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Application of nonlinear integer programming for vibration reduction optimum design of ship structure[†]

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Abstract

This paper presents a hybrid optimization algorithm which combines an external call type optimization method and a general stochastic iterative algorithm for the nonlinear integer programming with genetic algorithm (GA). GA can rapidly search the approximate global optimum under a complicated design environment such as a ship structure. Meanwhile it can handle optimization problems involving discrete design variables. In addition, there are many parameters that have to be set for GA which greatly affect the accuracy and calculation time of the optimum solution. However, the setting process is difficult for users, and there are no rules to decide these parameters. Therefore, to overcome these difficulties, the optimization of these parameters has been also conducted by using GA itself. It is proven using the trial function that the parameters are optimal. Finally, the verification of validity and usefulness of nonlinear integer programming is performed by applying this method to the compass deck of a ship where the vibration problem is frequently occurs.

Keywords: Combinatorial optimization; Genetic algorithm; Nonlinear integer programming; OPTSHIP; Ship structure; Vibration reduction

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1. Introduction

Optimization is utilized to determine the size or the geometric shape of the structure to obtain the maximum performance using minimal material with safety and availability of the target structure [1]. From a mathematical point of view, the optimization is a process to obtain the design variables which are the maximizing or minimizing a desired objective function while satisfying the prevailing constraints. Usually, optimization needs a great deal of time to obtain the desired information due to the repetitive process. Recently, optimization has been widely applied for decreasing the weight of structure in various industrial fields such as aerospace, civil engineering, mechanical engineering etc., through integrating methodology of engineering design with the technology of computer-aided engineering (CAE) and fast computer speed.

In the shipbuilding industry, optimum design principles have been used in many areas. However, the applications are limited and the most of researches have emphasized static optimization which did not consider dynamic factors [2-5]. Also, optimum design for ship vibration has been rarely studied. Yang et al. [6] worked on optimization of ship stiffened panel, and Kitamura et al. [7] carried out the optimal structural design of a ship engine room. They did the optimization considering static and dynamic constraints, and adopted a simplified analysis model to reduce the

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calculation time during the optimization process. Yang et al. [8] proposed a new optimization tool, called OPTSHIP which combined NASTRAN that was used as a solver with global optimization algorithm, namely random tabu search method (R-tabu) to enhance the optimum design for vibration reduction of the ship structure. In OPTSHIP, NASTRAN is called externally and used for calculation of the objective function. They applied it to the vibration optimal design of global and local containership using continuous variables, respectively. Also, Kong et al. [9] developed the IEOA(Integrated evolutionary optimization algorithm) and applied it to the optimum design of ship structures.

In general, the final design variables that have been chosen are bigger in size than optimized results to consider the safety margin in an actual application. Of course, this choice is possibly to make the structure more strong than the optimized model. But, the natural frequency of a structure may be more close to its resonance and more danger than the optimized design for the vibration aspects [10, 11]. Also, these optimization results are not suitable for actual application since the selected sizes of web and girder beam are limited in standard shape steel members that are commercially available. Therefore, the real values of programming need to be extended to nonlinear integer programming (NIP) in order to apply directly the optimized results to an actual design. NIP was suggested by Reiter and Rice for solving a general quadratic programming problem in 1966, where both the objective and constraint function are quadratic. They applied a modified gradient-type method, very similar to the methods used in the continuous nonlinear programming field, to solve the problem. NIP is an intrinsically hard problem. There literature is rich on the NIP problems [12, 13]. However, many of the NIP problems are computationally intractable and their solutions are nondeterministic polynomial (NP) complete. Thus, the optimal solutions cannot be obtained in a reasonable amount of time and memory [14]. Heuristic algorithms were developed to find approximations to the optimum. Current research is on the effective approximation methods such as genetic algorithm (GA) [15], simulated annealing (SA) [16] and tabu search (TS) [17]. These methods are mainly used to solve combinatorial optimization problems. Recently, it is remarkable to apply a GA to solve a combinatorial problem effectively as one of solution methods. GA is a very powerful tool for solving an NIP problem such as optimal design of system reliability and can handle any kind of objective functions and constraints.

In this paper, a method for solving the NIP problem is presented to easily get the best compromise solution while holding a nonlinear property by using the genetic algorithm for an actual design. GA is used to obtain global solutions in the proposed method. As we know, there are many parameters that have to be set for GA, such as the population size, mutation probability, crossover probability, selection methods and crossover methods that greatly affect the accuracy and calculation time of optimum solution. The setting process is difficult for users, and there are no rules to decide these parameters. To overcome these difficulties, the optimization for these parameters has been also conducted by using GA itself. The reliability of the proposed method has been demonstrated for solving the vibration problem on compass deck of a ship.

