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OPTIMAL DESIGN OF RING STIFFENED CYLINDRICAL SHELLS USING MULTI--ETC(U)
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Optimal Design of Ring Stiffened Cylindrical Shells Using Multiple Frame Sizes

by

Michael Pappas
and
Jacob Moradi

JULY 1978
N.J.I.T. Reporting NV-14

New Jersey Institute of Technology
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Newark, New Jersey 07102

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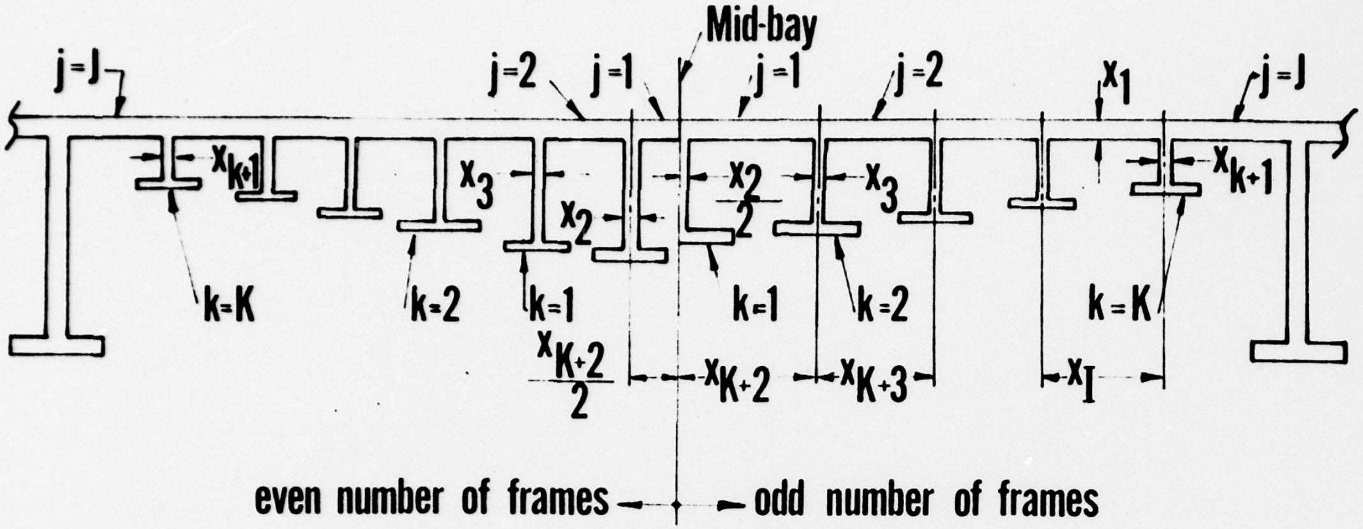
New Jersey Institute of Technology
323 High Street
Newark, New Jersey 07102

INTRODUCTION

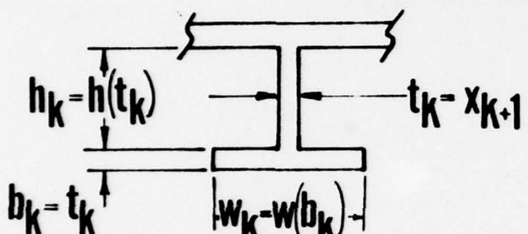
Optimal structural performance either in terms of cost and/or weight is, or should be, the goal of structural designers. Recent developments in optimization theory have now allowed the solution to a substantial class of important optimal problems making possible the achievement of this goal for these cases. The mathematical programming (MP) procedures, originally developed for use in operations research to treat the optimal resource allocation problem, have been applied to the design of submersed, stiffened, cylindrical shells [1-4]. These earlier studies employed uniform stiffening rings or frames of equal size. The problem that was addressed there was that of the most effective allocation of material between stiffeners and the shell assuming all stiffeners were of equal size.

The next resource allocation question that arises is; what is the most effective allocation of material among the stiffeners themselves? For example, if one wishes to suppress a buckling mode with one axial half-wave, and be efficient in the use of stiffener material, one would make the stiffeners largest near the center of the bay segment and smallest near the bulkhead (see Figure 1).

