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Design Minimization of Noise in Stiffened Cylinders Due to Tonal External Excitation

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A computational design tool was developed to perform a constrained optimization of the acoustic environment within a vibrating cylinder, incorporating finite element and boundary element methods. The tool comprises a UNIX shell script that coordinates an iterative design optimization process integrating a number of programs, the key components of which are MSC/NASTRAN for structural analyses, COMET/Acoustics for acoustic analyses, and CONMIN for nonlinear optimization. In addition to the structure and implementation of the tool, this paper presents the results of a number of trials of the tool applied to stiffened and unstiffened cylinders, considering different formulations of the objective function to be optimized, and for a constant frequency exterior monopole excitation. Models were constructed to investigate longitudinal vs circumferential variations in design properties as well. The results indicate that shell thickness variations tend to dominate interior acoustic response, as compared with stiffener variations. The results further indicate that longitudinal variation is more effective than circumferential variation. Effective longitudinal design variations include shell thickening toward the cylinder midplane or, equally effective, thinning toward the midplane. Effective circumferential designs exhibited periodic variation around the circumference.

Nomenclature

d = cylinder diameter
$f(\boldsymbol{x})$ = objective function
f_0 = initial value of objective function
h_f = frame height
$h_{f,\text{ref}}$ = reference frame height
h_s = stringer height
$h_{s,\text{ref}}$ = reference stringer height
l = cylinder length
$\hat{p}_i(\boldsymbol{x})$ = complex acoustic pressure at node i
t = shell thickness
t_{ref} = reference shell thickness
W = total structural weight
$W_{\max}$ = constrained maximum weight
W_{ref} = weight of the reference design cylinder
W_0 = initial structural weight
$\boldsymbol{x}$ = design variable vector
$\boldsymbol{x}_L, \boldsymbol{x}_U$ = vectors of lower and upper bounds
β_a = auxiliary acoustic design variable
β_w = auxiliary weight design variable
μ_a = auxiliary acoustic design variable weighting
μ_w = auxiliary weight design variable weighting

Introduction

THIS paper presents the end results of four different objective and constraint formulations, implemented within a computational design tool, for the optimization of stiffened and unstiffened cylinder models. The desired goals of the optimizations are related to the structural weight and interior noise environment of the cylinder models. The cylinders are considered to be excited at a single frequency by an external noise source, modeled here as a monopole. For the unstiffened model, the cylinder wall thicknesses are considered to be the design variables. For the stiffened model, shell thickness and stiffener heights are considered as design variables. The cylinders considered are not representative of aircraft structures, but they are suitable for assessing trends in optimized cylinders considering both weight and interior noise criteria, which is the key focus of this paper.

There has recently been increasing interest among airlines and aircraft manufacturers in controlling the acoustic environment within aircraft cabins to provide passengers with a more comfortable environment, and to provide the flight crew with a flight-deck environment that does not impair communication. In turboprop aircraft, the interior noise is dominated by noise caused by the propeller blades. A propeller produces a highly tonal and highly directional sound field that is dependent on such parameters as blade thickness, steady thrust, and blade tip speed. The principal noise components are associated with blade passage, and are therefore deterministic.[1] No current certification requirements exist to regulate interior noise, but airlines do require noise-level guarantees from the manufacturer.[2] Researchers and manufacturers have confronted this problem by two broad means: active and passive noise control.[3–5] Classical active noise control involves the placement of loudspeakers and microphones throughout the passenger cabin or flight deck in conjunction with a control system. Active structural acoustic control involves the placement of force or moment actuators at points on the structure, again in conjunction with a control system. Active noise control is an appealing technique because it can be fitted to existing airframes without major structural modification. Passive noise control, by contrast, seeks to reduce cabin noise by the modification of the airframe structure itself. For example, this could be in the form of acoustic treatments added to the cabin walls.

It is desirable that future aircraft be designed to have quieter passenger cabins, minimizing the need for supplementary active or passive noise control systems. Designing quiet structures is itself

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a form of passive noise control, and falls under the broad category of multidisciplinary design optimization (MDO).[6] The objective, constraint, and goal functions in an MDO problem model the often conflicting goals of a variety of disciplines. The design of quiet aircraft cabins is multidisciplinary in that it requires satisfaction of not only acoustic goals and constraints, but also structural goals and constraints such as weight, stress, and aeroelastic effects. This makes the multidisciplinary design process extremely complicated, requiring many iterations of time-consuming analyses coupled with a mathematical programming algorithm as a means of optimizing the system. In addition, due to the complexity and often high nonlinearity of MDO problems, it is nearly impossible to guarantee that a solution is truly optimal. As an example, consider an optimization of a stiffener's dimensions with the objective of minimizing weight while maximizing stiffness and obtaining certain specific modal frequencies. While the weight will be linear with respect to the dimensions, the stiffness and modal properties are nonlinear functions of the dimensions. It remains the task of the designer to choose from among a set of candidate solutions the one that not only best satisfies the goals and constraints defined in the MDO problem, but also meets more intangible standards such as design experience and intuition. Although MDO may be difficult to implement, it offers a number of far-reaching benefits to the designer: 1) goal-orientation of the design approach; 2) tedium reduction; 3) decision-making help, trend finding, and preliminary design guidance.[7]

Some examples of the use of structural acoustic optimization have appeared in the literature.[8–10] The methods used were all computational because automation of design optimization was necessary to keep design times down. In the present work, an approach similar to that used in earlier research is used to optimize the design of aircraft fuselage structures to minimize the noise in the interior. This algorithm is implemented through a computational design tool.[11–13] This form of passive noise control involves coupling structural and acoustic analysis packages with an optimization algorithm as part of an automated process. The heart of the design tool is a UNIX shell script that controls the automated iterative analysis and optimization procedure. The user provides a group of input data files describing the structural and acoustic models and the parameters of the optimization procedure, and the design tool returns updated data files describing the optimized sound field and design of the structural model. The purpose of the overall research project is to develop and validate a flexible, robust tool for the optimal design of stiffened or unstiffened cylinders subject to various acoustic and structural objectives and constraints and excited by an external harmonic noise source.