2. External call type optimization method (OPTSHIP)

Fig. 1 shows the flowchart of the OPTSHIP which shows the external call type optimization method developed by authors [8]. The OPTSHIP uses MSC/NASTRAN [18] as a solver to estimate a user-defined objective function. To run the OPTSHIP, a user-defined objective function, a design variable set and an analysis model file are needed. The OPTSHIP consists of five modules: initiation module, optimization module, interface module with NASTRAN, estimation module of the objective function and base module. The term "module" is not intended as an independent execution of each module, but to emphasize functional specialization of each module. All modules are functionally related to each other and needed to execute the OPTSHIP.

A more detailed sequential description about the optimization process of the OPTSHIP is as follows:

Step 0: An analysis model file is made by PATRAN or CAD and then the information of an objective function and a design variable set is determined and are saved into a file.

Step 1: The base module activates the initiation module.

Step 2: The analysis model, the information of a design variable set and an objective function are loaded. The analysis model and the required results

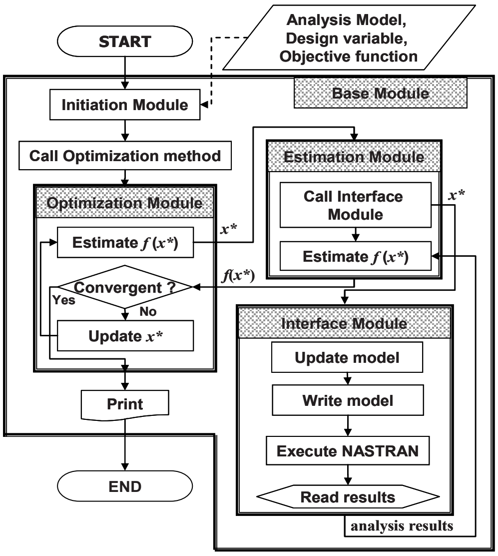


Fig. 1. Flowchart of OPTSHIP.

to estimate the objective function are informed to interface module.

Step 3: The base module activates the optimization module.

Step 4: The optimization module activates the estimation module and passes a trial design variable set x^* to the estimation module.

Step 5: The trial design variable set x^* is passed to the interface module. The interface module is activated.

Step 6: The analysis model is updated with consideration of the trial design variable set x^*.

Step 7: The updated analysis model is written into a NASTRAN input file.

Step 8: Interface module executes the NASTRAN.

Step 9: The analysis results by NASTRAN are loaded, which depends on the objective function.

Step 10: The selected results are returned to the estimation module.

Step 11: The objective function value is estimated by the analysis results and returned to the optimization module.

Step 12: The convergence condition is estimated.

Step 13: If the condition is satisfied, then the optimized design variable is returned to the base module and the optimization module is terminated. If it is not satisfied, then a new trial design variable set is generated. The procedure returns to step 4. The updating method of the design variable depends on the selected optimization method. However the generation of a new design variable is generalized by $x^* = x^* + \Delta x$ where Δx is the increment of a trial design variable which depends on the optimization method.

Step 14: Base module prints the optimized design variable set and the optimized analysis model. The OPTSHIP is terminated.

3. Nonlinear integer programming (NIP)

As with most domains of engineering, nonlinear problems are often solved by generating a solution sequence to linear problems which in some sense approximates the original nonlinear problem. The NIP problem can be mathematically expressed as follows:

$$\begin{array}{ll} \text{Maximize (or minimize)} & f(x) \\ \text{subject to the constraints} & a \le x \le b,\ x \in Z^n, \end{array}$$

where, $x = (x_1, x_2, \cdots, x_n)^T$ is a vector of variables or unknown in the NIP problem, Z^n is a set of n-dimensional integer vectors, $a = (a_1, a_2, \cdots, a_n)^T \in Z^n$ are $b = (b_1, b_2, \cdots, b_n)^T \in Z^n$ are n dimensional constant vectors, and $a \le b$.

Let $S = \left\{ x : a \le x \le b, x \in Z^n \right\}$ denote a solution space; thus $f : S \to R$ is a cost function. Some of the NIP problems can also be viewed as integer and combinatorial optimization problems [12].

4. Genetic algorithm (GA) parameter optimization

GA is a directed random search technique, first proposed by Holland [15], which can find the global optimal solution in complex multi-dimensional search spaces. The GA consists of three main strategies: reproduction, crossover and mutation. Using reproduction in the GA, individuals are selected from the population and recombined, producing offspring, which will comprise the next generation. Successive populations are produced primarily by the operations of selection, crossover and mutation. [19, 20]

However, GAs has also the following drawbacks or limitations:

- A binary code is not free to make a genotype of individuals.
- The fittest individual may be lost during the selection process due to its stochastic nature.
- Fit individuals may be copied several times and a fit

individual may quickly dominate the population at an early stage, especially, if the population size is small.

- The selection operation alone explores no new points in a search space. In other words, it cannot create new schemata.
- Different genetic parameters such as population size, crossover probability, mutation probability and etc. greatly affect the accuracy and calculation time of optimum solution.

