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Development of Maintenance Scheduling Model for the Safety Operational of Ship Machinery

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Abstract

Risk management of ship machinery is an important issue since machinery out of order can run into danger, especially for ships at sea. This paper implements risk based maintenance (RBM) to minimize the frequency and consequences of ship machinery failure. Not only the common steps of RBM, such as identification of problem, risk assessment, risk evaluation, and maintenance planning are conducted, but this paper also proposes a new model called ship position estimation. The preliminary identification i.e. identification of failure causes and symptoms as well as the history of failure time will be looked at first. In the risk assessment, quantification of the consequences of failure (*Cof*) considers system performance loss, while the probability of failure (*Pof*) is obtained from the reliability analysis of the failure time history. Risk evaluation compares the result of the risk assessment with the risk acceptance criteria in order to determine the level of risk. The proposed model of ship position estimation recognizes the ship position on the voyage when the analyzed machinery is in a high level of risk. Maintenance planning is further carried out to keep the machinery under the risk acceptance level. This paper utilizes a method called system dynamics to create simulation for each step of the RBM. As a case study, the parts of the pumps in the main engine cooling system are analyzed. The result of this paper is a proposed maintenance interval which is reasonable enough compared with the standard for pump maintenance. Additionally, the ship position is included when the pump reaches a high level of risk.

1. Introduction

The maintenance strategy of ship machinery should comply with the regulations of the ship classification bureau. General inspection is carried out every five years, when the ship is at dock. Some machinery is disassembled to examine its condition. This means that the real condition of ship machinery only can be known every five years on the general inspection dates. Unexpected machinery trouble can occur between the docking surveys. A corrective maintenance scheme is usually carried out when a symptom of machinery trouble first appears. If a severe symptom happens when the ship is under operation, it can lead to a catastrophic incident. Moreover, a maintenance tasks are sometimes difficult to carry out during ship passage because of limited spare parts availability or the requirement of shore base support [1]. Based on this background, analysis on the risk of machinery breakdown is urgent for sustainability of ship operation.

This paper implements a method called risk based maintenance (RBM) to estimate the risk of machinery failure during its operation between two docking surveys of ship. By applying RBM, a catastrophic failure of machinery can be minimized because the risk is kept at an acceptable level by applying preventative maintenance. The demand for doing maintenance is prioritized based on the magnitude level of the risk. This study also proposes a new model development for RBM, a ship position estimation for times when the machinery runs under a high level of risk. Benefit of this proposal is that it increases maintenance planning based on additional information of risk and can be used to guide an engineer to prepare for times of high level of risks. This research outcome should help management remain in budget since the optimum operation and maintenance can be reached without the reliability of ship machinery degrading.

Maintenance management has been through a long development process. In the beginning, corrective maintenance was conducted, after that periodic overhauls were introduced, and then planned preventive maintenance, condition monitoring, reliability centered maintenance, expert system which finally leads to the current research interest on the maintenance field, which considers risk as the main core study [2,3]. RBM focuses on the management of the risk of failure. Risk quantification is obtained by combining the results of *Cof* and *Pof* analysis. RBM was initially proposed as a structured comprehensive method comprised of a step of modules [2]. Since that time, RBM has been implemented in many fields of study. It was successfully employed to analyze the risk in ethylene oxide production facilities and brought down the original high risk of the equipment [4]. In another study, a proposed RBM development was applied in a power generating plant [5]. The outcome showed that critical risky equipment could be identified and the reliability of the equipment could be increased. Additionally, it reduced the cost of maintenance including cost of failure. In an oil refinery, a development of RBM has also been satisfactorily implemented [6].

The literature of RBM mainly discusses problems in the field of industrial applications and transportation systems [3]. In the industrial field, this method specifically appears in mechanical, chemical and electrical fields such in [2, 4, 7, 8, 9]. Its application on transportation system can be found in some research [9, 10]. In the marine field, there is little research considering risk analysis in the maintenance strategy for ship machinery. Some previous studies show a maintenance strategy which minimizes the total operation cost. The optimization process is carried out by adjusting the appropriate maintenance interval in order to obtain the minimum total cost of machinery operation [1,11,12,13]. There is a necessity to consider risk analysis in the maintenance strategy of ship machinery because not only total operation cost needs to be minimized, but the cost-incurring of loss caused by failure, as well. In this paper, the RBM method is adapted for use in the maritime field, especially for risk management of ship machinery operation.

