, Laboratory for Hydrodynamics Technology – Agency for Assessment and Application Technology, Indonesia*
[b)]*Sr.Research Engineer, Center of Technology for Maritime Industrial Engineering – Agency for Assessment and Application Technology , Indonesia*

*Corresponding author: ahmad.syafiul@bppt.go.id; ahmadsmujahid@gmail.com

Paper History

Received: 4-February-2017
Received in revised form: 17-March-2017
Accepted: 30-March-2017

ABSTRACT

The occurrence of the sinking ship in Indonesian water has to be put in consideration. One of the reasons is due to uncontrollable ship caused by a broken rudder stock. This condition will be dangerous when the high wave excitation can makes the ship capsize. In this paper a prediction of fatigue life of the rudder stock due to the ship roll motion at random sea wave is presented. The rudder stock is made from carbon steel (ST41) and installed on LCT Ship (L = 64 m, B = 12 m, T = 3.6 m). The prediction starts with a numerical simulation of ship roll motion in Indonesia water with sea states 5 (Hs = 3 m, Tp = 8.3 s), then the time history of the roll motion is used to calculate the time history of flexural moment of the rudder stock structural component. By knowing this flexural moment, the time history of flexural stress also can be found. This flexural stress time history can then applied to Palmgren – Miner formula to predict the life of the rudder stock, based on SN-Curve of the ST41. From this study it is found that the life of the rudder stock is 1.8 years.

KEY WORDS: *Fatigue, Rudder Stock, Palmgren-Minner*

NOMENCLATURE

A	Variable in spectrum definition
A_{44}	Added inertia coefficient for roll
B	Vessel beam, variable in spectrum definition
B_{44}	Damping coefficient for roll
C_{44}	Hydrostatic restoring coefficient for roll
F_4	Roll Exciting moment at the encounter frequency
I_4	Mass inertia of vessel in roll
$\frac{d\ddot{\theta}}{d^2t}$	Acceleration, the second derivative of roll time domain
η_4	Instantaneous roll displacement
ω	Wave frequency
ω_e	Encounter frequency
I_{xx}	Ship Mass Moment of inertia
I_x	Inertia at shaft area normal to Z-axis
$\hat{y}$	Distance from center shaft Z-axis to edge
N_{fi}	Number of cycles to failure
n_i	Number of cycles corresponding of m block

1.0 INTRODUCTION

News media in Indonesia report so many ship accidents occur in Indonesia water. One of the causes of the accident is the uncontrollable ship on the high wave due to rudder stock failure. This situation can sometimes lead to sinking the ship. Examples of the sinking ship are KM Citra Mandala Bahari, KM Surya Makmur Indah, KM Jonson, KM Arta jaya, and KMP Belinda. Such a failure on the rudder stock makes ship failed to operate properly, she will never arrive at the destination on schedule. When the accident was occured in the sea, this will make a ship brought by current and wave. This can lead to many worst conditions such as a ship sink, fuel tank leaks, oil spill into the sea, and environmental pollutions. Therefore, prediction of such a failure is very urgent because a small component failure such as the rudder stock will make high losses for human safety, economic factor, and environmental.

Some researchers have conducted studies in developing the method of predicting the fatigue life by applying cumulative - damage rules. A method used in Nugroho et al (2004) on predicting the fatigue life of mooring lines has been found useful also for this study. Nugroho et al. (2011) also presented a same method to predict the fatigue life for the wooden hull structure of the outboard motor boat. The fatigue life of wooden hull structures of the outboard motorboat due impact loads of the porpoising can be predicted by applying cumulative - damage rules. In 2012, Safety Investigation Authority Finland produced the report for analyzing Breaking of the Starboard Rudder Shaft in the Aland Archipelago on 22

November 2009. On this report the fatigue life of rudder shaft worsened by the present of stress corrosion cracking that made the bending fracture of the rudder shaft possible. The failure started first in the production stage a sharp, 0.3-mm-deep notch was left on the starboard shaft then there was sustained galvanic corrosion in the bearing housings caused by the structure of the pint bearings. These two together would have been enough to cause the failure, but the breaking process was significantly accelerated by a third factor, i.e. the resonant vibration of the rudder. Specifically the fatigue analysis of drive shaft have been investigated by Gujaran et al. (2014) in which the main object of analysis is to investigate the stress and deflection of the drive shaft subjected to combine bending and torsion, and check the fatigue life as well as comparing the result with analytical calculation.