The following section introduces the computational structure of the design tool, and describes the specific models to which the optimization formulations will be applied. Next, the four optimization formulations considered in this paper are introduced. The subsequent section presents the results pertaining to the formulations' performance, followed by a concluding remarks section.

Design Tool Description

The design tool is described in greater detail in earlier papers.[11–13] Its features and capabilities will be briefly reviewed here. The design tool is a UNIX shell script that controls the interaction of several main programs and supporting programs and implements error-trapping capabilities. Figure 1 is a flowchart depicting the structure of the script's algorithm.

The main programs within the design tool are CONMIN, MSC/NASTRAN, and COMET/Acoustics. CONMIN[14] is the actual optimizer. It uses the modified method of feasible directions algorithm to optimize a single objective function subject to inequality constraints and side constraints on the design variables.[15] Structural analyses are performed with MSC/NASTRAN Version 68. All structural models are assumed to have 2% structural damping. The models are presumed to be excited by external acoustics loads. The external acoustic loads are computed in advance using the boundary element code COMET/Acoustics, with the pressures applied as harmonic, single-frequency loads at the structural element nodes. COMET/Acoustics is used within the design tool to perform the

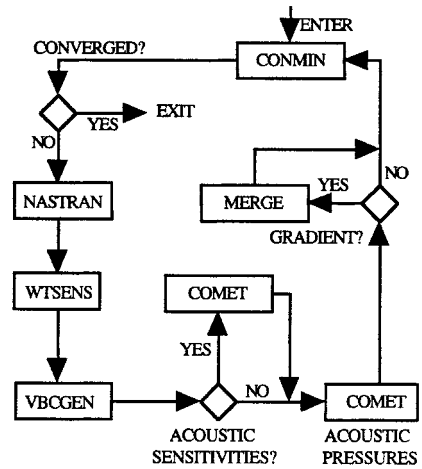

Fig. 1 Schematic of design tool algorithm logic.

acoustic analyses of the cylinder interior.[16] At present, the acoustic analysis for the cylinder models is uncoupled, meaning that the vibration of the structure is not considered to be affected by the bounding acoustic medium.

WTSENS, VBCGEN, and MERGE are supporting codes for the main programs. WTSENS extracts the total cylinder weight and weight sensitivities from the MSC/NASTRAN output. The weight sensitivities are the change in weight due to a change in a design variable (in the context of this paper, the design variables are shell element thicknesses and/or stiffener properties). VBCGEN translates surface velocities from the MSC/NASTRAN output into a COMET/Acoustics data set as velocity boundary conditions.

MERGE combines MSC/NASTRAN output structural design sensitivities (change in velocity due to change in design variable) and acoustic velocity sensitivities from COMET/Acoustics (changes in acoustic pressures due to changes in surface normal velocity) to produce the acoustic design sensitivity (change in acoustic pressure due to change in design variable). The acoustic sensitivities from COMET/Acoustics need be computed only once for each cylinder model, at the start of the optimization procedure. If the same model is optimized later (but at the same frequency), perhaps with a different objective or constraints, the same sensitivity data files are reused.

The user must prepare several files before invoking the script: a file containing parameters that define convergence criteria and constraint tolerances for CONMIN, desired objective formulation, a NASTRAN data set describing the structural model and the external loading conditions, and input files for COMET/Acoustics. The COMET/Acoustics input files include such information as the interface between the structural and acoustic models, and the locations of points within the model at which to compute acoustic pressures.

Cylinder Models

This work considers an unstiffened and a stiffened cylinder. The unstiffened cylinder is considered to be made of aluminum, with $l/d = 2.18$. The shell thickness at the reference design state has $t_{\text{ref}}/d = 0.0002$. This thickness is used as the reference or normalizing constant for the bounds on thickness, which are set here at $t_{\text{upper}}/t_{\text{ref}} = 2.0$ and $t_{\text{lower}}/t_{\text{ref}} = 0.59$. Both ends of the cylinder are clamped. The stiffened model presumes the same base shell with the addition of angle stringers and channel frames. The physical dimensions of the cylinder model are the same as the outer shell of the model used by Grosveld et al.[4] in their study of active structural acoustic noise control.

Because only symmetric modal response is being investigated at this time, analysis is simplified by modeling only one-quarter of the cylinder (depicted in Fig. 2). The base NASTRAN model for the unstiffened cylinder is composed of 800 linear quadrilateral elements. To study the influence of longitudinal variations in the design variables, i.e., shell element thickness and/or stiffener