This study focuses on a case study of the pumps in the cooling system of the ship′s main engine. Pumps are needed to support the main engine work. Pump failure could induce interruption on the cooling system as well as the main engine of a ship. This paper utilizes system dynamics (SD) simulation to construct a model of RBM on the pump operation. SD is a powerful tool developed for simulating a complex system [14]. Recently, it has being used in maintenance management appearing in [12, 13, 15, 16]. Novelty of this paper is the utilization of SD in constructing part of RBM and estimating the ship position estimation when high risk appears. In this paper, SD models the proposed RBM technique comprised of five steps: 1. Preliminary identification, 2. Risk assessment, 3. Risk evaluation, 4. Ship position estimation, and 5. Maintenance planning. The details of the steps of RBM will be discussed in the next chapter. The outcome of this work is a maintenance planning which reduces the risk of failures of cooling pump in a ship′s main engine, and identification of the ship position when the pump runs into high risk.

2. Risk Based Maintenance (RBM) : implemented in the operation of ship machinery

As previously mentioned, RBM is comprised of a number of steps. This chapter will discuss each step of the process in the application of ship machinery operation. The steps of RBM in this chapter are described as follows.

2.1. Preliminary identification

The focus system is analyzed in detail. The working principle and the potential failure mechanism of subsystems, machinery and parts of machinery are recognized based on the historical failure data and the result of literature study. The smallest parts which comprise the machinery are studied. In preliminary identification, the information related to the machinery′s symptoms and causes of failure are identified. These machinery′s symptoms and causes of failure are used for further analysis of the step of RBM.

2.2. Risk assessment

2.2.1 Consequence of failure (*Cof*) analysis

The outcome of a failure can be defined as system performance loss (A_i), financial loss (B_i), human safety loss (C_i) and environment loss (D_i). This paper adopted an equation from [2] to determine the *Cof*. The form of the equation is presented as Equation 1.

Table 1. *PDF* and Reliability function

Distribution	*PDF*		*R(t)*	
Weibull 2 par.	$f(t) = \frac{\beta}{\eta}\left(\frac{t}{\eta}\right)^{\beta-1} e^{-\left(\frac{t}{\eta}\right)^{\beta}}$	(1)	$R(t) = e^{-\left(\frac{t}{\eta}\right)^{\beta}}$	(2)
Gumbel max	$f(t) = \frac{1}{\sigma} e^{(-z-e^{(-z)})}$	(3)	$R(t) = 1 - e^{(-e^{(-z)})}$	(4)
Gumbel min	$f(t) = \frac{1}{\sigma} e^{(z-e^{z})}$	(5)	$R(t) = e^{(-e^{z})}$	(6)

* β = shape parameter, η = scale parameter (weibull 2 parameters)
σ = scale parameter, μ = location parameter (gumbel max and gumbel min)
* $z = \frac{t-\mu}{\sigma}$

The consequence of the failure symptom recognized in the step of preliminary identification is quantitatively calculated by using Eq. 7. The details on the usage of this equation appear in the case study in the next chapter.

$$Cof = \sqrt{(0.25A^2 + 0.25B^2 + 0.25C^2 + 0.25D^2)} \quad (7)$$

2.2.2 Probability of failure (*Pof*) analysis

The probability of a basic event failure of machinery found in the preliminary identification, is quantified. The record of machinery failure is utilized in order to know the probability of this failure occurring. This paper uses statistical analysis to find the failure distribution which best represents the characteristics of the time to failure data of the machinery. There are three distributions which appear in this paper, i.e. Weibull two parameters, Gumbel max and Gumbel min. The probability density function (*PDF*) and reliability function of these three distributions are summarized in Table 1. In the final risk assessment, risk estimation is determined by combining the results of *Cof* and *Pof* analysis. Risk level of each piece of machinery is found by multiplying the results of *Cof* and *Pof* analysis.

2.3. Risk evaluation

Risk evaluation is a step of Risk Assessment for evaluation the result of frequency and consequency analysis. The estimated risk which results from the previous step is compared with risk acceptance criteria. The machinery which exceeds the acceptance criteria is subject to maintenance to keep it at an acceptable risk level.

2.4. Ship position estimation

In this step, this paper includes the position of the ship during her voyage when the estimated risk of the machinery is in the unacceptable risk level. The recognized estimation of location of the ship is important for further planning such as spare part allocation and maintenance planning.

