Abstract

The Energy Finite Element Analysis (EFEA) has been validated in the past through comparison with test data for computing the structural vibration and the radiated noise for Naval systems in the mid to high frequency range. A main benefit of the method is that it enables fast computations for full scale models. This capability is exploited by using the EFEA for a submarine pressure hull design optimization study. A generic but representative pressure hull is considered. Design variables associated with the dimensions of the king frames, the thickness of the pressure hull in the vicinity of the excitation (the latter is considered to be applied on the king frames of the machinery room), the dimensions of the frames, and the damping applied on the hull are adjusted during the optimization process in order to minimize the radiated noise in the frequency range from 1,000Hz to 16,000Hz. Constraints on the total amount of damping that can be used are considered (resource driven constraints) and structural collapse constraints are also taken into account in order to avoid degrading the structural integrity of the pressure hull. Two different optimization strategies are exercised. First a concurrent multidisciplinary analysis is performed; optimal configurations for structural performance and for acoustic radiation are identified and the results are used for producing a single design with optimized performance in both disciplines. Then, an analysis based on set-based design principles is performed. The latter identifies several alternative and diverse hull configurations that provide similar levels of performance with respect to the radiated noise. Having several alternative solutions of nearly equal performance provides insight into the design trade-offs when configuring the pressure hull. The results from both optimization strategies are analyzed and discussed.

Introduction

The EFEA has been developed and utilized for simulating the vibro-acoustic behavior of complex systems in the mid to high frequency range where conventional finite element methods are no longer efficient [1, 2, 3, 4, 5, 6, 7, 8, 9, 10]. Finite Element Analysis (FEA) methods solve differential equations for which the fundamental solutions vary harmonically with space; thus, the wavelength becomes smaller as the frequency increases, requiring a prohibitively large number of elements and computational resources. In the EFEA method, the governing differential equations are formulated for an energy variable that has been spatially averaged over a wavelength and time averaged over a period. Differential equations are derived for all wave bearing domains within a system. Each differential equation represents a power balance over a control volume. The corresponding fundamental solutions vary exponentially with space, thus requiring only a small number of elements to capture numerically the smooth spatial variation. This feature allows large, complex systems to be analyzed with a small number of elements and in a short enough time that can influence the design cycle. Similar to the Statistical Energy Analysis (SEA), the concept of power balance and representing the energy of an acoustic or a vibrational field as the sum of the energy of each component of an expansion basis (groups of similar normal modes in SEA [11] or orthogonal waves in EFEA [6]) are used for developing the differential equations for the primary variables of the EFEA. The differences in the type of expansion basis employed by the SEA and the EFEA methods (modal basis and wave basis, respectively) lead to the difference between modeling a structural-acoustic system as an assembly of lumped parameters (SEA) or with finite elements (EFEA). As a consequence, the SEA requires the development of a completely new computational model based on a lumped parameter approach, while the EFEA can employ the same finite element model used for other finite element based simulations (e.g. structural analysis).

Joint matrices are required between the finite elements at locations where discontinuities in the primary EFEA variables exist [3, 5]. These discontinuities can originate from the geometry, from a change in material properties, from multiple components being connected together, or from different media interfacing with each other. The EFEA elements at the discontinuities are physically disconnected so that adjacent
elements have duplicate nodes at the joints. In this manner it is possible for the EFEA primary variables to acquire different values across the discontinuity in the numerical model, as dictated by the physics. The joint matrices provide the connection mechanism and describe the power transfer between the disconnected finite elements. Power transfer coefficients which are derived analytically using wave equations propagating in semi-infinite domains, along with compatibility conditions and equilibrium equations are used to characterize the transmitted and reflected intensity at a joint due to an impinging wave [12]. The power transfer coefficients are employed along with the shape functions of the EFEA elements to derive the entries for the joint matrices [5]. The latter offer the coupling mechanism between the finite elements which are present from each side of the connection. Joint matrices in EFEA have similar functionality with the coupling loss factors in SEA [11]; they provide the connection between different wave bearing domains (EFEA) or between different groups of similar modes (SEA subsystems). However, unlike the coupling loss factors, no estimates of modal densities are necessary for developing the EFEA joint matrices [5].

