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Investigation of The Value of Spring Constant and Mass on The Efficiency of Moving Mass Devices

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Abstract*—Simple configuration and has good efficiency, make a moving mass stabilizer one of the options that can be applied to reduce roll motion on a small vessel. This stabilizer however has a limitation in dealing with the ship's hydrostatic changes while on duty especially when the stabilizer is designed to be passive. The purpose of this research is to improve the ability of the stabilizer to be able to adapt the change in ship hydrostatics. A Tsunami 22' fishermen vessel model was selected to be used for this research. By conducting roll decay experiments, natural frequency data from the vessel is then used to design calculations for the device on two different load conditions. Moving mass stabilizer frequency is dependent on two parts that are the spring coefficient "k" and the weight of the mass moving. In this Research, Spring adjustment is selected to make the stabilizer able to change frequency following change on the vessel. It is found that the best frequency ratio between the frequency of stabilizer and vessel is 1. Adjusting the spring of the stabilizer turned out to give an increase in device performance by 8.9 % when compared to adjusting the mass. The results obtained in this research indicate the moving mass stabilizer has good potential to reduce the roll motion.

***Keywords*—Frequency Ratio, Moving Mass, Spring Constant.**

I. INTRODUCTION

Methods to reduce the rolling motion of ships have been around for thousands of years. Currently, there are several commonly used methods such as bilge keels, fin stabilizers, tube tanks, gyroscopes, and moving mass devices. Each method has advantages and disadvantages depending on the type of ship to be used. Moving mass is one method that is suitable for use on small ships considering it is a relatively simple configuration and good efficiency [1].

In its most basic form, a moving mass system consists of a ballast coupled to two springs aligned laterally. Later, this mass will move to generate a moment acting on the vessel, to dissipate the rolling kinetic energy. This is in line with what Frahm described in 1909 when he proposed the concept of shifting internal masses [2]. Chadwick then conducted more tests and examined the data, concluding that by utilizing moving mass, the peak roll may be lowered to 50%, however, the performance produced is irregular [1]. Koike et al. continued the development of this approach after a long vacuum by placing a hybrid system of moving mass on a survey supervision vessel and achieving satisfactory results from the experiment [3]. In further development, Treakle et al. used a numerical time-domain approach using PID in the moving mass system [4]. In addition to conducting experiments, the development of the moving mass method is also carried out using simulations such as those carried out by Hirotaki Sasaki and Ryo Watanabe. They both adopted an anti-windup technique in 2000 to improve the control performance of a movable range of masses [5]. The most recent is Montazeri et al. 2010 who used the Lagrange method to extract a mathematical model of a moving mass and simulated it on the Gaul trawler that sank in 1974 [6].

With all its advantages, the moving mass system also has disadvantages, especially when paired in passive mode. As with other passive stabilizers, the effectiveness of the moving mass system is very vulnerable when the load changes as the ship operate at sea. Therefore, it is necessary to develop a moving mass system that can adapt to changes in the hydrostatic of the ship. This is due to beam seas conditions; small vessels are prone to degradation of seakeeping qualities. One of the problems is indeed the rolling motion experienced by these vessels when encountering waves.

In this paper, the effect of changes in the value of the spring constants and mass is investigated to see which one can provide the best efficiency when it is necessary to change the initial setting of the moving mass following changes that occur in the ship. The investigation was carried out using experiments using the Tsunami fishing vessel model made by UTM in 2005.

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II. METHODOLOGY

A. Selecting a Vessel

In this research, a 7.2 m fishing vessel model scale as shown in Fig. 1 was selected. This model scale has a scale factor of 2.88 compared with its actual scale. The general particulars of the vessel are shown in Table 1.