2.5. Maintenance planning

The recognized position of ship is important if we are to construct an appropriate maintenance plan for the ship machinery. This is related to when and where the maintenance should be best done. The planned maintenance will reduce the risk of machinery failure in order to bring the risk down to an acceptable risk level. The following equation is utilized to determine the maintenance planning in this study.

$$m_p = I_m - t_r \tag{8}$$

Where m_p is the maintenance planning which interprets the remaining operation time for maintenance. I_m is the interval between maintenance which complies with the risk acceptance criteria. t_r is the current operation time which indicates how long the machinery has been in operation. If t_r equals zero, $m_p = I_m$. This means that the machinery has never been operated since it was installed or since the last maintenance. Determination of I_m and t_r are derived from Equations 4, 6 and 8 for Weibull 2 parameters, Gumbel max and Gumbel min respectively. They are defined as the following equations based on their type of failure distribution.

Table 2. Properties of the analyzed pumps of the cooling system of ship′ s main engine

Pump Name	Number installed	Capacity x head (m^3/h x m)	Rpm	Power (kW)
SW pump	3	285 x 15		18,5
CCFW pump	4	190 x 25	1800	22
JW pump	2	65 x 30		11

Equation (9) to Equation (14) shows the I_m and t_r for all the type of distribution, i.e. Weibull 2 parameters, Gumbel max and Gumbel min. The notation used for the distributions are as follow, β = shape parameter, η = scale parameter for weibull 2 parameters, while σ = scale parameter, μ = location parameter for gumbel max and gumbel min.

Weibull 2 parameters

$$I_m = \eta.\left(-ln\left(R_{I_m}(t)\right)\right)^{\frac{1}{\beta}} \tag{9}$$

$$t_r = \eta.\left(-ln\left(R_{t_r}(t)\right)\right)^{\frac{1}{\beta}} \tag{10}$$

Gumbel max

$$I_m = \mu - \sigma.ln\left(-ln\left(1 - R_{I_m}(t)\right)\right) \tag{11}$$

$$t_r = \mu - \sigma.ln\left(-ln\left(1 - R_{t_r}(t)\right)\right) \tag{12}$$

Gumbel min

$$I_m = \mu + \sigma.\ln\left(-ln\left(R_{I_m}(t)\right)\right) \tag{13}$$

$$t_r = \mu + \sigma.\ln\left(-ln\left(R_{t_r}(t)\right)\right) \tag{14}$$

3. Case study: development of RBM for the cooling system of the ship's main engine

The case study focusses on the pumps which are installed in the cooling system of a ship's main engine. This system has an important role in keeping the main engine at a working temperature. A breakdown in any part of the cooling system could disturb the main engine. One of the most important parts of the cooling system are the pumps, because they transfers the coolant fluid into the cooling system. This chapter will discuss the application of the proposed development of RBM method in the operation of the cooling pumps of a ship's main engine. Figure 2 illustrates the whole simulation model of RBM using SD.

			Failure symptoms															
			Insufficient discharge pressure	Intermittent operation	Insufficient capacity delivered	No liquid delivery	Overheat/ High bearing temperature	Short bearing life	Short mechanical seal life	High vibration	High noise level	Excessivepower demand	Elevated motor temperature	Elevated liquid temperature	Mechanical seal damage/ leaks excessively	Pump loses prime after starting	Internal clearances wear too rapidly	Coupling fails
Part name	**Failure causes**		S1	S2	S3	S4	S5	S6	S7	S8	S9	S10	S11	S12	S13	S14	S15	S16
Mechanical seal	Entrained air by seal leaks	C1	•	•	•					•	•		•			•		
	Improper mechanical seal	C2							•						•			
O-ring	Excessive compression/ pressure/ temperature	C3	•		•										•			
	Rough sealing surfaces	C4	•		•										•			
Shaft	Bent shaft	C5					•	•	•	•	•	•			•		•	•
	Parts loose on the shaft	C6								•	•							
	Shaft running off center because of worn bearing	C7													•			
	Excessive wear at internal running clearances	C8	•		•					•		•						
Discharge valve	Leakage valves	C9	•		•													
	Discharge valve failed to open/ partially open	C10	•		•	•				•			•	•				

Figure 1. Failure causes and symptoms of cooling pump of main engine [17, 18]

In Figure 1, SD model of RBM is constructed of pieces of sub models i.e. 1. Preliminary identification, 2. Risk assessment, 3. Risk evaluation, 4. Ship position estimation, and 5. Maintenance planning. The following description will discuss in detail about each step of the SD model of RBM. SD model is a method that can be used to interpreting the complex system. In this paper, some part of RBM will be simulated.