Figure 1: LCT ship and its rudders as a research object

This paper presents a method to predict fatigue failure of rudder stock by combining numerical simulation and strength of material analysis with application of Palmgren - Minner Formula. A case study of LCT ship on Figure 1 is selected on this paper to predict the fatigue life of her rudder stock. The method for predicting of the failure is presented on the flow diagram that presented in the Figure 2.

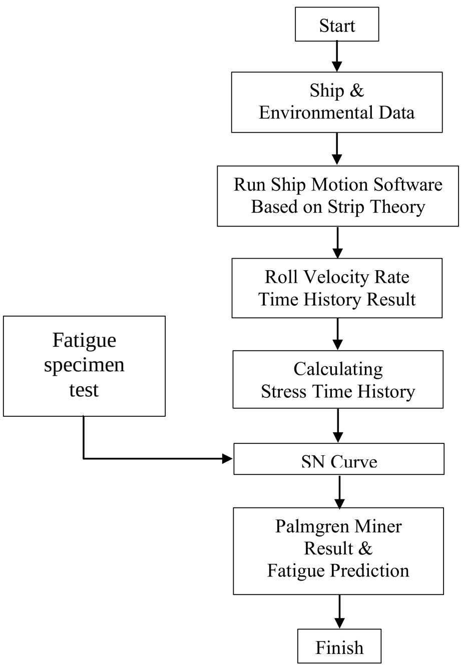

Figure 2: Prediction step of fatigue failure of a rudder stock

2.0 THEORY

2.1 Motion Numerical Simulation

The numerical simulation is made by solving the rolling motion equation below [1]:

$$I_{xx}\frac{d^2\theta}{dt^2} + b\frac{d\theta}{dt} + c\theta = M_0(\omega_e) \qquad \text{.................. (1)}$$

The equation above is approximated using strip theory method.

The roll motion is obtained by using RAO calculated from the numerical simulation. This RAO or Transfer Function value of the roll can be calculated by taking the ratio between the spectra density of the roll motion and spectra density of wave to each encounter frequency waves. The generic formula for RAO can be represented as below:

$$H_\theta = \frac{u_a(\omega_e)}{\zeta_a(\omega_e)} = \sqrt{\frac{S_{\theta\theta}(\omega_e)}{S_{\zeta\zeta}(\omega_e)}} \qquad \text{.................. (2)}$$

where:
H_θ = response function of roll motion,
$U_a(\omega_e)$ = amplitude of frequency ω_e of signal θ,
$\zeta_a(\omega_e)$ = amplitude of frequency ω_e of wave elevation
$S_{\theta\theta}(\omega_e)$ = spectra density of signal θ,
$S_{\zeta\zeta}(\omega_e)$ = spectra density of wave elevation ζ.
The spectra density of wave is JONSWAP spectra type. The spectra formula represented below [3]:

$$S_{JONSWAP\zeta}(\omega) = 0.658\ S_{ITTC\zeta}(\omega)\ C(\omega) \qquad \text{.................. (3)}$$

where

$$S_{ITTC\zeta}(\omega) = \frac{A}{\omega^5}\exp\left(\frac{-B}{\omega^4}\right) \qquad \text{.................................... (4)}$$

The solution of equation (1) is solved in frequency domain (RAO) using Software Seakeeper version 11.12. To obtain the roll time history the frequency domain is transformed to time domain using IFFT (Inverse Fast Fourier Transfer) formula below [5]:

$$x_j = \frac{1}{N}\sum_{n=0}^{N-1} F_n e^{\frac{2\pi i}{N}nj} \qquad \text{.................................. (5)}$$

The Equation (4) above is solved numerically by programming it into Matlab 9. The design drawing of rudder shaft in this study is shown in Figure 3. And the principal dimension and environmental data are presented on Table 1.