Fig. 1. Model Scale Fishing Vessel

TABLE I. GENERAL PARTICULARS OF MODEL VESSEL

Parameters	Value
Length Overall (LOA)	2.5 m
Length of Waterline (LWL)	2.25 m
Beam on Waterline	0.48 m
Draft	0.11 m Amidship
Long. Ctr of Buoyancy (LCB)	-0.21 m Fwd. Amidship
Long. Ctr of Gravity (LCG)	-0.21 m Fwd. Amidship
Displacement	74 kg
Vertical Ctr. Of Gravity (KG)	0.186 m

B. Device Design

The rolling natural frequency of the vessel can be known by conducting roll decay experiments. For each experiment, the model is given a 10-degree initiate angle, then released freely until it is in stable condition. In this research loading state of the vessel is divided into two conditions which represent vessel operation on the sea, full load (74 kg) and half load (64 kg). Fig. 3 Roll decay experiments for full load conditions without a device show the natural frequency of the vessel is 6.35 rad/s. After that, the device design calculation is made for two frequency ratios (1) for each parameter investigated as given in Table 2. Mass ratio μ (2) decided to be 1% in the recommendation range of 0.5-2% vessel displacement [8] except for the changed mass value case. This is because in that case, the use of a heavier mass is unavoidable to match the frequency ratio. Once the natural frequency of the device and mass value is known, spring constant k was obtained by using (3). The length of the device is made for 50 cm according to the beam of the hull at z_0 equals 0.038 cm.

$$FR = \frac{\omega m}{\omega \phi} \tag{1}$$

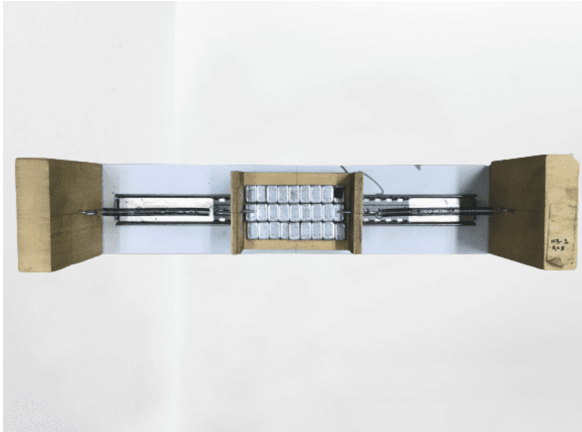

Fig. 2. Stabilizer device

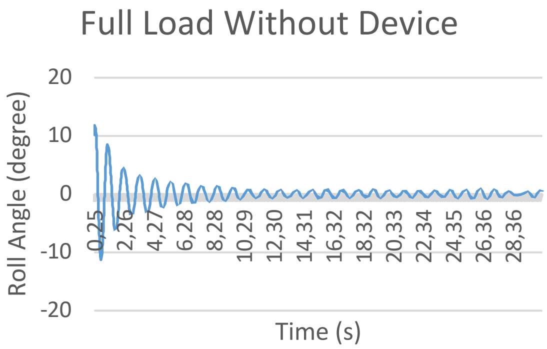

Fig. 3. Roll decay results for full load condition without device

$$\mu = \frac{mm}{\Delta} \tag{2}$$

$$\omega m = \sqrt{\frac{2k}{mm}} \tag{3}$$

$$Improvement = \frac{k\ with\ device - k\ without\ device}{k\ without\ device}\ x\ 100 \tag{4}$$

TABLE II. STABILIZER DEVICE DIMENSION FOR FULL LOAD CONDITION

Parameters	Fixed Spring Constant and Changed Mass Value		Changed Spring Constant and Fixed Mass Value	
Frequency Ratio	2	1	2	1
Stabilizer Frequency, ω_m (rad/s)	10.25	5.04	10.25	5.13
Moving Mass (kg)	0.73	3.02	0.73	0.73
Mass Ratio, μ	0.0096	0.051	0.0096	0.0096
Vertical Position of Mass from CG, Zo (m)	0.038	0.0096	0.038	0.038
Spring Constant, k (kg/s^2)	76.74	76.74	76.74	19.19

For half load conditions without the device, Fig. 4 shows that the natural frequency of the vessel is 5.72 rad/s. Change on the spring constant k and mass value is made for matching

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with two frequency ratios for each parameter investigated as given in Table 3. Due to a reduction in the displacement of the vessel, CG is moved upwards, and the impact to z_0 decreases to 0.015. In the end, an improvement from the device can be calculated using (4) [8].