In Naval applications of the EFEA method, where a structure can be either partially or fully submerged, the effects of heavy fluid loading must be included. The fluid loading effects on the structure’s vibration are represented as added mass and radiation damping, and are applied only on the outer elements of the structure which are immersed in heavy fluid. This reduces the computational requirements further since the exterior fluid domain is not modeled explicitly with finite elements. Internal spaces filled with fluid, such as tanks, bilges, or an inner bottom, are modeled as an acoustic volume with the appropriate material properties. At the interface between the internal heavy fluid and structural panels, radiation coupling is considered. In addition, mass-law coupling through the flexible panel can also be included when a structural panel separates two acoustic spaces, offering a direct coupling between the two acoustic spaces separated by the flexible panel [10]. Figure 1 presents representative Naval vehicles and/or EFEA models which have been used for conducting vibration and radiated noise computations.

The pursuit of innovative Naval vehicle designs, the use of new materials, and the increasingly demanding performance requirements dictate a broadly focused thrust for addressing noise and vibration issues during the design process. This expanded technical view also requires that the types of tools used for these assessments can provide results within a short enough time that can affect the design cycle. In this paper it is demonstrated how the EFEA can be used for conducting structural-acoustic assessments during the design of a notional submarine pressure hull. Two different design strategies are presented, one based on concurrent design, and one based on principles of set based design. The merits of each design approach are demonstrated through the analysis of a notional submarine pressure hull.

**Modeling the Pressure Hull**

A comprehensive discussion about the theoretical aspects of the EFEA has been presented in [13]. The EFEA model is developed based on that information and the following items are discussed in this section: description of the physical structure which is being modeled; definition of performance metrics for which improvement is sought (objective functions); parameters which are adjusted in order to achieve the desired improvements (design variables); and performance requirements that the design must meet (constraints).

The EFEA model is presented in Figure 2 along with an original figure from [14] that provides typical structural dimensions of a notional pressure hull. Only the aft section of the pressure hull is modeled since the excitation is considered to be applied at the flanges of the king frames of the aft section, making the radiated noise emitted from other compartments small in relation to that of the aft section. The external fluid is modeled implicitly through added mass and radiation damping terms as discussed in [10]. Information about the dimensions of the hull (in meters) and the structural parts which are included in the model are depicted in Figure 3.

The EFEA model is comprised by ~7,750 elements and ~7,250 joints (i.e. connections between elements with geometric or property discontinuities). Unlike the FEA
method that only needs a library of elements to operate, the EFEA needs libraries of both elements and joints. Figure 4 shows the location where the excitation is applied on the flange of a king frame (the same force excitation is applied on both king frames). The EFEA analysis requires defining the input power as excitation into the system. In this work the prescribed unit force excitation is converted into input power using a simple equation based on the impedance of an infinite flat plate that exhibit the same characteristics with the flange plate of the king frames. The analysis is done in the frequency range of 1,000Hz to 16,000Hz. The complete frequency sweep analysis requires approximately 10min on a single processor PC. Although the excitation is not realistic, it is selected to demonstrate that a mechanical excitation (i.e. force) is supported by a foundation (flange of king frames) and eventually transmitted to the hull (web section of king frame and hull plating). In this manner, although the analysis is done using a notional hull, it contains a path that represents power transfer from an excitation to the surrounding fluid.

The design variables are highlighted in Figure 5 along with their upper and lower bounds. There are two performance metrics (objective functions) which are considered: the total radiated power from the outer surface of the pressure hull in contact with the surrounding heavy fluid medium (i.e., not the transverse bulkhead), and the total weight of the sections of the pressure hull for which the thicknesses are adjusted during the design process. Minimizing both of them simultaneously would be an ideal scenario, and alternative approaches for pursuing this are discussed in this paper.

During the design process two conditions (i.e. constraints) are imposed on the design. The first set an upper limit on the amount of damping resources which will be used. Although an upper limit of 6% structural damping is assigned to the “green” and “purple” (Figure 5) colored section of the pressure hull, the structural damping values selected for the two sections must satisfy the following condition:

$$d_G \cdot A_G + d_P \cdot A_P \leq (A_G + A_P) \cdot 0.03$$  \hspace{1cm} (1)

where $d_G$ is the structural damping of the green colored section, $A_G$ is the area of the green colored section, and $d_P, A_P$ are the corresponding values for the purple colored section. The second set of conditions is associated with the structural integrity of the pressure hull, and it is based on technical information from [15].