JSOse | Received: 4-February-2017 | Accepted: 30-March-2017 | [(9) 1: 1-6]
Published by International Society of Ocean, Mechanical and Aerospace Scientists and Engineers, www.isomase.org., ISSN: 2442-6415

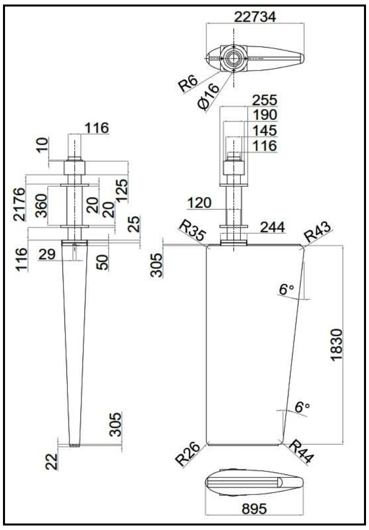

Figure 3: Rudder stock design drawing

Table 1: LCT Ship main dimension and Environmental Data

Measurement	Value	Units
Displacement	3000	Ton
Draft Amidships	3.6	m
Immersed depth	4.8	m
WL Length	64	m
Beam max extents on WL	12	m
Block coeff. (Cb)	0.891	
Wave Height (Hs)	3	m
Wave Period (Tp)	8.3	Sec

This numerical simulation performed in order to obtain the response amplitude operator (RAO) roll motion of the LCT ship. The axis system of the motion is shown on Figure 4. The LCT Ship was assumed to operate in the sea with environmental conditions of JONSWAP spectrum with Hs = 3 m Tp = 8.3 seconds or a range of sea state 5. Wave direction was in a beam sea condition. The wind speed and wave current were ignored in this simulation.

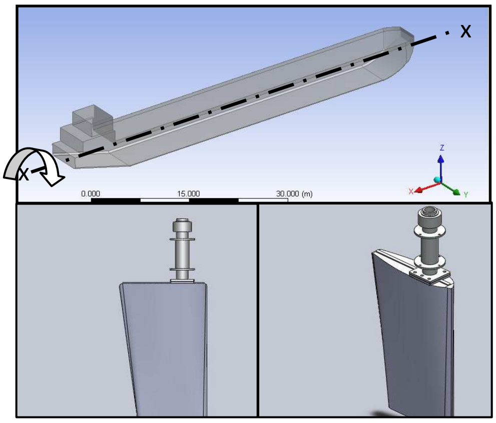

Figure 4: Ship and rudder stock design modeling

2.2 Rudder Stock Stress Calculation

In this study the irregular stresses that acting on the rudder stock was resulted from the roll motion of the LCT ship in the beam sea condition. The flexural Stress at rudder stock (shaft) which shown on Figure 5 can be calculated by mechanical formula below:

$$\sigma(t) = \frac{\dfrac{d\ddot{\phi}}{d^2 t}.I_{x'x'}}{I_x / \bar{y}} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \text{(6)}$$

where $I_{x'x'}$ is mass moment inertia of rudder stock (include the rudder) from CG, and area moment of inertia I_x of rudder stock can be calculated by formula below:

$$I_x = \frac{\pi r^4}{4} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \text{(7)}$$

Where *r* is the radius of the rudder stock

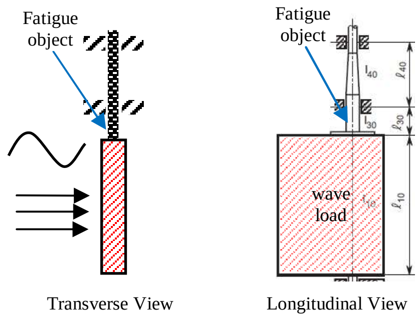

Figure 5: Spade Rudder type of LCT Ship

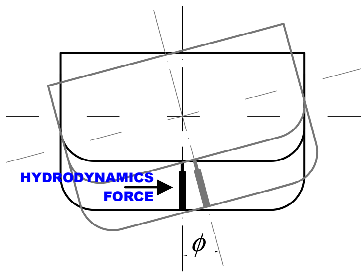

Figure 6: Hydrodynamics force to impact ship rudder

Since the roll angle θ is the same as the angle of rudder stock ϕ (see Figure 6), the roll angle time history can be applied to obtain the flexural moment that acting on the rudder stock.