As mentioned above, the initial parameter setting of GA is hard for users that influences the optimization results. For example, a proper mutation probability can increase the probability for a getting a global optimum solution due to the diversity of solutions, but high mutation probability has an effect on the convergence speed. Also, population size is critical to get a precise solution. If population size is very small, it maybe fails to reach the optimal solution; and if not, it brings out falling-off in convergence speed.

In this paper, the optimization for GA parameters is carried out based on GA itself using trial function. The flowchart is shown in Fig. 2. Where, N_e, N_{ea} and A_{ne} mean the number of evaluations, the number of all evaluations and the number of average evaluations, respectively. *GAF* represents GA for function optimization. *GAP* means GA parameter optimization. *GAP* consists of design variables with *GAF*'s parameters, namely, population size, crossover probability, mutation probability, selection method and crossover method. When the *GAF* is terminated, the individual fitness of *GAP* is determined on the number of average evaluations of the objective function in *GAF*. *GAF* will be terminated if the condition of Eq. (7) is satisfied. Since GA is a probability search, we did the same process M times ($M = 5$) using the same parameters, and then obtained the number of average evaluations. The objective function of *GAF* is defined as the trial function Eq. (4). Design variables and constraints are expressed as Eqs. (5) and (6). This trial function has a global solution ($f(x = 0) = 0$) and 27^N local solutions. There are ten design variables that are the same with the number of applied structures, shown in Eq. (5).

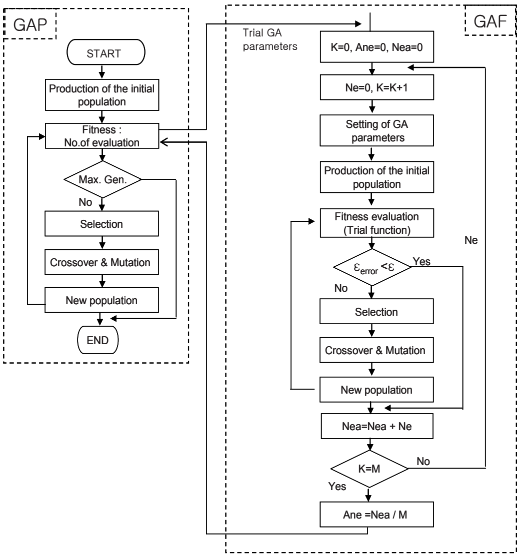


Fig. 2. Flowchart of GA for parameter optimization.

4.1 Formulation for optimization

In this paper, five GA parameters have been considered to optimize: population size, crossover probability, mutation probability, selection method and crossover method which have an effect on genetic calculation, shown in Eq. (2).

4.1.1 Formulation for GAP

Minimize

$$f(x) = A_{ne} \; (= N_{ea}/ M) \tag{1}$$

where M means the number of evaluation of GAF for identical GA's parameters, here $M = 5$.

Design variables

$$x = \{P_s \; P_c \; P_m \; M_s \; M_c\}^{\mathrm{T}} \tag{2}$$

Subject to:

$$P_s = \{10, 20, 30, \cdots, 180, 190\} \tag{3}$$

$P_c = \{0.1, 0.2, \cdots, 0.8, 0.9\}$

$P_m = \{0.05, 0.01, \cdots, 0.065, 0.9, 0.95\}$

M_s = {roulette wheel selection, ranking based selection}

M_c = {simple crossover, multi-point crossover, uniform crossover}

where, P_s, P_c and P_m are population size, crossover probability and mutation probability of GAF, respectively. And M_s and M_c represent selection method and crossover method.

4.1.2 Formulation for GAF

Minimize

$$f(x) = \sum_{i=1}^{N}[x_i^2 - \alpha_i \cos(\frac{2\pi x_i}{\beta_i}) + \alpha_i] \tag{4}$$

Design variables:

$$x = \{x_1 \ x_2 \ x_3 \ x_4 \ x_5 \ x_6 \ x_7 \ x_8 \ x_9 \ x_{10}\}^T \tag{5}$$

subject to

$$-10 \le x_i \le 100, i = 1, \ldots, N \tag{6}$$

where, $\alpha_i = 1$, $\beta_i = 4$, $N = 10$.

Table 1. Comparison of GA's parameter before and after optimization.

Parameter	Original	Optimum
Population size	100	10
Crossover probability	0.8	0.1
Mutation probability	0.1	0.065
Selection method	Roulette	Roulette
Crossover method	Uniform	Simple

Table 2. Comparison of the optimization results according to GA's parameter.