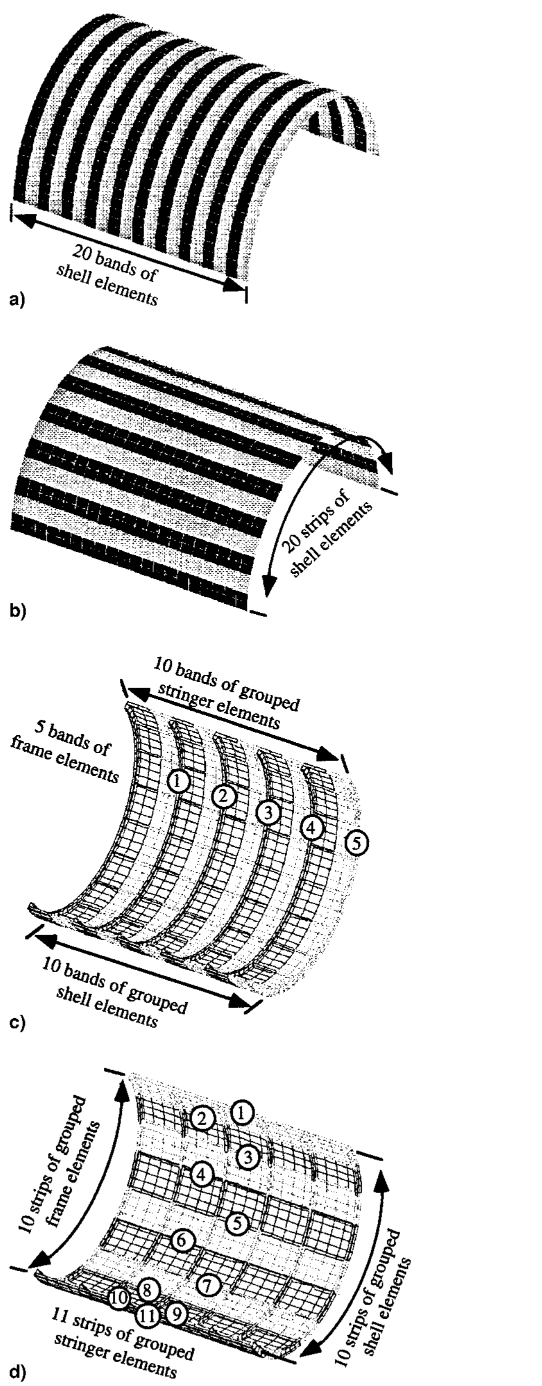

Fig. 2 a) Unstiffened circumferential, b) unstiffened longitudinal, c) stiffened circumferential, and d) stiffened longitudinal models.

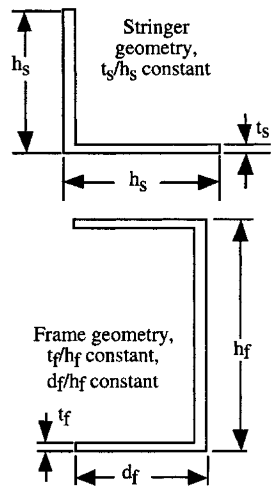

Fig. 3 Stiffener geometry.

dimensions, the shell elements were grouped into 20 circumferential bands of 40 elements each. This model is referred to as the circumferential model. To study the influence of circumferential variations in the design variables, the shell elements were grouped into 20 longitudinal bands of 40 elements each. This model is referred to as the longitudinal model. Each of these models has 20 design variables, representing the thickness of the elements in each of the 20 groups.

The MSC/NASTRAN model for the stiffened cylinder model is somewhat more complex. For the longitudinal variation, there are 10 shell groups, 10 stringer groups, and 5 frame groups, for a total of 25 design variables. For the circumferential variation, there are 10 shell groups, 11 stringer groups, and 10 frame groups, for a total of 31 design variables. Figure 3 depicts the chosen stiffener geometry. Note that these are low-torsional stiffness sections. Therefore, the torsional stiffness is held constant at its initial value for the given model throughout the optimization process.

The design variables for the shell groups are shell thicknesses. For the frame groups, the design variables are frame heights. For the stringer groups, the design variables are stringer heights. Design equations were integrated into the MSC/NASTRAN model to hold the separate aspect ratios (width/height) of the frame and stringer sections constant, as well as holding the ratios of height to thickness constant. The shell variables in the stiffened models have the same bounds as for the unstiffened model. With respect to the reference design, the bounds on the stringer heights are $h_{s,\text{upper}}/h_{s,\text{ref}} = 1.57$ and $h_{s,\text{lower}}/h_{s,\text{ref}} = 0.67$. Bounds on the frame heights are $h_{f,\text{upper}}/h_{f,\text{ref}} = 1.38$ and $h_{f,\text{lower}}/h_{f,\text{ref}} = 0.59$.

The cylinder is assumed to be excited by a single exterior monopole source at dimensionless wave numbers of $kd = 4.73$ for the unstiffened model and $kd = 4.23$ for the stiffened model. This forcing excites a (2, 1) structural resonance in the cylinder with design variables at reference conditions. The source is located $0.1d$ to one side of the cylinder at $l/2$. The acoustic loads produced by this source on the cylinder are computed beforehand by COMET/Acoustics and included in the MSC/NASTRAN data set. For both the unstiffened and stiffened cylinders, two additional starting design models were considered. These other models are initially not on resonance. In all instances the effectiveness of the optimization was determined based on the ending noise level, not on the reduction from the starting point. Therefore, the fact that the reference models start on resonance is not significant.

Optimization Problem Formulations

The following briefly presents the mathematical formulation for each of the objective function/constraint options implemented in the design tool. An assessment of these formulations' relative performance may be found in our earlier papers.[11–13]

Baseline Formulation

The baseline formulation minimizes the sum of the squares of the acoustic pressure amplitudes at discrete data recovery points within the volume, whereas side constraints bound the design variables:

Minimize:

$$f(\boldsymbol{x}) = \sum_{i=1}^{\text{NDRN}} \hat{p}_i(\boldsymbol{x})\hat{p}_i^*(\boldsymbol{x}) \Big/ f_0 \tag{1}$$

Subject to:

$$\boldsymbol{x}_L \le \boldsymbol{x} \le \boldsymbol{x}_U \tag{2}$$

In Eq. (1), the asterisk indicates complex conjugation, and NDRN is the number of data recovery nodes within the volume of interest (COMET/Acoustics yields pressure values only at discrete points, the data recovery nodes, within the model).