2.3 Fatigue Analysis Method

This rudder stock are frequently object of the cyclic loads resulted from roll motion, and this cyclic stresses can make physical damage to the steel. When stresses although below the steel's ultimate strength, the microscopic damage can accumulate with continued cycling until it develops into a crack that leads to failure of the rudder stock Fatigue is the process of damage and failure due to cyclic loading. Palmgren-Miner cumulative damage rule is a simple criterion for predicting the extent of fatigue damage induced by a particular block of constant amplitude cyclic stresses, in a loading sequence consisting of various blocks of different stress amplitudes, and the rule is applied on the calculation for metal structural. Palmgren-Miner formula shown below:

JSOse | Received: 4-February-2017 | Accepted: 30-March-2017 | [(9) 1: 1-6]
Published by International Society of Ocean, Mechanical and Aerospace Scientists and Engineers, www.isomase.org., ISSN: 2442-6415

$$\sum_{i=1}^{m} \frac{n_i}{N_{fi}} = 1 \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \text{(8)}$$

3.0 RESULT AND DISCUSSION

Results being used from the numerical simulation are presented in a roll spectral of LCT ship motion in beam sea. The LCT Ship in a beam sea condition is shown at Figure 7.

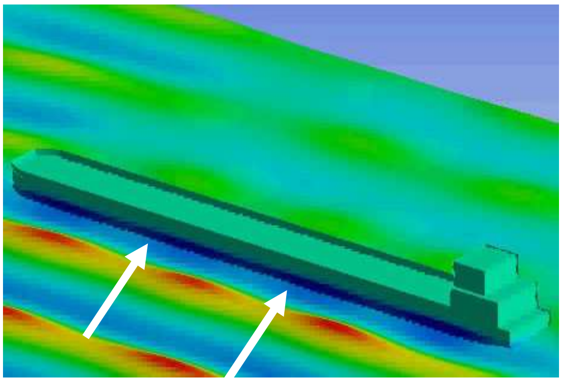

Figure 7: LCT Ship in a beam sea condition.

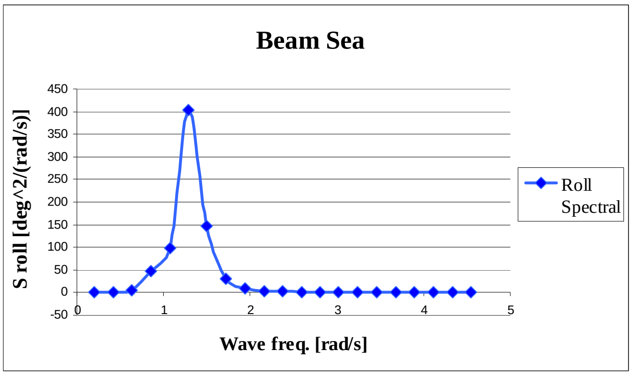

Figure 8: Roll spectra of LCT ship in beam sea

The result of roll spectra is obtained and presented on Figure 8. This spectra is on frequency domain therefore needs to change in time domain. The Inverse Fast Fourier Transformation (IFFT) in equation (5) then was applied to the roll spectra to obtain the time history of the roll motion that is shown on Figure 9. To obtained the angular acceleration which is needed to calculate the acting moment on the rudder stock the value of the roll time history had to be integrated twice.