The question of optimal stiffener material allocation has been investigated by Kunoo and Yang [5] for aerospace structures stiffened with both rings and stringers. They obtain about a 5% saving in weight with the use of multiple



a) Typical cross-section showing variable designations for odd and even numbers of frames



b) Dimensions of the k th frame

Figure 1. Hull Variable Designations

rather than equal stiffener sizes for the example they studied. Their computationally demanding doubly reinforced buckling problem, where the stiffeners are treated as discrete, and the relatively ineffective conventional MP procedure they employ [6] require the use of approximation methods to approach a solution in a reasonable period of time (about 2,000 seconds on a CDC 6500).

The reliability of their optimization scheme,seems questionable for two reasons. First, the basic approach they employ produced designs with substantially different weights using two different search schemes. Secondly, no mention is made in their work of the coalescence of buckling modes, a characteristics of optimal designs controlled by buckling behavior, and their procedure apparently does not treat this situation.

This report describes the solution to the simpler singly reinforced discrete stiffener problem by direct optimization without use of approximations like those used in [5]. The optimization formulation and procedure used here admit a large number of simultaneous buckling modes thus allowing optimization under conditions of mode coalescence.

It should be noted that this is a preliminary study, the purposes of which are; to develop and evaluate methodology for the treatment of this problem, to develop preliminary insights into how multiple frame sizes may efficiently be employed in submersible structures, to examine the nature of design improvement resulting from use of multiple size frames, and to investigate the nature of such optimal designs.

PROCEDURE

Mathematical Programming (MP) methods are basically search procedures that iteratively approach the solution to the problem: Find those values $\bar{x}_i$ of the variables x_i that minimize the objective function $f(x_i)$ subject to constraining conditions [7]. The problem is usually stated; Find the $\bar{x}_i$ such that

$$f(\bar{x}_i) = \min [f(x_i)] \tag{1}$$

and such that all functional constraints

$$g_i (\bar{x}_i) \leq 0 \tag{2}$$

and regional constraints

$$x_i^{\ell} < x_i < x_i^{u} \tag{3}$$

are satisfied, where x_i^{ℓ} and x_i^{u} are the upper and lower regional limits respectively.

Variable Designation

The variables employed for this problem are the skin or plating thickness and ring or frame dimensions and spacing. Each frame size used introduces a variable set associated with its dimensions. Thus the number of variables is dependent on the number of sizes employed. To reduce problem dimensionally it is useful to introduce several assumptions.

It will be assumed that the bay is symmetrical with respect to a plane at mid-bay, normal to the cylinder axis. Then, referring to Figure 1, if N_f is the number of frames used in the structure (excluding the deep end frames which are considered rigid simple supports for this problem) then the number of problem variables I may be taken as

$$I = [(N_f + 1)/2]^T N_d + [N_f/2]^T + 1 \qquad (4)$$

where $[\phi]^T$ is ϕ truncated to an integer, and N_d is the number of dimensions of an individual frame treated as variables. Thus a shell problem with four variable quantities per frame utilizing 20 frames would have 51 variables.

In an earlier study of the characteristics of optimal shells [1] it was found that all frame dimensions need not be independent to achieve nearly optimum designs. Thus in order to minimize the number of variables for this preliminary study the frame dimensions will be proportioned as follows (refer to Figure 1): Let x_1 be the plating thickness and let there be K frames with x_{k+1} the thickness of the k^{th} frame, where the frames are numbered from mid-bay outward. Let the web height be dependent on the frame thickness where

$$h_k = h(x_{k+1}) \quad k = 1,2...K \qquad (5)$$

where h_k is a value of web height that will just produce buckling in the web of frame k. For the preliminary study, let $h_k = 18\ x_{k+1}$ for all frames [1]. This assumption apparently carries no weight penalty since it was found in earlier studies that the web buckling constraint is always active in optimal structural designs.

Also let the flange thickness

$$b_k = x_{k+1} \tag{6}$$

and the flange width be that which will just produce buckling of the flange. That is, let the flange width be dependent on the flange thickness by

$$w_k = w(b_k) \tag{7}$$

where for this study $w_k = 12.6\ b_k$ for all k [1].

Such proportioning of the flange does produce slightly heavier designs (less than 2% greater than optimum [1,2]). The use of this simplification is however justified for this preliminary work. Then for this study N_d is unity.