The following five modes of failure are considered, each of which is evaluated independently using prescribed mathematical equations relating scantling dimensions, material properties, and desired collapse pressure: shell buckling, shell yielding, general instability, frame buckling, and frame stress. A given design must satisfy all five design criteria in order to be considered acceptable. The following dimensions, material properties, and characteristics are considered for all design configurations: Nominal Frame Spacing: 1.00 m; Nominal Hull Diameter: 10.0 m; Design Submergence Depth: 250 m; Young’s Modulus, $E = 207$ GPa; Poisson Ratio, $\nu = 0.30$; Mass Density, $\rho = 7746$ kg/m$^3$; Yield Strength, $\sigma_Y = 672$ MPa.

The five collapse criteria are discussed next.

Shell Buckling - The shell buckling failure mode is characterized by the formation of discrete lobes around the periphery of the shell plating between two adjacent frames. Windenburg [16] developed the following approximate expression for the external pressure, $P_o$, resulting in failure of
the shell plating of a ring-stiffened cylinder in this mode (by considering Poisson ratio to be 0.3):

\[ P_c = \frac{2.60E \cdot t/d}{L/D - 0.45(t/D)^{1/2}} \]

(2)

where \( E \) is Young’s Modulus, \( D \) is the diameter of the shell midplane, \( t \) is the shell thickness, and \( L \) is the maximum unsupported length of shell plating (i.e., frame spacing). The relation is valid in the elastic regime (i.e., elastic buckling), and assumes the frames are widely-spaced and the shell is relatively thin.

Shell yielding is characterized by the formation of a typically-axisymmetric accordion-type pleat in the shell plating between two adjacent frames. Von Sanden and Günther [17] derived the following expressions for the pressure, \( P_c \), at which yielding will occur in the shell plating at the frame location and at midbay, respectively:

\[ P_c = \frac{2\sigma_Y \cdot t/D}{0.5 + 1.815K \cdot \left( \frac{0.85 - B}{1 + \beta} \right)} \]

(3)

\[ P_c = \frac{2\sigma_Y \cdot t/D}{1 + H \cdot \left( \frac{0.85 - B}{1 + \beta} \right)} \]

(4)

where \( \sigma_Y \) is the yield strength of the material, \( K \) and \( H \) are intermediate functions representing the effects of shell bending near the frame and at midspan, respectively, \( B \) is the ratio of shell plating area under faying flange of the frame to the total frame area plus shell plating area under faying flange, and \( \beta \) characterizes the amount of flexure allowed by the frame. Detailed definitions of \( K \), \( H \), \( B \), and \( \beta \) are provided in [17].

General Instability - A failure by general instability is characterized by the total collapse of shell plating and frames as a unit, often between effectively rigid boundaries such as transverse bulkheads, and can occur in discrete numbers of longitudinal and circumferential half-sine waves. The critical pressure, \( P_{cr} \), resulting in failure by general instability was presented in [18] and takes the following form:

\[ P_{cr} = \frac{Et}{R} \left( \frac{m^4}{n^2 - 1} + \frac{m^2}{2} \right) + \frac{(n^2 - 1) \cdot EI}{R^4L_f} \]

(5)

where, \( m = \frac{\pi R}{L} \). In the above expressions, \( R \) is the mean shell radius, \( I \) is the area moment of inertia of a typical frame plus one frame space of shell plating, \( L_f \) is the frame spacing, \( L \) is the transverse bulkhead spacing, and \( n \) is the number of circumferential half-sine waves (or lobes) in the assumed mode of failure. In general, this expression is evaluated for several integer values of \( n \), taking the lowest calculated \( P_c \) as the buckling pressure.

Frame buckling is characterized by the pressure at which a given transverse frame becomes unable to maintain circularity and provide adequate support to the shell plating. Reference [19] derived the following expression for the frame instability pressure of a ring-stiffened cylinder of infinite length, assumed to collapse in the \( n = 2 \) mode:

\[ P_f = \frac{25EI}{D_f^2 L_f} \]

(6)

where \( D_f \) is the diameter of the ring-stiffener at its neutral axis.

Frame Stress - The total stress, \( \sigma_f \), in a typical ring frame can be characterized by the superposition of compression stress, \( \sigma_C \), and bending stress, \( \sigma_B \), due to uniform external pressure on the shell plating. A requirement for the total stress not to exceed the yield strength of the material is imposed:

\[ \sigma_f = \sigma_B + \sigma_C \]

(7)

The bending stress in the ring-frames stems from the presence of eccentricity (i.e., out-of-roundness) in the hull, resulting in a reduction of collapse strength. The frame bending stress was formulated in [20], and takes the following form:

\[ \sigma_B = \frac{E c e \left( n^2 - 1 \right)}{R^2} \frac{P_{design}}{P_{cr} - P_{design}} \]

(8)

\( P_{design} \) is the desired collapse pressure, \( P_{cr} \) and \( n \) represent the collapse pressure due to general instability and the corresponding \( n \)-value as determined using Equation (5). \( R \) is the hull radius to the shell midplane, \( c \) is the distance from shell midplane to extreme fiber of a typical frame, and \( e \) is the deviation (eccentricity) from roundness. For the sake of simplicity, \( e \) was assumed to be half the shell thickness in this paper.