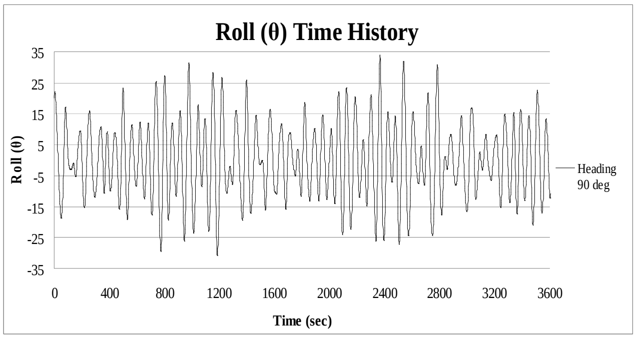

Figure 9: Roll time history in beam sea

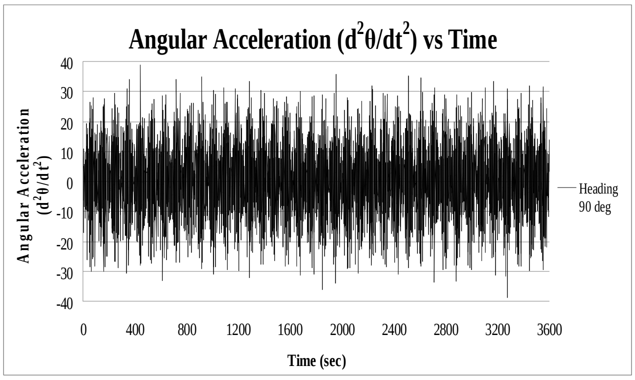

Figure 10: Angular acceleration time history in beam sea

Table 2: Rudder stock (shaft) properties

Item	Value	Units
$I_{xx\ mass}$	74.79	kg.m
I_{Area} (I_z)	0.0003976	m^4
ŷ	0.045	M

To obtain stress time history, equation (6) was applied in which the values on the Table 2 were used. The time history of the stress that working on the rudder stock are presented on the Figure 11 below:

t

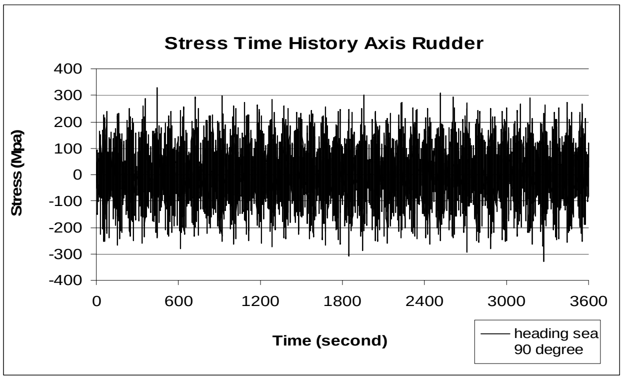

Figure 11: Stress rudder shaft time history in beam sea.

JSOse | Received: 4-February-2017 | Accepted: 30-March-2017 | [(9) 1: 1-6]
Published by International Society of Ocean, Mechanical and Aerospace Scientists and Engineers, www.isomase.org., ISSN: 2442-6415

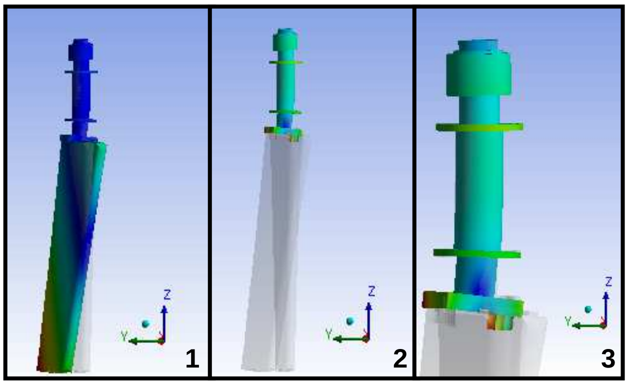

Figure 12: Simulation of LCT udder stock deformation due to hydrodynamics force.

A simulation that is shown in Figure 12 was made to explain the phenomena deflection of the rudder stock due to the bending stress. The fatigue life prediction can be calculated by knowing the fatigue characteristic of the rudder stock material (ST41). This characteristic can be found by conducting a fatigue test on rudder stock in National Laboratory for Structural Strength Technology – Agensy for Assessment and Application of Technology. The speciemnt of fatigue test and the result of S-N Curve of LCT Rudder Stock are presented on Figure 13 and 14 respectively.