Parameters		Function value	No. of evaluation
Population size	10	0.98304	242
	20	0.98304	758
	30	0.99246	1000
Crossover probability	0.1	0.98304	242
	0.2	0.99469	1000
	0.6	0.99524	1000
Mutation probability	0.02	0.98304	392
	0.04	0.99159	1000
	0.065	0.98304	242
	0.08	0.99410	1000
Selection method	Roulette wheel	0.99004	1000
	Ranking based	0.98304	242
Crossover method	Simple	0.98304	242
	Multi point	0.99004	1000
	Uniform	0.99004	1000

The termination condition of GAF is as follows :

$$\varepsilon_{error} = \frac{1}{N} \sum_{i=1}^{N} \frac{\left| x_i^{(best)} - x_i^{(opt)} \right|}{\Delta x_i} \le \varepsilon \tag{7}$$

where, ε is predefined value, here 0.01, $x_i^{(best)}$ is the best solution at each generation, $x_i^{(opt)}$ is the optimum solution of *i*th design variable. Δx_i represents the interval of design variables.

The optimization results are shown in Table 1.

To confirm the validity of the optimization results, the objective function is evaluated by using other parameters and optimum parameters. The compared results are shown in Table 2 and Figs. 3-7. According to the results, the optimum parameters are good for the accuracy and speed of convergence in GA. Based on the above demonstration, the optimum GA parameters can be used for the integer optimum design of a compass deck.

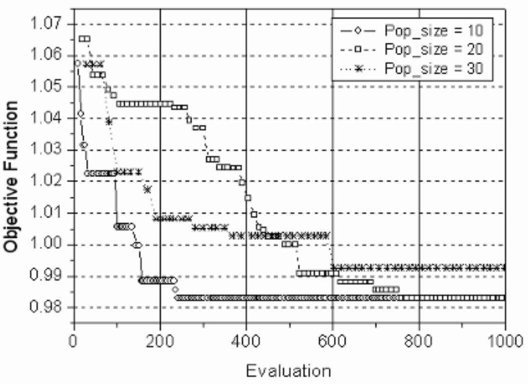


Fig. 3. Comparison of the population size.

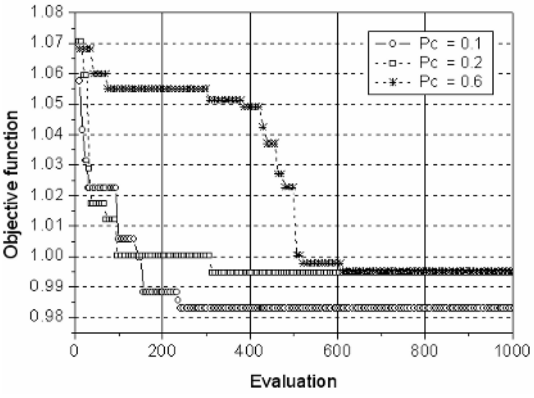


Fig. 4. Comparison of crossover probability.

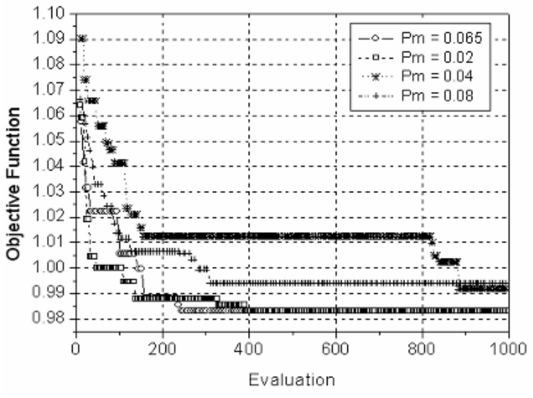


Fig. 5. Comparison of mutation probability.

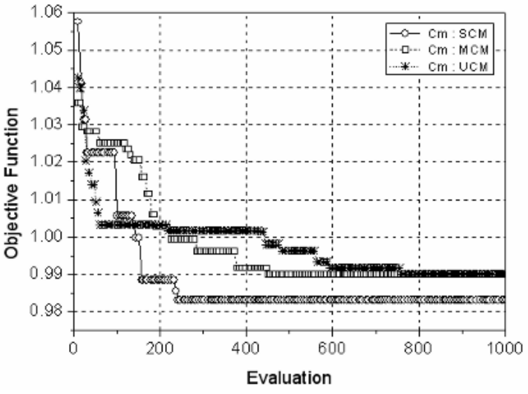


Fig. 7. Comparison of crossover method.

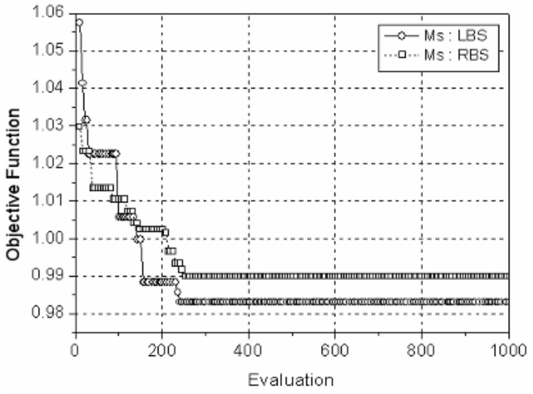


Fig. 6. Comparison of selection method.