Acoustic Formulation

The acoustic formulation minimizes the sum of the squared acoustic pressures subject to design variable bounds and to a constraint on the total weight:

Minimize:

$$f(\boldsymbol{x}) = \sum_{i=1}^{\mathrm{NDRN}} \hat{p}_i(\boldsymbol{x})\hat{p}_i^*(\boldsymbol{x}) \Big/ f_0 \tag{3}$$

Subject to:

$$(W/W_{\max}) - 1 \le 0 \tag{4}$$

$$\boldsymbol{x}_L \le \boldsymbol{x} \le \boldsymbol{x}_U \tag{5}$$

Weight Formulation

The weight formulation minimizes the weight of the structure subject to design variable bounds and a constraint on the maximum value of the sum of the squared pressures:

Minimize:

$$f(\boldsymbol{x}) = W/W_0 \tag{6}$$

Subject to:

$$\left[\sum_{i=1}^{\mathrm{NDRN}} \hat{p}_i(\boldsymbol{x})\hat{p}_i^*(\boldsymbol{x}) \Big/ \left(\sum |\hat{p}|^2\right)_{\max}\right] - 1 \le 0 \tag{7}$$

$$\boldsymbol{x}_L \le \boldsymbol{x} \le \boldsymbol{x}_U \tag{8}$$

In Eq. (7), $(\Sigma|\hat{p}|^2)_{\max}$ is the constrained maximum sum of the squared pressures. In the implementation of Eq. (7), the user specifies a desired decibel level for this maximum, and the algorithm converts it to an appropriate sum of squared pressure magnitudes.

Compound Formulation

The compound formulation simultaneously minimizes the sum of the squared acoustic pressures and the structural weight. The objectives, the pressure sum and the weight, are normalized by their initial values and are required to be less than bounds β_a and β_w, respectively. These bounds are considered to be two additional design variables, and their weighted sum is taken as the new objective function, where μ_a and μ_w are the respective weighting factors. The weighting factors allow one or the other objective to be considered more important. The mathematical statement of the formulation is:

Minimize:

$$\mu_a \beta_a + \mu_w \beta_w \tag{9}$$

Subject to:

$$\left[\sum_{i=1}^{\mathrm{NDRN}} \hat{p}_i(\boldsymbol{x})\hat{p}_i^*(\boldsymbol{x}) \Big/ \left(\sum |\hat{p}|^2\right)_0\right] - \beta_a \le 0 \tag{10}$$

$$W/W_0 - \beta_w \le 0 \tag{11}$$

$$\boldsymbol{x}_L \le \boldsymbol{x} \le \boldsymbol{x}_U \tag{12}$$

where $\Sigma|\hat{p}|^2)_0$ is the initial value of the sum of the squared pressures (before the optimization process begins).

Results

As compared with the authors' earlier work in this subject area,[11–13] the focus of the present work is on the optimal design states and objective functions for the more complex models. Multiple initial designs are needed because there is no guarantee that a single global optimal solution exists for each set of objectives and constraints. The existence of multiple local minima, each representing a local optimum, is a distinct probability. In practice, one chooses a suitable number of starting points, so as to adequately sample the design space, and then examines the results for each to select the best optimum. For the work presented here, three starting designs were chosen for each of the unstiffened and stiffened models. The three initial states correspond 1) to all design variables at their upper bounds, 2) all design variables at their lower bounds, and 3) all design variables at an approximate middle point between the bounds. Tables 1 and 2 summarize these three initial design states for both models. Note that the data presented in the following sections were normalized with respect to the design states and weights for the middle starting point, such that this state will be referred to as the reference design state. In the remainder of this paper, the starting design states will be referred to as the upper, reference, and lower cases, corresponding to all design variables at the upper bound, intermediate state, and lower bound, respectively. Each objective function was applied to each of the three initial design states, for each of the different design variable groupings.

Table 1 Initial designs and starting interior relative SPL, unstiffened model

Initial state	Shell thickness, t/t_{ref}	Cylinder weight, W/W_{ref}	Relative SPL, dB
Upper	2.00	2.00	−10.6
Middle	1.00	1.00	−4.6
Lower	0.59	0.59	0.0

Table 2 Initial designs and starting interior relative SPL, stiffened model

Initial state	Shell thickness, t/t_{ref}	Frame height, $h/h_{f,\mathrm{ref}}$	Stringer height, $h/h_{s,\mathrm{ref}}$	Cylinder weight, W/W_{ref}	Relative SPL, dB
Upper	2.00	1.38	1.57	1.90	−29.7
Middle	1.00	1.00	1.00	1.00	−10.3
Lower	0.59	0.59	0.67	0.62	−26.8

The relative average sound pressure level (SPL) within the cylinder is used as the figure of merit in determining the impact of the optimization process. The lower this level, the quieter the interior. The relative average SPL is determined by subtracting a reference level from the absolute level for a given design. The justification for this approach is that all of the models were subject to the same constant frequency exterior monopole excitation with fixed but arbitrary strength. Because the source strength is arbitrary, the absolute interior levels are not significant, though differences in level with respect to a reference will be significant. Therefore, the absolute interior average SPL for the unstiffened model's lower design state was used as a common 0-dB reference and applied to all other unstiffened and stiffened designs. The initial levels in Tables 1 and 2 reflect this choice of reference. The relative interior level prior to optimization is also determined, and the difference between the optimized and unoptimized levels is used as a further indicator of the impact of the optimized design.

Note that the time required to complete an optimization is dependent on the complexity of the model and the performance of the objective function within the optimization algorithm. Reference 13 presents relative performance measures of the objective formulations used for the work at hand. The bulk of the computational expense is in the modeling, not in the optimizer itself.