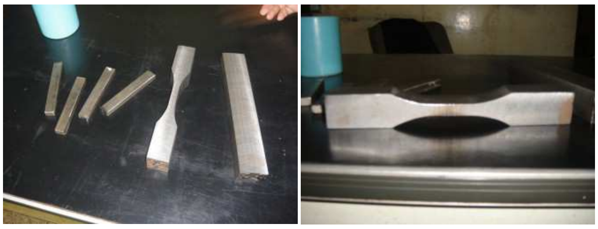

Figure 13: Fatigue test specimens of rudder stock material.

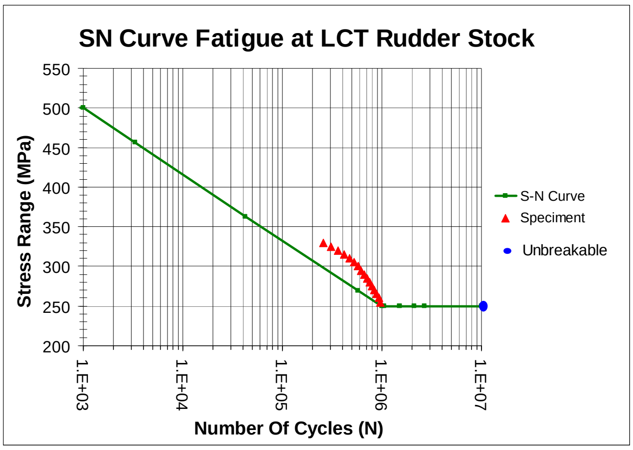

Figure 14: S-N Curve fatigue at LCT rudder stock.

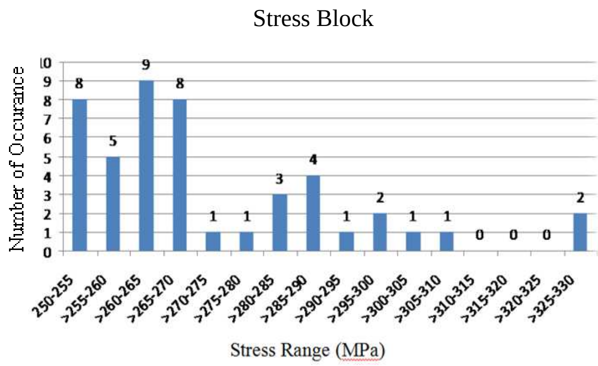

Figure 15: Stress blocks at The Rudder Stock

An endurance limit (treshold value) 250 Mpa of the rudder stock material can be found from the S-N curve. From this value a number of occurance of the higher stress than endurance limit can be constructed which is shown on Figure 15. To applied the Palgram-Miner rule the stresses are grouped in to 16 stress blocks. From these blocks of stress amplitude the life of the rudder stock can be calculated. Table 3 shows the fraction of life for the rudder stock material in every stress block.

Table 3: Rudder stock fatigue life in beam sea

No	Stress Range Mpa	Stress CF Mpa	Number of Occurences (n_i)	Number of cycles to fatigue (N_{fi})	Fraction of Life n_i / N_{fi}
1	250 - 255	255	8	9.67E+05	8.27E-06
2	> 255 - 260	260	5	9.23E+05	5.42E-06
3	> 260 - 265	265	9	8.80E+05	1.02E-05
4	> 265 - 270	270	8	8.37E+05	9.56E-06
5	> 270 - 275	275	1	7.94E+05	1.259E-06
6	> 275 - 280	280	1	7.50E+05	1.333E-06
7	> 280 - 285	285	3	7.07E+05	4.243E-06
8	> 285 - 290	290	4	6.64E+05	6.024E-06
9	> 290 - 295	295	1	6.20E+05	1.613E-06
10	> 295 - 300	300	2	5.77E+05	3.466E-06
11	> 300 - 305	305	1	5.24E+05	1.908E-06
12	> 305 - 310	310	1	4.70E+05	2.128E-06
13	> 310 - 315	315	0	4.17E+05	0
14	> 315 - 320	320	0	3.63E+05	0
15	> 320 - 325	325	0	3.10E+05	0
16	> 325 - 330	330	2	2.56E+05	7.813E-06
				Total	6.33E-05