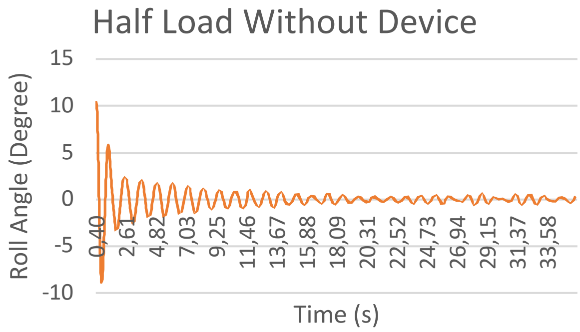

Fig. 4. Roll decay results for half load condition without device

TABLE III. STABILIZER DEVICE DIMENSION FOR HALF LOAD CONDITION

Parameters	**Fixed Spring Constant and Changed Mass Value**		**Changed Spring Constant and Fixed Mass Value**	
Frequency Ratio	2	1	2	1
Stabilizer Frequency, ω_m (rad/s)	9.86	4.94	9.86	4.93
Moving Mass (kg)	0.79	3.14	0.73	0.73
Mass Ratio, μ (%)	1.2	4.6	1.1	1.1
Vertical Position of Mass from CG, Zo (m)	0.015	0.001	0.015	0.015
Spring Constant, k (kg/s^2)	76.74	76.74	70.94	17.73

III. Result And Discussion

Figure 5 to 9 shows that the stabilizer device succeeds to reduce vessels' rolling amplitude and making the vessel become quicker to be back into it is stable condition. Furthermore, results from the roll decay experiment conducted point out that when the frequency ratio is equal to 1 is way better than the frequency ratio equal to 2 for every condition and case. This is because the heavier the masses the better result obtained [9]. The opposite result happened to spring cases, the softer the spring, got the better result. When the frequency ratio is equal to two, either the spring is too stiff, or the masses are too light then making the device, not really function. Only extra weight gained by the device contributed to the enhancement damping of the vessel.

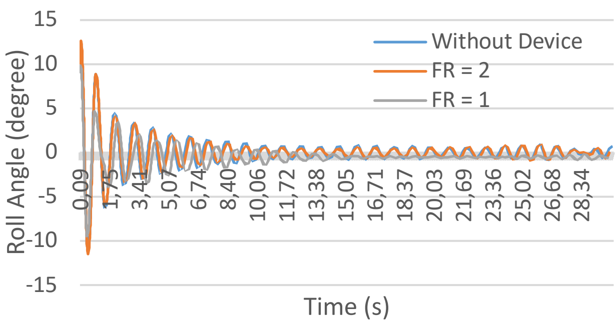

Fig. 5. Roll decay experiments for full load condition mass change

TABLE IV. ROLL DECAY ANALYSIS RESULT FOR FULL LOAD CONDITION MASS CHANGE

Parameters	**Without Device**	**FR = 2**	**FR = 1**
Critical Damping, b_c (tonnes m2/s)	0.04244	0.0416	0.041
Damping Coefficient, b (tonnes m^2/s)	0.001866	0.002037	0.002572
Virtual Mass Moment of Inertia, a (tonnes m^2)	0.00334	0.00343	0.00355
Natural Frequency Model, ω_ϕ (rad/s)	6.3530	6.0729	5.7713
Natural Period Model, T (s)	1.0278	1.0411	1.1078
Decaying Factor, v	0.2793	0.2969	0.3623
Non-Dimension Damping Coefficient, *k*	0.0440	0.04889	0.06278
Improvement (%)		11.1865	42.7660