3.1. Preliminary identification

There are three types of pumps analyzed which have typical properties as shown in the Table 2. The total number of pumps is nine units comprised of sea water (SW) cooling pumps (4 units); central cooling fresh water (CCFW) pumps (3 units); jacket water (JW) pumps (2 units). The pumps' failure modes are identified. The common failure causes and symptoms of the pumps are studied from the pump operation history and reference studies. The overview of some failure causes and symptoms in the operation of cooling pumps are shown in Figure 1. This figure shows the relation of the common causes (C1 ~ C10) and the possible resulting symptoms (S1~S16). Out of all the pump parts, the mechanical seal, the O-ring, the shaft and the discharge valves are the parts which experience the most trouble based on the records of the ship operation history. Considering the results of the data, this paper focusses on these common pump part failures.

3.2. Risk assessment

3.2.1 *Cof* analysis

The possible symptoms of failure found in the preliminary analysis are taken into account in order to quantitatively measure the consequence of failure. Actually Cof analysis can be performed in terms of some types of loss as shown in the Equation 1. The symptoms of failure recognized in the previous step indicate that the consequences of the failure of the cooling pump can be measured by considering an assessment of the system performance loss conducted in this study. This study does not perform analysis on human safety, environmental effects or financial consequences. Performance loss indicated by the symptoms of failure in Figure 1 is classified into their level by utilizing performance function which is provided in the Table 3. After finding the *Ai* for each symptom, the result of Cof analysis is obtained by inserting the value of *Ai* into Equation 1.

A part of the SD model of RBM in Figure 2 performs *Cof* analysis. The highest value of *Ai* is inserted into the number 1 unit of the SD model. The highest value of *Ai* is used because it has the highest possibility to induce more serious consequences greater than the result of *Ai* from other causes of failure. In this model, the Equation 1 is used at number 2 unit of the SD model. The results of Cof analysis are then shown at the number 2 unit of the SD model. Table 5 summarizes the results of the *Cof* analysis for all of the parts of cooling pump in focus. It clearly shows that entrained air by seal leaks (C1), excessive compression/ pressure/ temperature and rough sealing surface (C3 and C4), bent shaft (C5) and discharge valve failed to open (C10) result in the most catastrophic consequences, i.e. pump loses prime after starting (S14), mechanical seal damage/ leaks excessively (S13), coupling fails (S16), no liquid delivery (S4) respectively.