The compressive force in a ring frame is obtained via summation of forces on a transverse section of ring with associated shell plating. Dividing the force by the cross-sectional area yields the following expression for the compressive stress in a typical frame:

\[ \sigma_C = \frac{P_{design} \cdot QR}{A + bt} \]

(9)

\( Q \) is an intermediate parameter in determining the total radial load per length of circumference, and can be found in [20].

All the aforementioned failure criteria are checked based on the hydrostatic pressure loading associated with the desired design subsmergible depth of 250 meters.

Single Point Concurrent Design

The algorithm that is used for conducting the concurrent optimization [21] is briefly described in this Section. It is a bi-level algorithm; at the lower level it determines the optimal configuration for each performance objective (i.e. minimum mass or minimum radiated noise in this case). Then at the
upper level a consolidated single point configuration (i.e. a single design) is determined; it exhibits the best possible performance at each one of the disciplines.

At the lower level a single discipline (i.e. objective) optimization is carried out:

$$\min (Obj_i)$$

$$\text{DV}_i$$

subject to:

$$g_i(DV_i) \leq 0$$

$$h_i(DV_i) = 0$$

$$DV_{iL} \leq DV_i \leq DV_{iU}$$

where subscript “$i$” indicated the $i$th discipline; $Obj_i$ is the objective function; $DV_i$ are the design variables; $g_i(DV_i) \leq 0$ are the inequality constraints; $h_i(DV_i) = 0$ are the equality constraints; $DV_{iL}$ and $DV_{iU}$ are the lower and the upper bounds for the design variables. The optimization for each discipline is conducted first individually. The outcome from each optimization results into the $Obj_i^{\text{best}}$ value for the corresponding objective function. The discipline level optimizations need to be performed only once in the single point design process.

After each single discipline optimum has been identified, the values of all other objective functions at the single discipline optimum must also be computed in order to identify the worst value $Obj_i^{\text{worst}}$ that the objective function of a discipline can acquire at the optimal points of all other disciplines. Then the upper level optimization analysis of the bi-level algorithm is conducted. At the upper level the optimization statement is:

$$\min \left\{ \sum_{i=1}^{N} \frac{(Obj_i^{\text{best}} - Obj_i^{\text{worst}})^2}{PRR_i^i} \right\}$$

$$DV_i \text{ } i = 1, \ldots, N$$

subject to:

$$g_i(DV_i) \leq 0 \text{ } i = 1, \ldots, N$$

$$h_i(DV_i) = 0 \text{ } i = 1, \ldots, N$$

$$DV_{iL} \leq DV_i \leq DV_{iU} \text{ } i = 1, \ldots, N$$

where $N$ is the total number of disciplines (i.e. minimizing the mass or minimizing the radiated noise in this case) which are considered; $PRR_i^i$ is the Plausible Reduction Range for each discipline. The $PRR_i^i$ is defined as the difference between the best and the worst value that an objective can acquire:

$$PRR_i^i = obj_i^{\text{best}} - obj_i^{\text{worst}}$$

The scaling between the terms included in the objective function at the system level (Equation 11), is important because the Euclidian distance from the optimum and the Pareto front will favor the discipline with the highest value of objective function. In this formulation the $PRR_i^i$ for each discipline “$i$” is used for defining the scaling in the definition of the system level objective function in Equation (11).

The constraints from all disciplines are accumulated at the system level optimization in order to make certain that the concurrent optimum solution is acceptable by all disciplines. Various disciplines may exhibit different sets of commonality between their design variables. When a particular design variable is encountered in more than one discipline, it is treated as a shared design variable in the system level optimization. In the submarine pressure hull design all design variables except of the damping applied in the green and the purple colored sections are shared design variables because they are used for optimizing the radiated noise and also for reducing the weight of the hull. The two design variables associated with the damping are only used when the radiated noise is minimized.