The following results for the unstiffened and stiffened models are a subset of the number of analyses we have performed. They illustrate the variations in the ending design states for different starting states, as well as the variations in the optimal design states generated through the different objective functions.

Unstiffened Model

Figure 4a presents the relative optimal SPL levels obtained with the circumferential model, and similarly with Fig. 4b for the longitudinal model, for the various cases analyzed here. The letters next

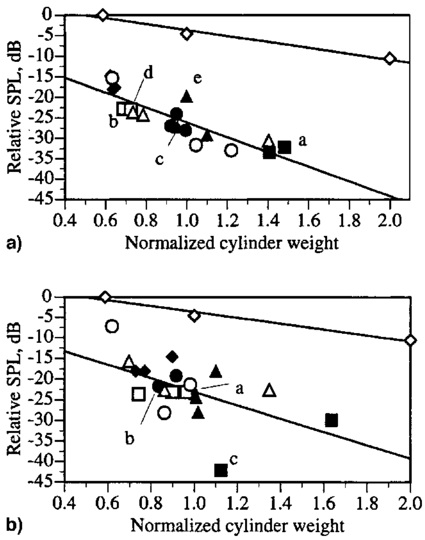

Fig. 4 Optimal relative SPL vs normalized weight for a) circumferential and b) longitudinal unstiffened models: ■, baseline; ●, acoustic with weight <W_{ref}; ▲, acoustic with weight <$1.1W_0$; ♦, weight with SPL <− 15 dB; □, weight with SPL <− 20 dB; ○, compound $\mu_a/\mu_w = 2$; △, compound $\mu_a/\mu_w = 20$; and ◇, nonoptimized uniform cylinders. Letters by data points on (a) identify subfigures in Fig. 5 for the corresponding design. Letters by data points on (b) identify subfigures in Fig. 6 for the corresponding design.

to individual data points are to assist the reader in associating selected design states, depicted in later figures, with their performance. Figures 4a and 4b also include data for uniform thickness cylinder models of different weights, as a means of providing a reference between optimized and unoptimized cylinders. Linear curve fits have been applied to the data, yielding the lines on the figures. The upper line in both figures is for the data from the unoptimized models, and because all of the data from the optimized models are below this line, the optimization indicates that nonuniform shell thickness can provide reduced levels of interior noise. Another observation that may be made is that, within each objective function and constraint combination, there is no single global optimum. Indeed, all 20 of the analyses represented in Fig. 4a yielded different ending design states, as did the 20 analyses in Fig. 4b.

Considering Fig. 4a for the circumferential variation, at the lighter weights, the interior noise level difference between the optimized and nonoptimized models is on the order of 15 dB. Figure 4b indicates that there is greater scatter in the longitudinal results, and that at the lighter weights the noise level difference is on the order of 10 dB. Both figures indicate that the difference increases with increasing weight.

Figures 5a–5e present the beginning and ending design state for a number of the circumferential model analyses. The design state (thicknesses) has been normalized by the thickness of the reference starting design state. The dashed line in each of the figures indicates the normalized initial design state prior to optimization. The relative acoustic performance for each of these states may be determined by associating their subfigure letter designation with the corresponding letter in Fig. 4a. This model has its design variables grouped circumferentially, such that all elements in a circumferential band have the same thickness. Therefore, the thickness may vary only longitudinally in this model (see Fig. 2a). The 20 analyses conducted on this model yielded several alternative trends in the optimal design states. These alternative trends were for increasing thickness toward the plane of the source, decreasing thickness toward the plane of the source, and a more complex trend of decreasing thickness toward an intermediate point followed by increasing thickness toward the plane of the source.

The design state in Fig. 5a illustrates the decreasing shell thickness trend. This state was produced by the baseline formulation. Figure 5b is a variant of this trend, produced by the weight objective with a −20-dB acoustic constraint. The design state in Fig. 5c illustrates the trend for increasing thickness toward the plane of the source, and was produced by the acoustic formulation with a maximum weight constraint of $1.0W_{\mathrm{ref}}$. Figure 5d illustrates a variant of this trend, produced by the weight objective with a −20-dB acoustic constraint. Figure 5e illustrates the decreasing/increasing trend. This state was produced by the acoustic objective with a constraint on the weight of $1.1W_{\mathrm{ref}}$.

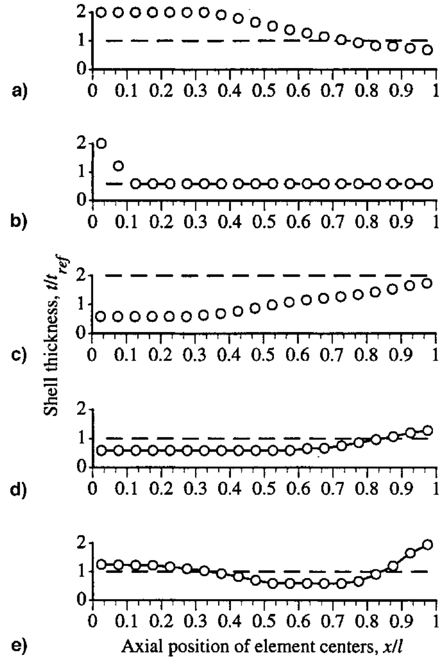

Fig. 5 Optimized and initial shell thickness, unstiffened circumferential model: ○, optimized value, and – – –, initial value. a) reference starting design, baseline formulation; b) lower starting design, acoustic formulation with weight <$1.1W_0$; c) reference starting design, baseline formulation; d) reference starting design, weight formulation with SPL <− 20 dB; and e) reference starting design, acoustic formulation with weight <$1.1W_0$.