Now let the remaining variables represent the frame spacing counting from the center of the bay outward. These variables are

$$x_i \quad i = K + 2, K + 3, \ldots I$$

<u>Objective Function</u>

The objective funciton for this problem is the weight displacement ratio, W_D, of the hull segment excluding the weight of the deep end frames [1]. Thus

$$W_D = \begin{cases} W/\gamma_w V_D & \text{internal frames} \\ \\ W/[\gamma_w(V_D + V_D)] & \text{external frames} \end{cases} \tag{8}$$

where γ_w is the specific gravity of the immersion fluid, V_D and V_F the volume displaced by the hull plating and frames respectively and the weight of the hull W is

$$W = \gamma_s(V_s + V_F) \tag{9}$$

where γ_s is the specific gravity of the hull material and V_s the volume of the hull plating. In this problem

$$V_F = \begin{cases} 2 \sum_{k=1}^{K} V_k , & N_f \text{ even} \\ V_1 + 2 \sum_{k=2}^{K} V_k , & N_f \text{ odd} \end{cases} \tag{10a}$$

where

$$K = [(N_f + 1)/2]^T \tag{11}$$

and V_k is the volume of frame k.

Constraints

It is assumed that, since the plating thickness is uniform only the smallest frame can be active in yielding. Thus frame yielding is controlled by specifying that

$$g_1 = (\sigma_F - \sigma_a)/\sigma_a \leq 0 \tag{12}$$

where σ_a is the allowable frame stress and σ_F is the hoop stress in the smallest frame. This stress is computed in the following fashion. Find the index c where

$$x_c = \min\ (x_{k+1}) \quad k = 1,2\ldots K \tag{13}$$

Then in the equations [1]

$$\sigma_f = Q\ pR/(A+bt)$$

$$Q = b[1 + (1-\mu/2)\beta/B]/(1+\beta)$$

$$\beta = [2N/(A + bt)][1/3\ (1-\mu^2)]^{1/4}\ (Rt^3)^{1/2}$$
$$N = (\cosh\theta - \cos\theta)/(\sinh\theta + \sin\theta) \quad (14)$$
$$\theta = L[3\ (1-\mu^2)/(Rt^2)]^{1/4}$$
$$B = bt/(A+bt)$$

let

$$t = x_1$$
$$b = x_c$$
$$A = 30.6\ x_c \quad (15)$$
$$L = \max\ (x_{a-1}, x_a),\quad N_F \text{ odd and } c \neq 2$$
$$L = \max\ (x_a, x_{a+1}),\quad N_F \text{ even}$$
$$t = x_a,\quad N_F \text{ odd and } c = 2$$

where

$$a = c + K,$$

p is the hydrostatic pressure and μ is Poisson's ratio.

If plating yield is active, as is often the case, the optimization procedure will try to adjust the hull dimensions so that several panels are simultaneously active in yield. It is therefore necessary to check all panels for yielding. Thus one has a series of constraints

$$g_{j+1} = (\sigma_{pj} - \sigma_{pa})/\sigma_{pa} \leq 0 \quad j = 1,2...J \quad (16)$$

where the number of different panels J is

$$J = K + 1 \quad N_f \text{ even}$$
$$J = K \quad N_f \text{ odd} \quad (17)$$

and where σ_{pa} is allowable plating stress and σ_{pj} is the stress at the center of the j^{th} panel. This stress will be estimated here by averaging.

Thus let

$$\sigma_{pj} = \frac{\sigma^L_{pj} + \sigma^R_{pj}}{2} \tag{18}$$

Equations (9-12) of [1] are used to calculate σ^L_{pj} and σ^R_{pj}.

For these computations the quantities t and L in these equations are replaced by

$$\begin{aligned} t &= x_1 \\ L &= L_j = x_{K+j+1} \end{aligned} \tag{19}$$

For j = 1,2...J-1 in computing σ^L_{pj} the quantities b and A are replaced by

$$b = x_r \begin{cases} r = j & N_f \text{ even} \\ r = j + 1, & N_f \text{ odd} \end{cases} \tag{20a}$$

$$A = 30.6\ b$$

and for σ^R_{pJ}

$$b = x_r \begin{cases} r = j + 1, & N_f \text{ even} \\ r = j + 2, & N_f \text{ odd} \end{cases} \tag{20b}$$

$$A = 30.6\ b$$

except when j = 1 and N_f is even then

$\sigma^L_{p1} = \sigma^R_{p1} = \sigma_{p1}$ and b and A are replaced by

$$\begin{aligned} b &= x_2 \\ A &= 30.6\ b \end{aligned} \tag{20c}$$

For the end panel j=J let $\sigma^R_{pJ} = \sigma^L_{pJ}$ and replace

$$\left.\begin{array}{l} L = L_J \\ \qquad\qquad r = J, \quad N_f \text{ even} \\ b = x_r \\ \qquad\qquad r = J+1, \; N_f \text{ odd} \\ A = 30.6\ b \end{array}\right. \tag{20d}$$

The other constraints used in Ref. [1] will not be used here.