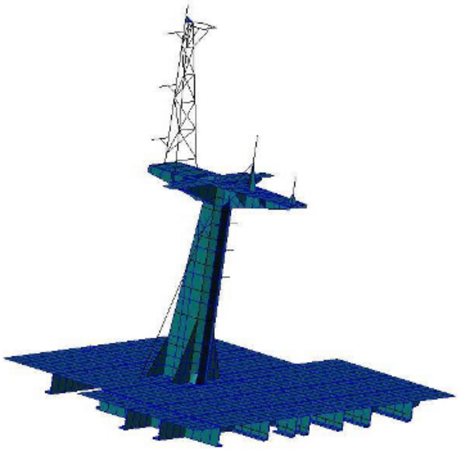


Fig. 8. Model of compass deck and radar mast.

5. Vibration analysis of compass deck

The vibration analysis of a compass deck is performed by using NASTRAN which is a commercial finite element program widely used for big structures such as a ship. Fig. 8 shows the model of a compass deck and radar mast. In particular, the girder and web of the compass deck is displayed in three dimensions, which are design variables in this study. Fig. 9 shows the arrangement of stiffeners and boundary conditions of the compass deck. The main dimensions of the subject vessel are shown in Table 3 and the main modeling data of the compass deck are listed in Table 4.

Considering the precision of analysis and time consuming modeling process, the modeling range of the compass deck is constrained to its deck only based on experience of analysis and impact test of the shipyard. The boundary conditions for the model are specified: the simple supports are used to the bulkheads shown as solid lines and two pillars connected between compass deck and navigation deck. Fixed supports are used at the cross-points of bulkheads. The arbitrary box is modeled at the location of radar mast and the weight with adjusting the mass density is considered because the weight of the radar mast on the compass deck has much effect on the vibration behavior of the compass deck. Table 5 shows the specification of the main excitation sources.

In general, the design for avoiding local structural resonance of the ship is required so that the natural frequency of the structure must be two times higher than the blade passing frequency of the propeller under the maximum main engine speed(rpm). In this study, the design target frequency is set above 18.87 Hz which is considered a safety margin and twice blade passing frequency of the propeller (16.33Hz).

Table 3. Principal dimensions.

Length overall	208 m
Length between perpendiculars	196 m
Breadth moulded	29.8 m
Depth moulded	16.4 m
Draft design	10.2 m

Table 4. Main data of modeling.

Geometry data		Material data	
Plate thickness	8.0 m	Elastic modulus	206 GN/m^2
Web & girder size	250×90×10/15A	Poisson ratio	0.3
Frame/ longitudinal space	800 mm	Mass density	7850 kg/m^3

Table 5. Specification of main excitation sources.

Excitation source	MCR	Excitation	
		Order	Frequency
Main engine (6RTA72U)	98 rpm	3rd	4.90 Hz
		4th	6.53 Hz
		6th	9.80 Hz
Propeller (Blade: 5EA)		1st	8.17 Hz
		2nd	16.33 Hz

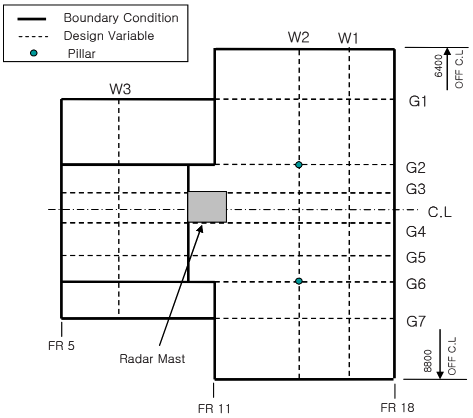
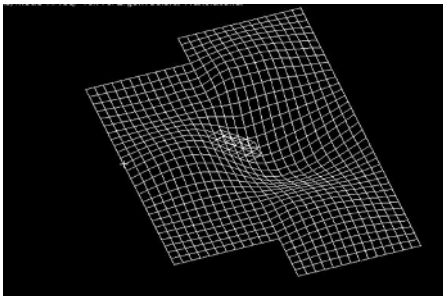
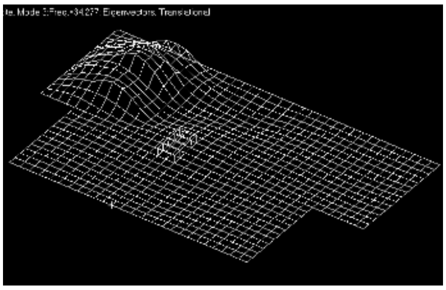


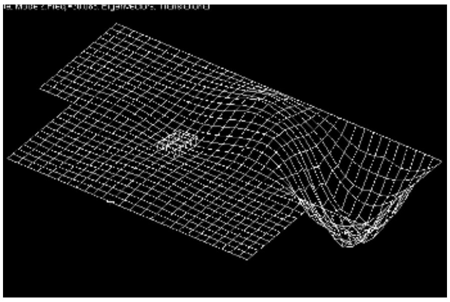
Fig. 9. Design variables and boundary conditions of compass deck.