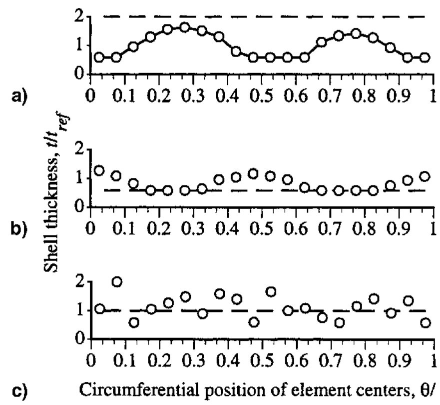

Fig. 6 Optimized and initial shell thickness, unstiffened longitudinal model: ○, optimized value, and – – –, initial value. a) upper starting design, acoustic formulation with weight <$1.1W_0$; b) lower starting design, acoustic formulation with weight <$1.1W_0$; and c) reference starting design, baseline formulation.

Figures 6a–6c present the beginning and ending design states for a number of the longitudinal model analyses. The relative acoustic performance for each of these states may be determined by associating their subfigure letter designation with the corresponding letter in Fig. 4b. This model has its design variables grouped longitudinally, such that all elements in a longitudinal band have the same thickness. Therefore, the thickness may vary only circumferentially in this model (see Fig. 2b). The 20 analyses conducted on this model yielded several alternative trends in the optimal design states. These trends were for a periodic variation in thickness around the circumference with maxima at 45 and 135 deg; a periodic variation in thickness around the circumference with maxima at 0, 90, and 180 deg; and a state that at best could be termed chaotic. The design state in Fig. 6a illustrates the first periodic trend. This state was generated by the acoustic formulation with a constraint on weight of $1.0W_{\text{ref}}$. The design state in Fig. 6b illustrates the second periodic trend, and was also generated by the acoustic formulation with a constraint on weight of $1.0W_{\text{ref}}$ (note the different starting states). The design state in Fig. 6c illustrates the chaotic state, of which 6 of the 20 analyses could be classed. The state in Fig. 6c was produced by the baseline formulation.

Stiffened Model

Figure 7a presents the relative optimal SPL levels obtained from the stiffened circumferential model, and similarly with Fig. 7b for the stiffened longitudinal model, for the various cases analyzed here. The numbers next to individual data points are to assist the reader in associating selected design states, depicted in later figures, with their acoustic performance. Figures 7a and 7b also include the relative levels obtained for each of the three initial design states, as a means of providing a reference between optimized and unoptimized cylinders. Note that the results for these stiffened models do not exhibit the simple relationship to total model weight that the unstiffened models did (this may also be observed in Table 2, where the lower design state has a lower initial level than the middle state). This is a consequence of the presence of stiffeners leading to a dramatically different response from mass-law-like behavior, as exhibited by the uniform unstiffened cylinders. As with the unstiffened model, there appears to be no single global optimum. Note that the symbols for a number of analyses overlay each other in Figs. 7a and 7b, indicating that formulations converged to clusters of similar designs.

Figures 8–10 present the beginning and ending design states for a number of the stiffened circumferential model analyses. The relative acoustic performance for each of these states may be determined by associating their subfigure letter designation with the corresponding letter in Fig. 7a. This model has its design variables grouped circumferentially, such that in a circumferential band all shell elements have the same thickness, all frame elements have the same height, and all stringer elements have the same height. Therefore, these design variables may vary only longitudinally (see Fig. 2c). Because there are three different types of design variables (thickness, frame

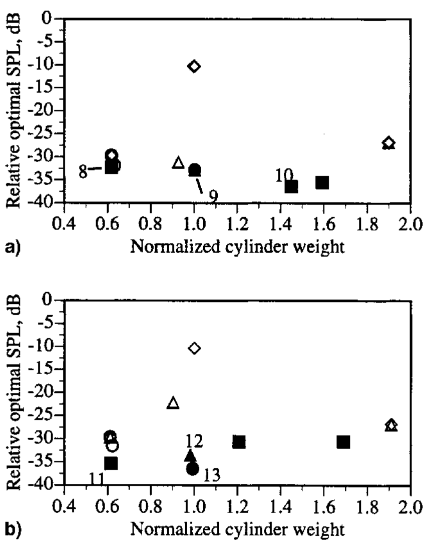

Fig. 7 Optimal relative SPL vs normalized weight for circumferential a) and longitudinal b) stiffened models: ■, baseline; ●, acoustic with weight $<W_{\text{ref}}$; ▲, acoustic with weight $<1.1W_{\text{ref}}$; ○, compound $\mu_a/\mu_w = 2$; △, compound $\mu_a/\mu_w = 20$; and ◇, nonoptimized. Numbers by data points identify figure numbers for corresponding design.

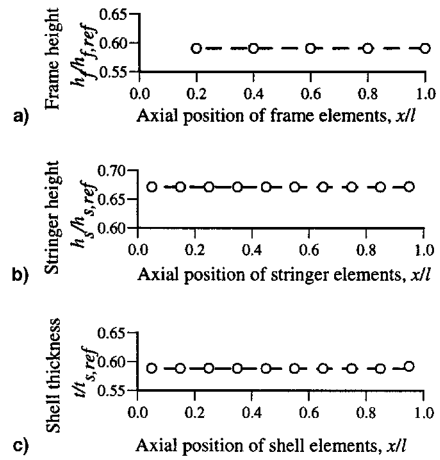

Fig. 8 Optimized and initial heights and thickness, stiffened circumferential model, baseline formulation applied to lower starting design state: a) frame, b) stringer, and c) shell elements: ○, optimized value, and - - -, initial value.