The above formulation may also be used to treat a form of the problem employing equal size stiffeners. In this case I = 2, K = 1, and J = 1 and in the objective function calculation

$$V_F = N_f V_1 \tag{10b}$$

For the plating yield constraint $\sigma^L_{p1} = \sigma^R_{p1}$ and

$$\begin{array}{l} b = x_2 \\ A = 30.6\ b \end{array} \tag{20e}$$

To determine the minimum buckling pressure for the range of parameters of interest in this preliminary study one should examine all mode combinations where n (the number of circumferential waves) ranges from 0 through 20 and m (the number of axial half-waves) from 1 through 40. Since a typical optimization run using even a relatively efficient MP algorithm such as [7] typically requires several hundred sets of functional constraint evaluations it would be extremely costly to utilize the buckling analysis of reference [8,9] employed here, for each buckling constraint evaluation if all modes are examined simultaneously. This would involve the solution of several hundred eigenvalue problems of rank 840 (21x40) during a single optimization run. Such computational effort is impractical and unnecessary in this problem.

It may be seen from an examination of the equations of reference [8] that for the case of uniform stiffeners the buckling modes are uncoupled with respect to n and interact only with respect to even or odd m. Thus the 21 x 40 by 21 x 40 problem can be reduced to a series of forty two, 20 by 20 problems substantially reducing computational effort required. Furthermore, since most constraint function evaluations are for very similar designs, computational effort may be again reduced by further restricting the range of odd or even m terms included in the formulation of the eigenvalue problem for a particular n. This choice is based on a knowledge of the range necessary to include all m terms making a significant contribution. Likewise only those n values which appear to be "critical" with respect to buckling need be examined.

For this study the following procedure is used to determine the buckling behavior of the shell.

Let v^s_{mn} be the value of an element of the matrix of eigenvectors which represents the buckling behavior of the design x^s, and m^s_n be the value of m associated with the largest value of the component v^s_{mn} of vector v^s_n. Now using the procedure of [8] starting from $n = n_{min}$ where n_{min} is the lowest n considered and setting an index i = 1 set up and solve an M by M eigenvalue problem p_n^s for design x^s using terms associated with

$$m = m_{min}, m_{min} + 2, m_{min} + 4, \ldots\ldots, m_{max} \qquad (21)$$

where all m are odd. Here

$$m_{max} = m_{min} + 2\ (M-1) \qquad (22)$$

where M is the number of m terms used for the analysis, And

$$m_{min} = \begin{cases} 1 \quad , & m_n^{s-1} \leq M \\ m_n^{s-1} - M, & m_n^{s-1} \geq M \end{cases} \qquad (23)$$

where m_n^{s-1} is the largest m component of its associated eigenvector for the last design $x^{s=1}$. Now if

$$m_{rn}^s = m_n^{s-1} \qquad (24)$$

or

$$m_n^s < M \qquad (25)$$

it means that for a given number of terms M the range for the above problem was properly placed and thus the problem p_n^s was the "best" problem. On the other hand if one of these conditions is not met a new problem p_n^* is formulated per equations (21-23) where m_n^s replaces m_n^{s-1}. If conditions (24) or (25) are now satisfied where m_r^* replaces m_r^s and m_r^s replaces m_r^{s-1} in these equations then problem p_n^* is the "best" problem. If not, the process is repeated until conditions (24) or (25) are satisfied or oscillation is detected whereupon that problem of the last three solved producing the lowest eigenvalue is taken as the best problem. The m_n^* or m_n^s associated with the best problem which is now called P_n^S is then called m_n^s and used at the next design iteration x^{s+1}.

The first T eigenvalues of λ_{tn}^s t = 1,2....T (which are the collapse pressures) of this problem are used to form r' constraints for this n where m are odd by letting

$$g_r^s = \frac{\lambda_{tn} - F_p}{F_p} \qquad r = J + 1 + T\,(u-1) + t \qquad (26)$$

The index u is then increased by 1. This process is repeated for this n and even m. Constraints are then evaluated in similar fashion for all n to be examined.