(a) 1st mode (16.78 Hz)



(b) 2nd mode (30.09 Hz)



(c) 3th mode (34.28 Hz)

Fig. 10. Mode shape of compass deck.

Fig. 10 shows the first three modes and natural frequencies of the compass deck structure by NASTRAN. The 1st mode (16.78 Hz), which frequently occurs on the compass deck during the voyage, is the vertical mode on the front area of the radar mast as shown in Fig. 10(a). The lower part of the compass deck could not be installed the bulkhead because of securing the workspace compared to the other cabins. Therefore, the corresponding weak stiffness of the structure results in low natural frequency which is close to the main excitation source of the ship. In this model, the 1st natural frequency of structure is also within the resonance region where twice blade passing frequency of the propeller is 16.33 Hz. The safety margin is only 2.8 %, which is usually 10 %. The 2nd and 3rd modes occur on the sides of the compass deck. Their natural frequencies are higher than the main excitation frequency of the ship and the possibility of resonance is rare. So, in order to design the safety structure, the 1st vertical mode of compass deck is specified as the concerned mode in this study.

6. Optimum design of compass deck

6.1 Formulation for optimum design

6.1.1 Design variables

The main vibration mode on the compass deck is the global mode of the girder and web in the vertical direction. One of the most important factors is the stiffness of the girder and web. In this study, the size of the girder and web on compass deck in Fig. 9 is defined as design variables in Eq. (8)

$$x = \{SW_1\ SW_2\ SW_3\ SG_1\ SG_2\ SG_3\ SG_4\ SG_5\ SG_6\ SG_7\}^T \tag{8}$$

where SW and SG mean the size of girder and web, respectively.

6.1.2 Constraints

The web length of stiffener L_w is restricted as in Eq. (9) due to ceiling height, namely the distance from the navigation deck to the compass deck, which is based on the building specification. The stiffener is also restricted to available standard sizes in the field as shown in Table 6.

$$200 \le L_w \le 550 \text{ mm} \tag{9}$$

Table 6. Corresponding cross section of steel members.

Stiffener size	$L_w \times L_f \times T_w / T_f$
200A	200 × 90 × 9/14
250A	250 × 90 × 10/15
300A	300 × 90 × 11/16
350A	350 × 100 × 12/17
400A	400 × 100 × 12/18
450A	450 × 125 × 11.5/18
500A	500 × 150 × 11.5/18
550A	550 × 150 × 12/21

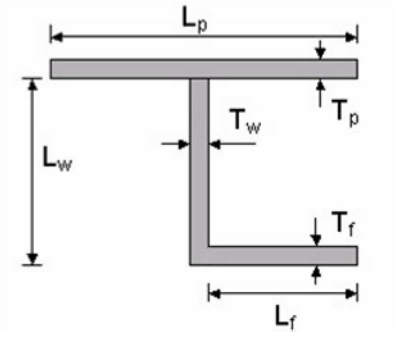


Fig. 11. Section of stiffener and plate.

Also, the basic concept of local vibration design is the minimization of the response at each point. However, it is difficult to evaluate how much the excitation force influences the local structure. So, in this study, the natural frequency of the structure is restricted as Eq. (10) which is considered a safety margin with twice blade passing frequency of the propeller.

$$\omega_n \ge 18.87\text{Hz} \tag{10}$$

Fig. 11 shows the section of stiffener and plate.

6.1.3 Objective function

In general, the main target of the optimum design is to decrease the weight of the structure or to reduce the vibration level on the specific point with avoiding the resonance between the excitation source and the natural frequency of the subject structure. In this study, we considered the objective function as two cases as follows:

Case 1: The objective function combines linearly the weight of the compass deck, W_1 with maximum vibration velocity response, R_1 at a range of interest (below MCR rpm) like in Eq. (11)

$$\text{Minimize} \quad f(x) = \alpha\left(\frac{W_1}{W_0}\right) + \beta\left(\frac{R_1}{R_0}\right) \tag{11}$$

where, α and β are weighting factors, W_0 means initial weight, R_0 is a basis vibration velocity response(vertical direction, the maximum amplitude at center).

Case 2: The objective function combines linearly the weight of compass deck, W_1 with natural frequency of structure expressed by Eq. (12). This object is to get an economic and sound structure to reduce the weight of stiffener and to increase the natural frequency.

$$\text{Minimize} \quad f(x) = \alpha\left(\frac{W_1}{W_0}\right) + \beta\left(\frac{\omega_0}{\omega_t}\right) \tag{12}$$

where, ω_t and ω_0 mean target and current natural frequency, respectively. α and β are weighting factors.