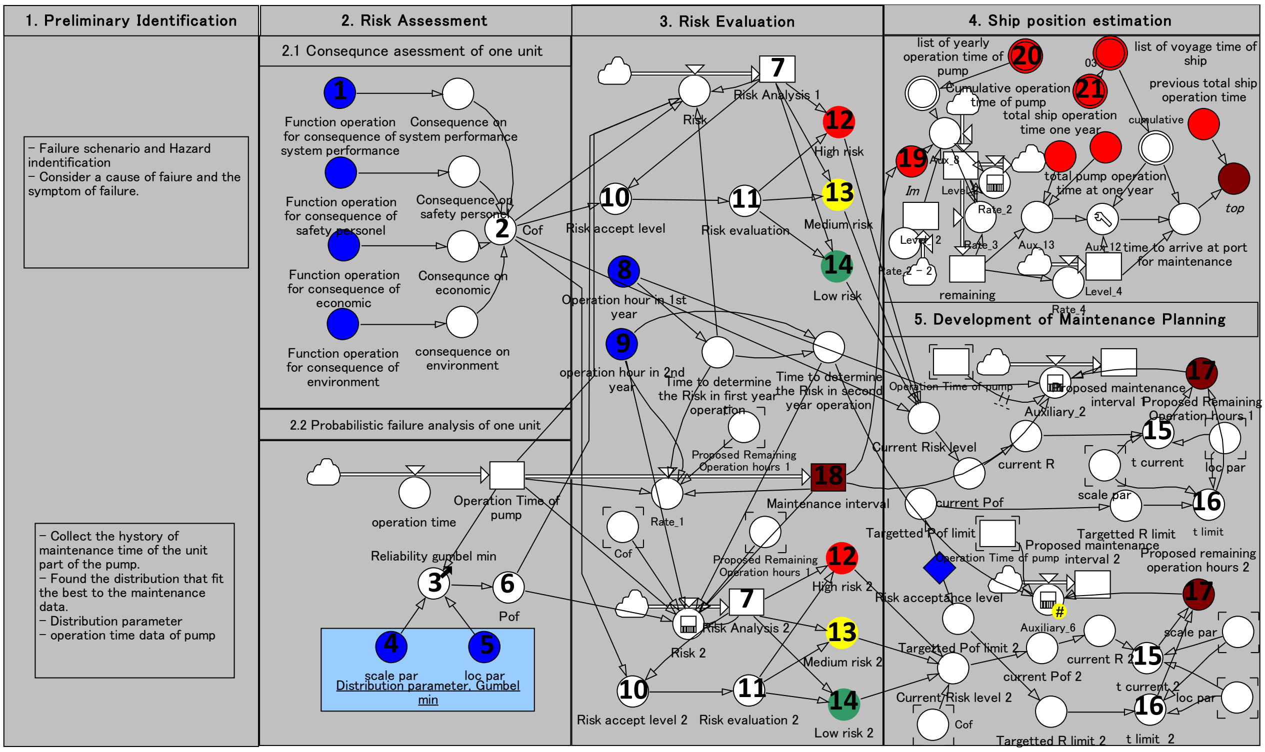

Figure 2. SD Model of RBM

Table 3. Performance functions. Modified after [2]

Level	Description	Func. (A_i)
I	Very important for operation of cooling pump ~Failure would cause the pump to stop functioning	8-10
II	Important for good pump operation ~Failure would cause impaired performance and adverse consequences	6-8
III	Required for good pump operation ~Failure may affect the pump performance and may lead to subsequent failure	4-6
IV	Optional for good performance ~Failure may not affect the performance immediately but prolonged failure may cause pump to fail	2-4
V	Optional for operation of cooling pump ~ no effect to the performance of cooling pump if failure happened	0-2

3.2.2 *Pof* analysis

This study analyses the operation history of the cooling pumps of a ship's main engine under 16 years of operation from 1997 until 2012. Failure time history has been recorded and analyzed. Table 4 depicts the failure distribution for all of the analyzed parts of the cooling pumps. The failure distributions listed in Table 4 is the distribution that best fits into the data of failure time. The quantitative *Pof* analysis utilizes these failure distributions by inserting the related equation and distribution parameters into the SD model of RBM. The SD model of *Pof* analysis appears in Figure 2 . In this model, reliability function in Table 1, is inserted into the number 3 unit of the model, while the distribution parameters are inserted into numbers 4 and 5. The result of *Pof* analysis comes up in the number 6 unit of model. The results of *Pof* analysis for all of the analyzed parts are completely presented in Table 5.

Table 4. Failure distribution of the analyzed parts of the cooling pumps

No	Pump name	Part Name	Distribution Name	Distribution Parameter			
1	SWP 1	Mechanical seal	Gumbel max	σ	2727.7145	μ	6090.5733
		O ring	Gumbel max	σ	3591.3595	μ	13099.3139
		Shaft	Gumbel max	σ	916.9122	μ	11555.8849
		Discharge valve	Gumbel min	σ	1826.0322	μ	34357.5373
2	SWP 2	Mechanical seal	Gumbel max	σ	3167.5149	μ	8720.3298
		O ring	Gumbel min	σ	1655.4744	μ	21848.7532
		Shaft	Gumbel max	σ	583.4896	μ	13353.7449
		Discharge valve	Gumbel min	σ	1016.2718	μ	37105.1991
3	SWP 3	Mechanical seal	Weibull 2 Par.	β	5.9175	η	14893.2709
		O ring	Weibull 2 Par.	β	6.2210	η	25786.8388
		Shaft	Weibull 2 Par.	β	7.9968	η	27817.3633
		Discharge valve	Gumbel max	σ	2252.0440	μ	31945.4698
4	CCFW 1	mechanical seal	Gumbel min	σ	2917.4479	μ	18831.2752
		O ring	Gumbel min	σ	835.0361	μ	19902.5203
5	CCFW 2	mechanical seal	Gumbel min	σ	1526.7017	μ	11268.6248
		O ring	Gumbel min	σ	742.2342	μ	18790.0776
		mechanical seal	Gumbel max	σ	9432.8196	μ	20488.8841