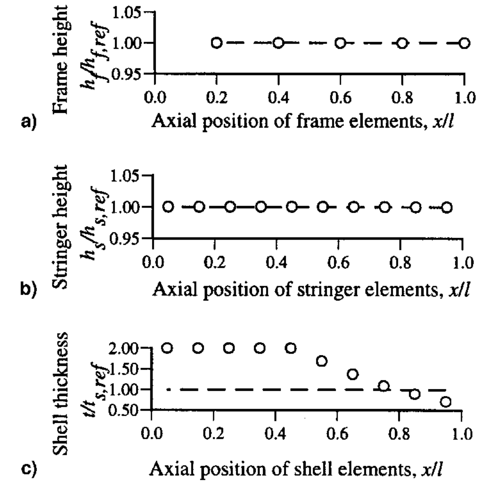

Fig. 9 Optimized and initial heights and thickness, stiffened circumferential model, acoustic formulation with weight constraint $<1.1W_{\text{ref}}$, applied to reference starting design state: a) frame, b) stringer, and c) shell elements: ○, optimized value, and - - -, initial value.

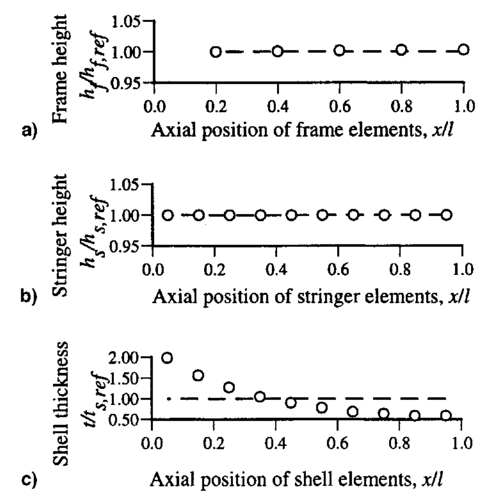

Fig. 10 Optimized and initial heights and thickness, stiffened circumferential model, baseline formulation applied to reference starting design state: a) frame, b) stringer, and c) shell elements: ○, optimized value, and - - -, initial value.

height, and stringer height), the design states are presented as three subfigures. As before, the design variables have been normalized by the values for the reference design state.

Note in Figs. 8–10 that it appears that only the shell design variables are changed in the optimization. Close examination of the stiffener design variables reveals some very minor changes compared with their starting states, such that their contribution to any noise reduction is questionable. All of these models exhibited either a flat thickness, as in Fig. 8, or a decreasing thickness trend as in Figs. 9 and 10.

Figures 11–13 present the beginning and ending design state for a number of the stiffened longitudinal model analyses. The relative acoustic performance for each of these states may be determined by associating their subfigure letter designation with the corresponding letter in Fig. 7b. This model has its design variables grouped longitudinally, such that in a longitudinal band all shell elements have the same thickness, all frame elements have the same height, and all stringer elements have the same height. Therefore, these design variables may vary only circumferentially in this model (see Fig. 2d). Because there are three different types of design variables (thickness, frame height, and stringer height), the design states are presented as three subfigures. As before, the design variables have been normalized by the values for the reference state.

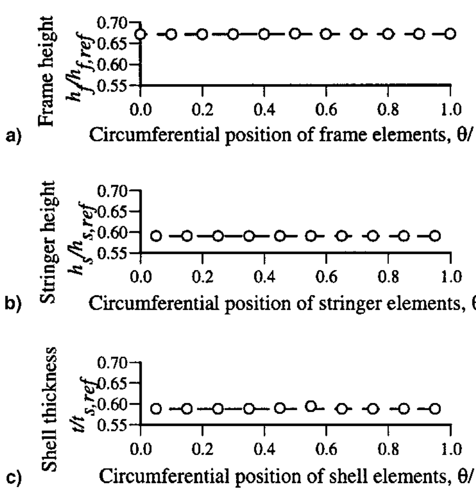

Fig. 11 Optimized and initial heights and thickness, stiffened longitudinal model, baseline formulation applied to lower starting design state: a) frame, b) stringer, and c) shell elements: ○, optimized value, and – – –, initial value.

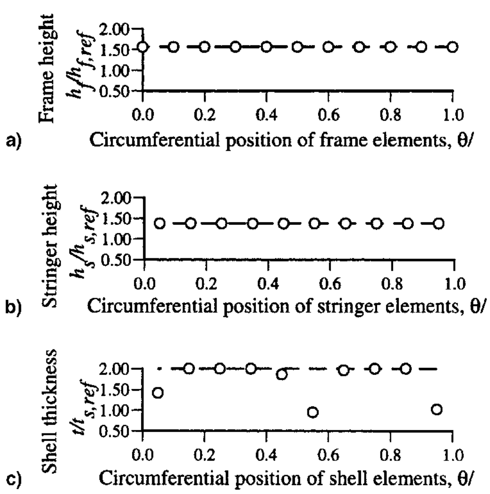

Fig. 12 Optimized and initial heights and thickness, stiffened longitudinal model, acoustic formulation with weight constraint <$1.1W_{\mathrm{ref}}$, applied to upper starting design state: a) frame, b) stringer, and c) shell elements: ○, optimized value, and – – –, initial value.