## Design Space Reduction Based on Principles of Set Based Design (SBD)

Comprehensive technical information about using set based design (SBD) for system analysis and product development is described in [22] and the major points are outlined below:

a. SBD performs design discovery by way of elimination. SBD concentrates on eliminating infeasible and highly dominated regions of the design space.

b. Reducing the design space by selecting the best solutions is not the same as reducing the design space by eliminating the worst solutions. The reason is that the assessment of the best solutions can change significantly as the design progresses and the quality of information or the variety of analysis increases, while there is much more certainty that the best solutions will not reside in a bad region (far from the Pareto front) of the design space. Therefore it is much preferable to morph the design space through elimination of bad solutions instead of selecting best solutions.

c. SBD evaluates and makes decisions with respect to regions of a design space, not a specific configuration. In performing this evaluation, however, the design space is “sampled” by creating and analyzing specific configurations and generalizing the results across the design space. When a sample design is removed, the region of the design space where the sample design resides is also eliminated.

d. SBD differs from all other multiple criteria decision making processes by considering sets of alternatives and eliminating possibilities as opposed to finding optimal or multiple optimal points as basis of moving forward in design reasoning.

e. It is important to have a diverse set of alternative designs in the retained set of solutions in order to have viable solutions which can actually meet the performance expectations when more concrete information becomes available (i.e. exact capabilities of new technology, simulation models of increased fidelity, etc.) during later design stages.

The SBD approach is used in this work for reducing the design space, and increasing the understanding of the remaining space. According to the aforementioned main SBD...
points, the algorithm is intended to find the best region of the design space through the elimination of highly dominated and highly infeasible designs.

Instead of pursuing a single point design through solution of the optimization statement of Equation (11), an exhaustive sampling of the design space $DV_j = DV_i, i = 1, \ldots, N$ is conducted first using either a random selection or a Latin Hypercube approach. At each sampling point the objective function $e^{g_i(DV_j) \leq 0}$ and the constraint values $g_i(DV_j) \leq 0$ from Equation (11) are evaluated, but without conducting an optimization. All sampling points are divided into two groups. Group A contains all feasible configurations (i.e. no constraints are violated); all such points will be ranked in terms of the goodness of the objective function $e^{g_i(DV_j) \leq 0}$ and the constraint values $g_i(DV_j) \leq 0$. Group B contains all sampling points which violate the constraints. These points are ranked based on the number of constraints which are being violated and on the level of violation that they exhibit.

Configurations contained in Groups A and B are eliminated gradually from both groups. Within each elimination round a user defined percentage of configurations is eliminated from each group and designs from the outer boundaries of the design space are eliminated first. The criterion for eliminating points from group A is the goodness in the objective function. Configurations exhibiting the least goodness are eliminated first (i.e. highly dominated solutions). The elimination of points from Group B are based on the level that constraints are violated. Configurations that violate the largest number of constraints and by the highest amount are eliminated first (highly infeasible).

In this application this algorithm is used for reducing the design space by considering only the radiated noise as the main objective of interest. The reduction in the mass is not pursued in order to evaluate how diverse the solution set will be. The constraints associated with failure of the structure as strictly enforced (i.e. any configurations that do not meet the conditions in Equations (2) through (9), are completely eliminated from group B). This is done because structural failure of a pressure hull is catastrophic, in contrast with the resource constraint (Equation 1) that limits the amount of structural damping applied to the hull. No catastrophic event will be encountered if the latter is a flexible performance limit.

**Results from Design Simulations**

The results for the performance of the optimal configuration identified by the concurrent single point design are summarized in Table 1. Each column corresponds, respectively, to the configuration that radiates the lowest amount of noise, the configuration that exhibits the lowest mass (these two designs are identified by the lower level discipline optimizations, Equation (10)), and the concurrent optimum which is identified by the upper level optimization (Equation 11).

A main difference between the concurrent optimization algorithm used in this work and any multi-objective optimization approach is that the concurrent optimization identifies first the optimal configurations of each discipline and then uses the information for determining the concurrent optimal design at the upper level. Multi-objective algorithms combine the performance for each discipline in a single objective and used the combined performance metric for guiding the design. The concurrent algorithm uses the Euclidean distance between the Utopia point (determined by the optimal configurations of the individual disciplines) and the non-dominated design alternatives. The design with the smallest Euclidean distance to the Utopia point comprises the best balanced solution identified by the concurrent optimization algorithm.