It should be noted that the treatment of buckling in this formulation is substantially different than that used in earlier optimization studies using orthotropic shell theory. These earlier studies required only two buckling constraints, one for general and one for shell or panel (interframe) buckling. Here all modes need to be examined and constraints established for all those that may be active. Thus this problem formulation considers a large number of buckling constraints. The earlier formulation was at first attempted in this study. It was found, however, that the search converged a design where more than 2 buckling modes were active. The search terminated at such a design since attempts made to lighten the design while moving to avoid the constraints associated two buckling modes produced a violation in some other mode. This situation is analagous to the frequency separation problem discussed in [10].

Calculation of Buckling Constraint Derivatives

The MP procedures employed here require the use of derivatives to the functions involved. These derivatives are calculated by a simple forward difference method at each point where the direction finding problem (a key element of the procedure) is formulated. This problem is set up each time, a direction change is indicated such as when a new active constraint is encountered [7]. It was found that a number of constraints fluctuated between active and inactive. In this situation the algorithm essentially reduced to a series of moves of fixed step based on the direction indicated by the solution to this problem. The changes in direction under these circumstances were primarily due to changes in the active contraint set rather than as the result of changes in the values of the derivatives. Thus in order to reduce computational effort derivatives were recalculated only after four moves were taken.

To calculate the buckling constraint derivatives at a point x^s for constraints derived from a particular eigenvalue problem P_n^s, a similar problem P_n^{si} is formed using $m_n^{si} = m_n^s$ and solved using $x^{si} = x^s + \Delta x_i$ where Δx_i is a small change in i^{th} coordinate direction of vector x^s. The lowest T eigenvalues of this problem are then used to compute the i^{th} components of the T derivatives associated with the constraints derived from P_n^s where the lowest eigenvalue of problem P_n^{si} is associated with the lowest of P_n^s to estimate the derivative of the lowest eigenvalue. The second lowest eigenvalue of P_n^{si} is associated with the second lowest of P_n^s etc.

If there are L active constraints, one therefore must solve LI eigenvalue problems at each point where derivatives must be calculated. Thus the computation of buckling constraint derivatives requires substantial effort.

RESULTS

A FORTRAN IV computer program was developed using the methodology described above. The 1,000 foot immersion depth study of [1] was repeated here using steel with an allowable hull and frame stress of 90,000 psi. For this study R = 198 in, $\gamma_w = 0.0374$ lb/in^3, $\gamma_s = 0.282$ lb/in^3, $E = 30 \times 10^6$ psi and $\mu = 0.25$. Only configurations using odd numbers of frames were studied. The n modes from $n = 3$ to $n = 16$ were investigated at all design points. A single buckling constraint (T=1) was used for n modes with odd m and $n \leq 5$ with even n. The problem formulations for these cases used seven m terms (M=7). For $n > 5$ with even m terms two constraints (T=2) were generated for each n mode and fifteen m terms were used (M=15). The use of these conditions was based on experience gained during early debugging runs. All optimal designs were however checked for all $0 \leq n \leq 20$. The range $3 \leq n \leq 16$ was found to contain all active buckling constraints.

Three sets of two optimization runs were made using 3, 11 and 19 frames. In the first run, equally spaced equal frame sizes were employed. The optimal equal size frame configuration was then used to start the second run where the frame sizes and spacing were allowed to vary. The results of a given set of runs then allowed a direct comparison between optimal designs using identical and multiple frame sizes. Multiple starting points were used to confirm optimality.

The results are summarized in Table 1. The 19 frame problem required about 3,000 sec CPU time on an IBM 370/165 using the H level compiler. Consider first the hulls reinforced with only three frames. The design using identical frames is, as expected [2], controlled by buckling modes where $m = 1$ and 4, the general and shell buckling modes. In the design using multiple frame sizes, it may be seen that, as expected, the center frame is largest in order to suppress the $m = 1$, mode. However, substantial frames are still needed to suppress a mode where $n = 6$ and $m = 2$ which become active as the frames nearest the bulkheads were reduced in size in an effort to improve frame efficiency.

Plating thickness is controlled in both designs by the $n = 8$, $m = 4$ modes. Thus the two designs have identical plating thicknesses since for this configuration the torsional stiffness of the frames does not significantly effect these modes. Now since the plating represents most of the weight of the hull segment in these designs the small improvement in frame efficiency produces a negligable savings in overall weight (about 0.45%).

Thus it appears there is little to be gained from multiple frame sizes in sparsely stiffened frames.