6	CCFW 3	O ring	Gumbel min	σ	4563.1935	μ	32716.6392
7	CCFW 4	mechanical seal	Gumbel min	σ	877.9233	μ	11886.6141
		O ring	Gumbel max	σ	4040.7997	μ	16061.7769
8	JWP 1	mechanical seal	Gumbel min	σ	250.0669	μ	5848.3950
		O ring	Gumbel max	σ	583.4896	μ	4353.7450
9	JWP 2	mechanical seal	Gumbel min	σ	683.8604	μ	7735.6860
		O ring	Gumbel max	σ	625.1674	μ	4879.0125

As pump operation time goes on, the failure probability of the parts of the pump increases, in the same time followed by the degradation of reliability[12]. The RBM technique enables us to know the risk of pump failure by considering increases in the probability of failure. Risk estimation of the pump failure is determined by multiplying the result of the *Cof* and *Pof* analysis. The number 7 unit of the SD model in Figure 2 calculates the risk estimation of cooling pump failure. In this paper, the result of risk estimation is shown in two different periods of t_r. Table 5 lists the results of the risk estimation for the first year of operation and the second year period of operation. In the first year, the t_r of SW pumps, CCFW pumps and JW pumps are 1336, 1177 and 430 hours and in the second year operation are 4569, 3852 and 1660 hours respectively. This data was taken from the real operation history of the analyzed pumps. In Figure 2, the data is inserted into numbers 8 and 9 units of the SD model for first year and second year operation respectively.

3.3. Risk evaluation

SD simulation of RBM calculates the risk estimation of the operation of the cooling pump of the ship's main engine. After risk estimation has been conducted, risk evaluation is presented to classify the risk of failure into the low, medium and high risk. Risk evaluation determines the need of the cooling pumps to be maintained in order to bring down high risk to an acceptable level. In this step, risk acceptance criteria need to be set to give the minimum risk level of cooling pumps during operation. This study uses the Pof_{limit} which is obtained from the conversion of the risk acceptance limit. Because the level of *Cof* in Table 5 is 4 and 5, the result of the conversion value for the Pof_{limit} is 1.0E-02 [19]. The risk is classified in unit model number 11 after the value of Pof_{limit} has been set in unit number 10 of the SD model. The result of risk classification appears in units 12, 13 and 14 in Figure 2. In the constructed SD model, the red, yellow and green colors of the units respectively represent high, medium and low levels of risk.

Table 5. Result of simulation in the first and second year pump operation

Pump	Part name	Causes	Symptoms	*Cof*	After 1st year operation			After 2nd year operation		
					Pof	Risk	m_p (hr)	*Pof*	Risk	m_p(hr)
SWP 1	*Mechanical seal*	C1	S14	4,5	2.20E-07	9.88E-07	2920	7.16E-11	3.22E-10	3940
	O-ring	C3, C4	S13	4	3.24E-12	1.30E-11	6280	2.14E-05	8.55E-05	3050
	Shaft	C5	S16	5	≈ 0	≈ 0	8820	≈ 0	≈ 0	5590
	Discharge valve	C10	S4	5	1.40E-08	7.00E-08	24620	8.23E-08	4.11E-07	21390
SWP 2	*Mechanical seal*	C1	S14	4,5	3.39E-05	1.53E-04	2550	3.27E-06	1.47E-05	3200
	O-ring	C3, C4	S13	4	4.16E-06	1.66E-05	12900	2.93E-05	1.17E-04	9660
	Shaft	C5	S16	5	≈ 0	≈ 0	11130	≈ 0	≈ 0	7890
	Discharge valve	C10	S4	5	≈ 0	≈ 0	31100	1.24E-14	6.22E-14	27860
SWP 3	Mechanical seal	C1	S14	4,5	6.36E-07	2.86E-06	5510	9.19E-04	4.13E-03	2280
	O-ring	C3, C4	S13	4	1.01E-08	4.02E-08	10970	2.11E-05	8.44E-05	7740
	Shaft	C5	S16	5	2.86E-11	1.43E-10	14310	5.33E-07	2.66E-06	11080
	Discharge valve	C10	S4	5	≈ 0	≈ 0	27160	≈ 0	≈ 0	23930
CCFW 1	Mechanical seal	C1	S14	4,5	2.35E-03	1.06E-02	4230	5.87E-03	2.64E-02	1560
	O-ring	C3, C4	S13	4	1.82E-10	7.30E-10	14880	4.49E-09	1.80E-08	12210
CCFW 2	Mechanical seal	C1	S14	4,5	1.35E-03	6.06E-03	3070	7.74E-03	3.48E-02	390
	O-ring	C3, C4	S13	4	4.95E-11	1.98E-10	14200	1.82E-09	7.27E-09	11520