6.2 Result of optimization and discussion

The optimum design was carried out to obtain an optimal size of web and girder on the compass deck

Table 7. Comparison of original and optimal design variables for case 1.

Design variable	Original	Optimum	Remarks
W_1	250	200	– 20%
W_2	250	200	– 20%
W_3	250	200	– 20%
G_1	250	200	– 20%
G_2	250	200	– 20%
G_3	250	450	+ 80%
G_4	250	200	– 20%
G_5	250	200	– 20%
G_6	250	200	– 20%
G_7	250	200	– 20%

Table 8. Comparison of original and optimal design variables for case 2.

Design variable	Original	Optimum	Remarks
W_1	250	200	– 20%
W_2	250	200	– 20%
W_3	250	200	– 20%
G_1	250	200	– 20%
G_2	250	200	– 20%
G_3	250	200	– 20%
G_4	250	550	+ 120%
G_5	250	200	– 20%
G_6	250	200	– 20%
G_7	250	250	0%

Table 9. Comparison of results for case 1.

Item	Original	Optimum	Remarks
Natural frequency	16.78 Hz	18.91 Hz	12.69%
Response at MCR	10.50 mm/s	4.07 mm/s	– 61.24%
Weight	2760 kg	2537 kg	– 8.08%

Table 10. Comparison of results for case 2.

Item	Original	Optimum	Remarks
Natural frequency	16.78 Hz	23.28 Hz	38.74%
Response at MCR	20.17 mm/s	0.69 mm/s	– 93.48%
Weight	2760 kg	2757 kg	– 0.11%

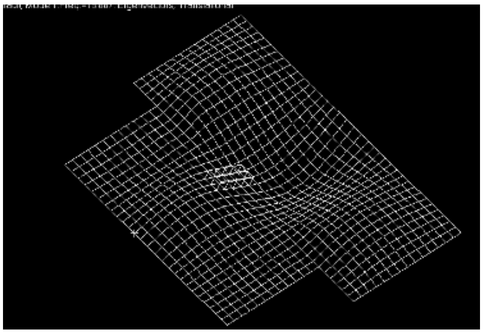


Fig. 12. Mode shape of compass deck after optimization.

to maintain the anti-vibration design of it. Nonlinear integer algorithm by GA is used as an optimal algorithm in order to apply directly the optimized result to an actual design. As above stated in section 5, the optimum GA parameters are applied to this problem.

Tables 7 and 8 show the results of the design variables before and after optimization for case 1 and case 2, respectively. The center girder of structure G3 is shown to increase 80% and the others are reduced 20% in case 1. In case 2, the center girder of structure G4 is increased 120 % and the others are similar to case 1.

These results indicate that the most reasonable modification method is to increase the stiffness of the member where the maximum amplitude exists in vibration mode. To get a higher natural frequency of structure, the stiffness of G_4 must be increased, which is located in a wider area than G3. The role of G_7 in case 2 supports the stiffness of G_4 due to the limitation of stiffener size. Tables 9 and 10 show the natural frequency, vibration response at an MCR in a unit excitation force and the weight of the compass deck before and after optimization for case 1 and case 2, respectively. According to the results, the 1st natural frequency increased 12.69% and 38.74% from 16.78 Hz to 18.91Hz and 23.28 Hz, and the safety margin with twice passing frequency of propeller correspondingly changed from 2.80% to 15.80% and 42.60% for case 1 and case 2, respectively. Therefore, the structure is free from the resonance. Moreover, the amplitude of vibration velocity response for case 1 and case 2 is reduced 61.24% and 93.4%.

The weight of stiffeners which is applied to design variables also decreased in spite of higher natural frequency and lower the vibration response.

In summary, the local vibration problems which avoid structural resonance through the movement of natural frequency without additional weight has been successfully solved by the proposed optimization method. Fig. 12 shows the 1st vibration mode after optimization. Although we noticed that there was a little change on the 1st vibration mode shape due to the mode of global compass deck, the natural fre quency

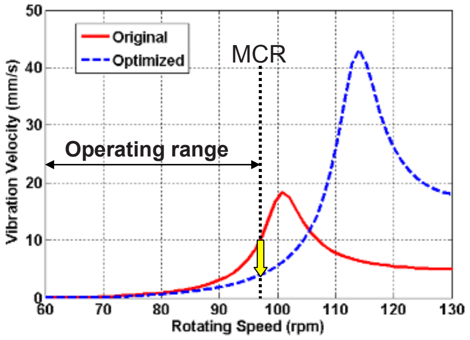


Fig. 13. Comparison of response between original and optimum results for case 1.