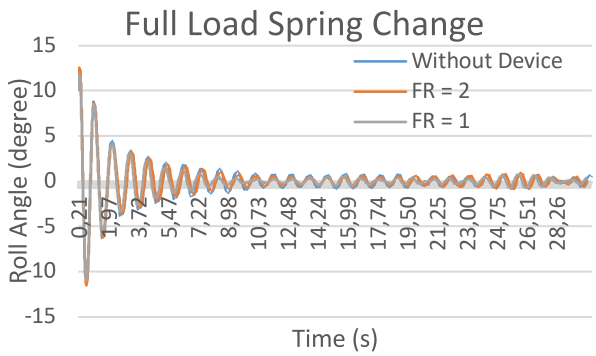

Fig. 6. Roll decay experiments for full load condition spring change

TABLE V. ROLL DECAY ANALYSIS RESULT FOR FULL LOAD CONDITION SPRING CHANGE

Parameters	**Without Device**	**FR = 2**	**FR = 1**
Critical Damping, b_c (tonnes m2/s)	0.04244	0.04169	0.04169
Damping Coefficient, b (tonnes m^2/s)	0.001866	0.002037	0.002891
Virtual Mass Moment of Inertia, a (tonnes m^2)	0.00334	0.00343	0.00343
Natural Frequency Model, ω_ϕ (rad/s)	6.3530	6.0729	6.1624
Natural Period Model, T (s)	1.0278	1.0411	1.026
Decaying Factor, v	0.2793	0.2969	0.4214
Non-Dimension Damping Coefficient, *k*	0.0440	0.0489	0.0684
Improvement (%)		11.1865	55.5183

Based on data from Table 4 and 5, an improvement on non-dimensional damping coefficient k in frequency ratio equal to

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one using spring change is higher than using mass change. The same thing happened in half-load conditions as seen in Table 6 and 7 below. The application of a softer spring allows the mass to move further so it is implicated in higher energy of rolling vessel absorbed.

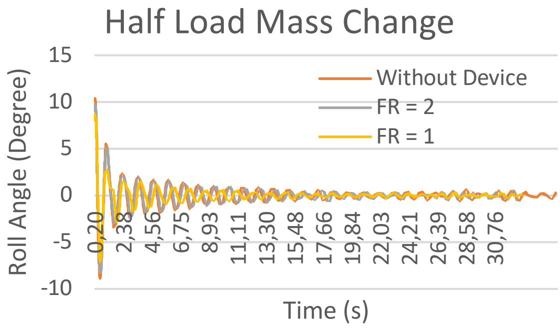

Fig. 7. Roll decay experiments for half load condition mass change

TABLE VI. ROLL DECAY ANALYSIS RESULT FOR HALF LOAD CONDITION MASS CHANGE

Parameters	Without Device	FR = 2	FR = 1
Critical Damping, b_c (tonnes m2/s)	0.03218	0.03289	0.03267
Damping Coefficient, b (tonnes m^2/s)	0.001258	0.001308	0.001766
Virtual Mass Moment of Inertia, a (tonnes m^2)	0.00284	0.002932	0.00305
Natural Frequency Model, ω_ϕ (rad/s)	6.3530	5.6097	5.3586
Natural Period Model, T (s)	1.0278	1.134	1.206
Decaying Factor, v	0.2215	0.2231	0.2895
Non-Dimension Damping Coefficient, *k*	0.03915	0.03976	0.05402
Improvement (%)		1.9468	38.5078