CCFW 3	Mechanical seal	C1	S14	4,5	4.32E-04	1.94E-03	4900	2.93E-03	1.32E-02	2230
	O-ring	C3, C4	S13	4	9.96E-04	3.98E-03	10550	1.79E-03	7.15E-03	7870
CCFW 4	Mechanical seal	C1	S14	4,5	5.04E-06	2.27E-05	6670	1.06E-04	4.77E-04	4000
	O-ring	C3, C4	S13	4	≈ 0	≈ 0	8710	1.22E-09	4.88E-09	6040
JWP 1	Mechanical seal	C1	S14	4,5	5.32E-08	2.39E-07	3040	7.28E-06	3.28E-05	1810
	O-ring	C3, C4	S13	4	≈ 0	≈ 0	1800	4.61E-06	1.84E-05	570
JWP 2	Mechanical seal	C1	S14	4,5	1.39E-04	6.23E-04	2930	8.37E-04	3.76E-03	1700
	O-ring	C3, C4	S13	4	≈ 0	≈ 0	2260	3.47E-11	1.39E-10	1030

The results of the SD simulation listed in the Table 5 show that there is no maintenance needed for any of the analyzed pump parts in the first year of operation, since the value of *Pof* is under the Pof_{limit}. During the second year of operation, there is maintenance/replacement for mechanical seal of SWC pump 1 and 2 (italicized writing). The *Pof* value of these parts reaches the Pof_{limit} when they enter the second year operation time. Maintenance is indicated by the changing value of m_p, which becomes longer by the end of the second year of operation, i.e. 2920 hours into 3940 hours and 2550 hours into 3200 hours respectively for mechanical seal of SW pump 1 and 2.

3.4 Ship position estimation

Previously, risk estimation has been quantified followed by risk evaluation which determines the level of risk. In this step, the position of the ship is taken into account when a high level of risk occurs in any of the cooling pumps. SD model of ship position estimation is proposed to allow this step to work. The construction of the model is based on real data of the ship voyage history over the past 16 years. The SD model of ship position estimation is shown in Figure 2. Some types of data such as I_m, yearly pump operation and yearly ship voyage time are inserted into this SD model, units 19, 20 and 21 respectively. The outcome of this proposed model is the total ship voyage time after arrival at port for pump maintenance (t_{op}) which is calculated in the number 20 unit of the SD model. t_{op} is the time spent during voyages until the ship reaches a port where the value of *Pof* of the pump exceeds the Pof_{limit}. The detailed results of the proposed model are shown in Table 6 in the column of ship position estimation. It shows clearly, when the ship should be maintained, at what over ground distance (OG dist.), and where the port/ anchorage of maintenance should be. In the column of port/ anchorage, the italicized type means that the ship is moored in the port while the normal type means that the ship is anchored.

Table 6. Ship position estimation and comparison of I_m and I_m standard

Pump	Part name	Ship position estimation			Comparison of I_m (hr)	
		t_{op} (hr)	OG. dist. (miles)	Port/ anchorage	I_m	I_m standard
SWP 1	Mechanical seal	2805	47769	*Nagasaki*	4260	5000
	O-ring	5259	90166	Ishigaki offing	7620	15000
	Shaft	6923	118644	*Kushiro*	10160	12000
	Discharge valve	17549	301989	Great bitter lake	25960	-
SWP 2	Mechanical seal	2555	43739	Tsu offing	3880	5000
	O-ring	9688	166338	*Osaka*	14230	15000
	Shaft	8513	145932	*London*	12460	12000
	Discharge valve	21012	354462	*Takamatsu*	32430	-
SWP 3	Mechanical seal	4684	80818	*Muroran*	6850	5000
	O-ring	8410	143971	Panama canal	12310	15000
	Shaft	10854	186472	*Recife*	15650	12000
	Discharge valve	19165	326146	*Brisbane*	28500	-
CCFW 1	Mechanical seal	4440	76369	Suez canal	5410	5000
	O-ring	13360	230357	*Curacao*	16060	12000
	Mechanical seal	3546	60260	*Tokyo*	4250	5000