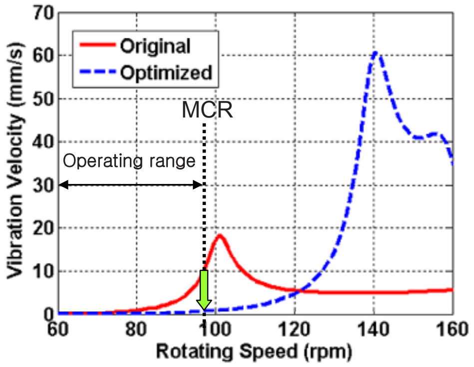


Fig. 14. Comparison of response between original and optimum results for case 2.

was increased based on calculation results. And we confirmed that the vibration response at the MCR rpm has been significantly reduced as shown in Fig. 13 for case 1 and Fig. 14 for case 2, respectively.

7. Conclusions

A method for solving the NIP problem is proposed for obtaining the best compromise solution while holding a nonlinear property by using the genetic algorithm and is applied to the vibration optimum design of real ship. GA is used to obtain global solutions in the proposed method. Also, in order to get proper GA parameters, the optimization of GA parameters is also carried out through the trial function by GA itself. The reliability of the proposed method has been demonstrated for solving the vibration problem on the compass deck of a ship. After optimization, local vibration problem has been successfully solved: the structure is free from the resonance, safety margin increased, and the amplitude of vibration velocity response reduced without additional weight. The results indicated that the proposed method can be used as an optimum.

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References

[1] J. S. Arora, *Introduction to Optimum Design*, McGraw-Hill, New York, USA, (1989).

[2] C. D. Jang, S. I. Seo and K. K. Kim, A study on the optimum structural design of surface effect ships, *Marine Structure* (1996) 519-544.

[3] H. W. Leheta and A. E. Mansour, Reliability-based method for optimal structural design of stiffened panels, *Marine Structure* 10 (1997) 323-352.

[4] M. Valorani, D. Peri and E. F. Campana, Sensitivity analysis methods to design optimal ship hulls, *Optimization and Engineering* 4 (2003) 337-364.

[5] O. Reyhan, A review of recent ship vibration papers, *The Shock and Vibration Digest* 36(3) (2004) 207-214.

[6] B. S. Yang, B. G. Choi, Y. H. Yu and H. T. Nan, Optimum design of a damping plate with an unconstrained viscoelastic damping layer using combined genetic algorithm, *Trans. KSME International Journal* 13(5) (1999) 387-396.

[7] M. Kitamura, H. Nobukawa and F. Yang, Application of a genetic algorithm to the optimal structural design of a ship's engine room taking dynamic constraints into consideration, *Marine Science and Technology* 5 (2000) 131-146.

[8] B. S. Yang, Y. M. Kong, S. H. Choi, S. I. Chae, J. D. Song and Y. H. Kim, Development of NASTRAN-based Optimization Framework for Vibration Optimum Design of Ship Structure, *Trans. of the Korean Society for Noise and Vibration Engineering* 15(11) (2005) 1223-1231.

[9] Y. M. Kong, S. H. Choi, B. S. Yang and B. K. Choi, Development of Integrated Evolutionary Optimization Algorithm and its Application to Optimum Design of Ship Structures, *Journal of Mechanical Science and Technology* 22(7) (2008) 1313-1322.

[10] T. Yokota, M. Gen and Y. X. Li, Genetic algorithm for non-linear mixed integer programming problems and its applications, *Computer in Industrial Engineering* 30(4) (1996) 905-917.

[11] O. B. Augusto and A. Kawano, A mixed continuous and discrete nonlinear constrained algorithm for optimizing ship hull structural design, *Ocean Engineering* 25(9) (1998) 793-811.

[12] G. L. Nemhauser and L. A. Wolsey, *Integer and Combinatorial Optimization,* John Wiley and Sons, New York, USA, (1988).

[13] R. G. Parker and R. L. Rardin, *Discrete Optimization*, Academic Press Inc., New York, USA (1988).

[14] M. R. Garey and D. S. Johnson, *Computers and Intractability: A Guide to the Theory of NP-Completeness*, W.H. Freeman and Company, San Francisco, USA, (1979).

[15] J. H. Holland, *Adaptation in Natural and Artificial Systems*, The University of Michigan Press, Michigan, USA, (1975).

[16] S. Kirkpatrick, C. D. Gelatt and M. P. Vecchi, Optimizations by simulated annealing, *Science* 220 (1983) 671-680.

[17] F. Glover, Future paths for integer programming and links to artificial intelligence, *Computers and Operations Research* 13(5) (1986) 533-549.

[18] G. Moore, *MSC.NASTRAN Design Sensitivity and Optimization, User's Guide*, The MacNeal-Schwendler Corp. USA, (1994).

[19] B. G. Choi and B. S. Yang, Optimum shape design of rotor shafts using genetic algorithm, *Journal of Vibration and Control* 6(2) (2000) 207-222.

[20] B. G. Choi and B. S. Yang, Optimal design of rotor-bearing systems using immune-genetic algorithm, *Trans. ASME Journal of Vibration and Acoustics*, 123(3) (2001) 398-401.

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