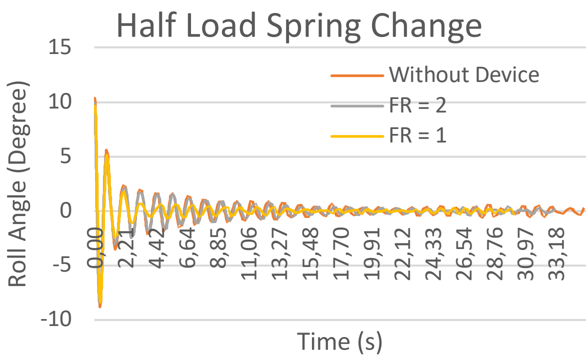

Fig. 8. Roll decay experiments for half load condition spring change

TABLE VII. ROLL DECAY ANALYSIS RESULT FOR HALF LOAD CONDITION SPRING CHANGE

Parameters	Without Device	FR = 2	FR = 1
Critical Damping, b_c (tonnes m2/s)	0.03218	0.03320	0.03320
Damping Coefficient, b (tonnes m^2/s)	0.001258	0.001407	0.001851
Virtual Mass Moment of Inertia, a (tonnes m^2)	0.00284	0.00293	0.00293
Natural Frequency Model, ω_ϕ (rad/s)	6.3530	6.0729	6.1624
Natural Period Model, T (s)	1.0278	1.0411	1.026
Decaying Factor, v	0.2215	0.2402	0.3160
Non-Dimension Damping Coefficient, *k*	0.03915	0.04235	0.05845
Improvement (%)		8.1774	49.2992

Figure 9 and 10 show a comparison of spring or mass adjustment with no adjustments following vessel load change. In both cases Table 8 and 9 indicate the adjustment resulting in improvement. When vessels load decreases, the natural frequency also impacted to become lower as the KG of the vessel were changed [10]. Therefore, to maintain the FR design and initial performance, it needed to do some adjustments.

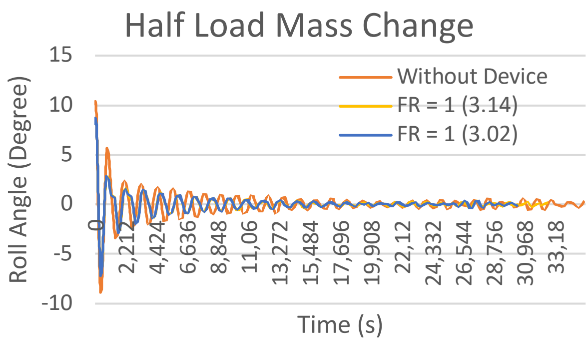

Fig. 9. Comparison roll decay experiments for half load condition spring change

TABLE VIII. COMPARISON ROLL DECAY ANALYSIS RESULT FOR HALF LOAD CONDITION MASS CHANGE

Parameters	Without Device	FR = 1 (3.02)	FR = 1 (3.14)
Non-Dimension Damping Coefficient, *k*	0.03915	0.04927	0.05402
Improvement (%)			9.65

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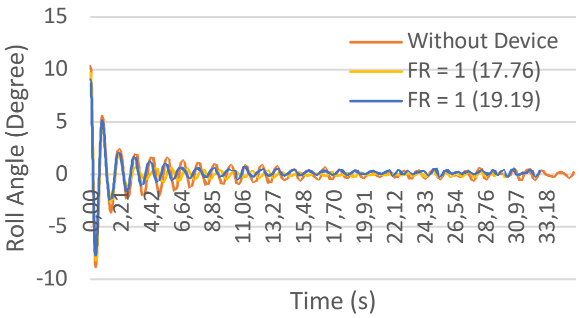

Fig. 10. Comparison roll decay experiments for half load condition spring change

TABLE IX. COMPARISON ROLL DECAY ANALYSIS RESULT FOR HALF LOAD CONDITION SPRING CHANGE

Parameters	Without Device	FR = 1 (19.19)	FR = 1 (17.76)
Non-Dimension Damping Coefficient, k	0.03915	0.055	0.05845
Improvement (%)			6.26

IV. CONCLUSION

By conducting a roll decay test, it could be known that a moving mass device success to reduce the amplitude of rolling. From two frequencies that had been tested, FR was equal to 1 giving more effectiveness than FR equal to 2. There are two setups for a moving mass device to get FR equal to 1, either adjusting the spring or the masses. Of the two available options, changing the spring is the more rational option to choose. Without the need for additional displacement, changing the spring is more effective at 8.9% for full load conditions and 8.19% for a half load condition than mass adjustment. Generally moving mass devices are designed only for one condition which is full load condition. However, when vessels operate at the sea there is always the possibility to occur load changes. Adjusting again the mass or spring of the devices following half load condition can give some improvement until 9.65% and 6.26 % respectively compared to maintaining the initial setup. The ability to change the part of the device can be an answer to the weakness of the moving mass stabilizer that is only set up for one adjustment without the need to sacrifice it is strengths which is simple and affordable.

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