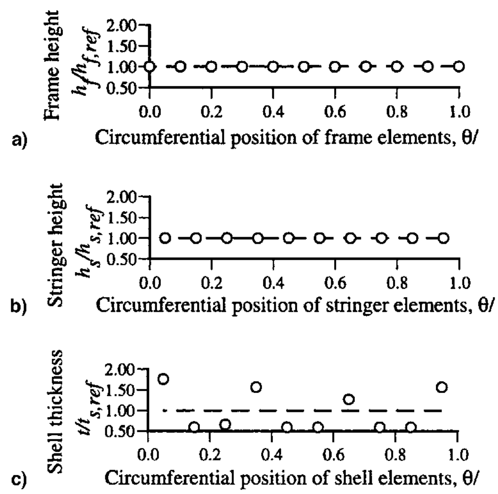

Fig. 13 Optimized and initial heights and thickness, stiffened longitudinal model, acoustic formulation with weight constraint <W_{ref}, applied to upper starting design state: a) frame, b) stringer, and c) shell elements: ○, optimized value, and – – –, initial value.

As with the circumferential model, it appears that only the shell design variables are changed in the optimization. Again, close examination of the stiffener design variables reveals some very minor changes as compared with their starting states, such that their contribution to any noise reduction is questionable. Figure 11 reflects a case where only slight changes occurred in all design variables, including the shell variables. Figures 12 and 13 reflect the periodic variation observed in the unstiffened model.

Concluding Remarks

The results indicate that there is significant potential for reducing tonal interior noise levels inside simple unstiffened and stiffened cylinders. For the cylinder models, excitation, and frequency conditions considered here, we found that longitudinal variability yields a somewhat greater impact on interior noise levels than circumferential variability. When both shell thickness and stiffener parameters were considered as design variables, the shell thickness variation dominated the impact on the noise reduction. Further work is in progress to determine the role of stiffeners with regard to interior noise control.

The results indicate that it is possible to reduce the interior noise levels within stiffened cylinders by varying the skin thickness and/or stiffener geometry. For aircraft application the feasibility of this approach depends on such factors as the fabrication details and structural complexity. Further, the analyses presented here were for single excitation frequency. Work is in progress to extend the design tool's capability to include multifrequency capability.

Further development work is in progress to assess the performance of the optimization considering multispectral excitation, more sophisticated structural optimization, and sensitivity to off-design excitation.

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References

[1]Magliozzi, B., Hanson, D. B., and Amiet, R. K., "Propeller and Propfan Noise," *Aeroacoustics of Flight Vehicles, Theory and Practice, Volume 1: Noise Sources*, Acoustical Society of America, Woodbury, NY, 1995.

[2]Powell, C. A., and Fields, J. M., "Human Response to Aircraft Noise," *Aeroacoustics of Flight Vehicles, Theory and Practice, Volume 2: Noise Control*, Acoustical Society of America, Woodbury, NY, 1995.

[3]Mixson, J. S., and Wilby, J. F., "Interior Noise," *Aeroacoustics of Flight Vehicles, Theory and Practice, Volume 2: Noise Control*, Acoustical Society of America, Woodbury, NY, 1995.

[4]Grosveld, F. W., Coats, T. J., Lester, H. C., and Silcox, R. J., "A Numerical Study of Active Structural Acoustic Control in a Stiffened, Double Wall Cylinder," *Proceedings of NOISE-CON 94, National Conference on Noise Control Engineering*, Noise Control Foundation, Poughkeepsie, NY, 1994, pp. 403–408.

[5]Elliot, S., Nelson, P., Strothers, I., and Bocher, C., "In-Flight Experiments on the Active Control of Propeller-Induced Cabin Noise," AIAA Paper 89-1047, April 1989.

[6]Padula, S. L., "Progress in Multidisciplinary Design Optimization at NASA Langley," NASA TM 107754, July 1993.

[7]Cohn, M. Z., "Theory and Practice of Structural Optimization," *Optimization of Large Structural Systems*, Vol. II, Kluwer, Dordrecht, The Netherlands, 1993.

[8]Bråmå, T., "The Structural Optimization System OPTSYS," *Software Systems for Structural Optimization, International Series of Numerical Mathematics*, Vol. 110, Birkhäuser Verlag, Berlin, Germany, 1993.

[9]Müller, G., Tiefenthaler, P., and Imgrund, M., "Design Optimization with the Finite Element Program ANSYS," *Software Systems for Structural Optimization, International Series of Numerical Mathematics*, Vol. 110, Birhäuser Verlag, Berlin, Germany, 1993.

[10]Yang, T. C., and Cheng, C. H., "Integrating and Automating Analysis and Optimization," *Computers and Structures*, Vol. 48, No. 6, 1993, pp. 1083–1106.

[11]Engelstad, S. P., Cunefare, K. A., Crane, S., and Powell, E. A., "Optimization Strategies for Minimum Interior Noise and Weight Using FEM/BEM," *Proceedings of the 1995 International Conference on Noise Control Engineering*, Noise Control Foundation, Poughkeepsie, NY, 1995, pp. 1205–1208.

[12]Cunefare, K. A., Crane, S. P., Engelstad, S. P., and Powell, E. A., "A Tool for Design Minimization of Aircraft Interior Noise," AIAA Paper 96-1702, May 1996.

[13]Crane, S. P., Cunefare, K. A., Engelstad, S. P., and Powell, E. A., "A Comparison of Optimization Formulations for Design Minimization of Aircraft Interior Noise," *Journal of Aircraft*, Vol. 34, No. 2, 1997, pp. 236–243.

[14]Vanderplaats, G. N., and Moses, F., "CONMIN-A FORTRAN Program for Constrained Function Minimization: User's Manual," NASA TM X-62282, Aug. 1973.

[15]Vanderplaats, G. N., and Moses, F., "Structural Optimization by Methods of Feasible Directions," *Computers and Structures*, Vol. 3, No. 4, 1973, pp. 739–755.

[16]*COMET/Acoustic User Document*, Automated Analysis Corp., Ann Arbor, MI, 1995.