CCFW 2	O-ring	12732	218816	*Tokyo*	15380	12000
	Mechanical seal	4968	85418	Kagoshima offing	6080	5000
CCFW 3	O-ring	9582	164629	*Nagasaki*	11730	12000
	Mechanical seal	6655	114413	El ballah by pass west	7850	5000
CCFW 4	O-ring	8145	139189	*Tokyo*	9890	12000
	Mechanical seal	8601	147550	*Barcelona*	4700	5000
JWP 1	O-ring	6373	109326	*Naples*	3460	12000
	Mechanical seal	8410	143971	Panama canal	4590	5000
JWP 2	O-ring	7158	122389	*Tokyo*	3920	12000

3.5 Maintenance planning

Maintenance planning is carried out after risk evaluation and ship position estimation. In this step, the cooling pumps have been prioritized for maintenance based on the level of risk, such as shown on some previous study [20, 21, 22]. As shown in Table 5, m_p for each pump is clearly defined. m_p is important, especially for the ship engineer, in order to make a priority list of time remaining until maintenance of the cooling pumps of the ship's main engine is necessary. In this paper, m_p is calculated by Equation 8 which is determined from I_m and t_r. Equation 8 is inserted into the number 17 unit of the SD model, while I_m and t_r are calculated by using Equation 9~14 and inserted into the units 15 and 16 of the SD model respectively.

In this paper, the maintenance planning also provides the I_m for all of the studied cooling pumps as presented in the Table 6. In order to compare the results of I_m in this study, the standard I_m published by the pump manufacturer is used. Table 6 provides the list of the I_m standard for all of the parts of the analyzed pumps except for the discharge valve because the pump company does not publish it. In pump operation, I_m standard is not always exactly applied because it is an approximation value. In reality, I_m can vary based on the operation condition of the pump, such as type of fluids, temperature and pump operation mode.

Based on the comparison of the I_m results with the I_m standard, a significant difference can be seen for the O-ring of JWP 1 and 2. Some possible reasons of this discrepancy are described as follows: 1). High fluid temperature, since JW pump is operated in the high temperature loop of the cooling system of main engine, 2). JW pump working pressure is the highest of all cooling pumps (see Table 2), and 3). There are only two JW pumps installed, fewer than the other cooling pumps. This condition may cause the JW pumps to work harder.

Overall comparison, it can be seen in Table 6 that most of the I_m resulting from the SD model has quite a similar value to the standard from the pump manufacturer. It can be concluded from this, that the SD model of RBM in this paper presents a reasonable outcome. SD model presented in this study results in not only I_m but also shows the m_p and ship position estimation which gives us the t_{op}, OG. dist., and port of mooring/ anchorage for maintenance. This outcome is very beneficial for the ship engineer in that it allows for a better maintenance strategy for the cooling system of a main engine.

4. Conclusion

This paper shows an SD simulation which is utilized to construct a model of RBM with a case study that focusses on the parts of the SW pumps, CCFW pumps and JW pumps. SD model of RBM as shown in Figure 2, is built up by adding together SD model of 1). Preliminary identification, 2). Risk assessment, 3). Risk evaluation, 4). Ship position estimation, and 5). Maintenance planning. The outcomes achieved by this SD model of RBM are *Pof*, *Cof*, 1st year and 2nd year estimation of risk, maintenance planning (m_p) and interval time between maintenance (I_m), while the ship position estimation of the proposed model development of RBM, gives a clear interpretation on the position, passage time and covered distance of the ship when the machinery runs into a high level of risk. These results should improve the existing maintenance strategy for the management of the ship company. Given the results of the ship position estimation and maintenance planning, they enable the ship engineer to better construct a maintenance strategy for the cooling system of the ship's main engine. The maintenance strategy which further can be planned is aimed to give a safe of ship operation.

Focusing on the analyzed parts in this case study, it is obvious that the I_m of similar pump parts in different pumps have quite different values. Cooling pump operation conditions causes this disparity. Although differences appear, the I_m results are in line with the I_m standard obtained from the pump manufacturer. There are only two parts that show an odd value of I_m i.e. O-ring of JW pump 1 and 2, but they are tolerable since the operation conditions of JW pumps are severe compared to the other pumps. It is possible to make the I_m shorter. Study improvement may be possible by extending the history data of failure time and failure mode of the cooling pump. In this study, limited data meant that only a few failure modes could be analyzed. More failure time data is needed in order to collect more type of failure modes. These improvements may develop the current SD model of RBM to become more complex. Focused equipment is also possible to be added since there are some other important components which also have an important function in the cooling system of the ship's main engine. We can improve the SD model of RBM in marine machinery operation by taking these matters under consideration for future work, so the maintenance strategy of main engine support system can be improved so ensure the safety during its life time.

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