All three designs summarized in Table 1 meet the structural integrity requirements (Equations 2 - 9) while the configuration that emits the lowest noise and the balanced optimum also meets the damping resource constraint (Equation 1). When evaluating the noise radiated from the configuration that exhibits the smallest mass, the damping values from the configuration of the optimal design for acoustics are used. The hull with the smallest mass exhibits poor acoustic performance (120% worse than the optimal design for acoustics). In order to make this configuration have similar acoustic performance with the optimum from the concurrent optimization excessive structural damping (400% higher than the damping upper limit) must be applied on the entire hull. This situation would correspond to the case where a hull is optimized first for structural concerns (i.e. minimize the mass subject to strength constraints) and then try to improve the acoustic performance through the application of damping. The hull that radiates the least noise exhibits 65% higher mass compared to the optimal design from the structural optimization. The concurrent optimal represents a well-balanced design that sacrifices performance only 14% with respect to the best possible acoustic performance and 20% with respect to the best possible minimum mass. This analysis demonstrates the value of the concurrent approach for balancing multiple performance objectives and considering the acoustic performance when configuring the structure. The structural configuration impacts the amount of vibrational power that enters a system and how the power is distributed within the system. Therefore it is preferable to consider structural and noise radiation objectives concurrently.

Next, the results from the design space reduction based on SBD principles are presented and discussed. One thousand design configurations are selected using a Latin Hypercube selection process within the bounds of the design variables (Figure 3). Out of them 517 configurations did not violate the structural constraints and were considered further in

| TABLE 1 Performance summary of optimal hull designs |
|----------------------------------|------------------|------------------|------------------|
| Acoustic optimum                 | Structural optimum | Concurrent optimum |
| Radiated power [W*Hz]            | 2.27E-3           | 5.01E-3           | 2.41E-3          |
| Mass [kg]                         | 4.00E+5           | 2.91E+5           | 2.91E+5          |

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the analysis. Figure 6 presents performance information for all configurations at the start and at the end of the search process. The horizontal axis represents the radiated noise and the vertical axis the value of the damping resource. Each point within the plot represents one hull configuration. The points are color coded based on the associated value of the mass.

From the remaining group of solutions it can be observed that configurations of varying mass and damping are retained in the last set. Out of all designs included in the last set, four of them marked with “1” through “4” in the plot are selected for further discussion. “4” corresponds to the acoustic optimum, “3” is associated with the configuration that uses the least damping out of all remaining designs. Finally “1” and “2” exhibit low use of damping, have good acoustic performances, but “2” has a moderate amount of mass while “1” has a high level of mass. A parallel axis plot is depicted in Figure 7 for the four selected hull configurations.

The seven design variables are represented in the horizontal axis and the associated numerical values in their respective units are presented in the vertical axis. As a reference, the values of the design variables of a baseline configuration (“Original Design”) are also included in the parallel axis plot. The four configurations are selected as representative of diversity (i.e. different values for the design variables) in the reduced set. The four designs do exhibit the expected variation in the values of their design variables with the exception of the flange thickness of the king frames where all designs converge towards the upper bound of that design variable. This convergence reveals the importance of that variable in reducing the radiated noise. In this notional study the simple model that determines the power into the system uses the thickness of the flange for determining the driving point impedance that the mechanical forces see, and then uses the impedance values for calculating the input power. The relative importance of the design variables in reducing the radiated noise can be identified even easier through the results summarized in Figure 8.

This Figure presents the ranges of the design variables as designs are eliminated though the outlined SBD inspired process. The speed at which the range of a given design variable reaches a more narrow set of values indicates its importance.
In this example, the thickness of the flange of the king frames is by far the most important design variable because it controls the input power into the system (due to the simple model used for converting the force excitation into input power for the EFEA analysis). The damping applied on the “green” section of the hull is the second most important design variable, which makes physical sense, since it controls the power dissipated in the vicinity of the excitation. The SBD inspired design space reduction process used in this work shows how alternative configurations that represent similar performances can be identified and how tracking the speed of convergence of the design variables during the elimination process allows determining the relative importance of the design variables.

Summary/Conclusions

This work demonstrates that the EFEA method enables considering the structural-acoustic performance of a notional submarine pressure hull as part of a representative early stage design effort. A design space reduction method based on principles of SBD allows identifying which design variables are important and which options of alternative design configurations can provide nearly optimum performance. The concurrent single point design analysis allows identifying a configuration that balances all performance objectives in the best possible manner. The results reveal the importance of considering concurrently the structural design and the acoustic performance of a Naval system since the structural arrangements also impact the vibrational power entering the system, the power transfer through the system, and the radiated noise.

References


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