Σn/N	6.33E-05	Hz
	15806.99	during 3600 sec
	15806.99	in hour
	1.8044505	in years
Rudder Stock Life Time	1.8	years

As presented on Figure 11 duration of the load is repeated in every 360 seconds. The total fraction of life is 6.33×10^{-5} Hz. To make this fraction of life to be unity the 3600-second cyclic load time must be multiplied by $1/6.33 \times 10^{-5}$. This corresponds to $6.33 \times 10^{-5} \times 3600$ seconds = 15806.99 hours, or about 1.8 years.

4.0 CONCLUSION

This study has presented a method for predicting fatigue life of the rudder stock due to the ship roll motion at random sea wave

JSOse | Received: 4-February-2017 | Accepted: 30-March-2017 | [(9) 1: 1-6]
Published by International Society of Ocean, Mechanical and Aerospace Scientists and Engineers, www.isomase.org., ISSN: 2442-6415

that applied on existing design of LCT. The result from the calculation in beam sea condition of sea state 5 in JONSWAP spectrum shows that the prediction of fatigue life of the rudder stock is 1.8 years. This finding also recommends:

1. The owner to conduct an annual survey of the LCT ship in particular to the rudder stock.
2. The owner to modify the structural design of the rudder stock construction system to increase the fatigue life.

ACKNOWLEDGEMENTS

The authors highly appreciate a research grant from the Ministry of Research and Technology Republic of Indonesia through PKPP Program on Year 2012 to conduct this study.

REFERENCES

1. Bhattacharyya Rameswar, 1978, *Dynamics Of Marine Vehicles*, John Wiley & Sons
2. Biro Klasifikasi Indonesia (BKI) Rule
3. Bentley Systems, Incorporated, 2013, *Maxsurf Motion User Manual Book*
4. F. Molland Anthony, Stephen R. Turnock, 2007, *Marine Rudders and Control Surfaces*. ELSAVIER.
5. Gujaran Sandeep and Shivaji Gholap, 2014, *Fatigue Analysis of Drive Shaft*. International Journal of Research in Aeronautical and Mechanical Engineering.
6. Http://Www.Originlab.Com/Doc/Origin-Help/Ifft, 10:18:00 (GMT+7), Oktober 31th, 2016
7. Mujahid Ahmad S, 2012, *Prediksi Umur Pakai Struktur Sistem Kemudi Akibat Pengaruh Beban Hidrodinamika pada Kapal Ferry LCT di Perairan Kalimantan*, Laporan Akhir PKPP-25 No. F1.138.
8. Nugroho Wibowo H, Samudro O, 2004, *On The Fatigue Analysis of Mooring Lines in Irregular Wave*, 4th Regional Conference of Marine Technology 2004 paper no III. A. 14 - 22, on 7 & 8 September 2004 at Johor, Malaysia
9. Nugroho Wibowo H., and Ahmad S Mujahid, 2015, *Head Sea Slamming Pressures Prediction on a Frigate Ship Hull (A Numerical Study)*. World Maritime Technology Conference, SNAME. Rhode Island, USA
10. Nugroho Wibowo H., Sahlan and Ahmad S Mujahid, 2011, *Prediksi Umur Pakai Struktur Kayu Lambung Motor Tempel di Sungai Musi karena Beban Impak dari Gerakan Porpoising*. Seminar Nasional Teori dan Aplikasi Teknologi Kelautan (SENTA), ITS - Surabaya.
11. Safety Investigation Authority, 2012, *M/S SILJA EUROPA (FIN), Breaking of the Starboard Rudder Shaft in the Aland Archipelago on 22 November 2009.* Multiprint Oy, Vantaa, Finland.
12. Sergeu Petinov, 2003, *Fatigue Analysis of Ship Structure*. Backbone Publishing Company.