Table 1. Optimal Hull Designs

		3 FRAMES		11 FRAMES		19 FRAMES	
		Identical	Multiple	Identical	Multiple	Identical	Multiple
W/D Ratio		0.220	0.219	0.140	0.138	0.112	0.105
Plating Thickness		2.683	2.683	1.553	1.520	1.113	1.061
Thickness Frame	1	.680	0.802	0.582	0.642	0.538	0.611
	2	.680	0.521		0.615		0.312
	3	—	—		0.515		0.667
	4				0.516		0.308
	5	—	—		0.644		0.539
	6			0.582	0.646		0.633
	7	—	—	—	—		0.361
	8						0.613
	9						0.303
	10					0.538	0.643
Spacing	1	148.500	150.135	49.500	49.563	29.700	29.816
	2	148.500	146.815		49.563		29.837
	3				49.563		29.741
	4				49.563		29.655
	5				49.563		29.615
	6			49.500	49.187		29.201
	7						29.585
	8						29.479
	9					29.700	29.485
n values of active buckling modes		4,8,9	4,6,8,9	3,13,14,15	3,12,13,14	3,11	3,9,10,11
Buckling Modes					15		
m values of active buckling modes		1,4	1,2,4	1,12	1,9,12	1,20,22	1,2,4,6,8,20
buckling modes							

Now consider the hulls using nineteen frames. Here the use of multiple frame sizes saves about 5-1/2% in weight. The design using identical frames is controlled, as expected, by buckling modes where m = 1 and m = 20 are dominant. The use of multiple frame sizes allows redistribution of framing material to help suppress these modes. This redistribution occurs until modes where m = 1,2,4,6,8, and 20 all control the design. Further significant improvement in frame material distribution then becomes impossible. In this case the plating thickness is reduced significantly by framing material redistribution because of the relatively short panel segments and the relatively large frame to plating thickness ratio. The torsional stiffness of the frames under such conditions is important in the panel buckling mode behavior. Thus redistribution of framing material can effectively be used to suppress such modes where frames are closely spaced. The plating thickness in such designs using multiple frame sizes can therefore be significantly thinner than in optimal designs using identical frame sizes. The combined effects of savings in the weight of both frames and plating then produces significant overall weight reduction.

Now finally consider the designs where eleven frames are employed. The changes resulting from the optimal use of multiple size stiffeners is, as expected, greater than when three stiffeners are used but less than when nineteen are employed. There is, a slight but significant decrease in weight and plating thickness but the reduction is much less than for the case of nineteen frames.

It may be seen therefore in cases where buckling controls hull design that the improvement possible through use of multiple frame sizes increases as the number of frames increases. This is fortunate since optimal designs using identical frames have a relatively large number of frames [1]. Thus the best designs are those that seem to benefit the most from use of multiple frame sizes.

The situation where plating and/or frame yielding controls the design has not been studied since it was felt that the greatest benefit resulting from multiple frame sizes occurs in cases where buckling is dominant and it was felt that the preliminary study should first explore those areas of greatest potential. One would expect negligable improvement in resistance to yielding from use of multiple size frames. However, where both buckling and yielding control design multiple size stiffeners may be of significant value.

Additional parametric studies are needed to more fully determine where multiple frame sizes can effectively be employed. Extrapolating the results of this work it appears where buckling alone controls the design weight savings greater than those obtained here may be expected from an optimal design using an optimal number of multiple size frames since the number of frames used in minimum weight designs tend to be substantially higher than the cases studied here.

CONCLUSION

It should be emphasized that this is a preliminary study to investigate the problem of optimal design of cylindrical structures reinforced with differing size frames The program developed here is a research program and no design capability is claimed.

Much, however, has been learned about such optimal shells, particularly the fact that many buckling modes are simultaneously active in these designs.

This mode coalesence has significant implications in analysis since it raises the question of the adequacy of design buckling criteria. Most failure criteria are based on study of a single failure mode and thus ignore interaction between modes. Their accuracy and safety under conditions where several modes are active is therefore suspect since one would expect interactions between failure modes to produce a reduction in structural strength. Thus for optimal designs to be used with confidence, existing failure criteria must be validated under simultaneously active mode conditions or, more likely, criteria considering mode interaction must be developed.

Long running time are required for the solution of optimal design problem using uniform frames of differing size. An attempt to use the above procedure for the more realistic case of nonuniform frames would be impractical because of the large increase in computational effort needed to solve the required eigenvalue problems. Thus in addition to the parametric studies needed to more fully examine the uniform stiffener problem a research effort is needed to produce substantial improvements in the optimization algorithm particularly in the method of calculating constraint derivatives if the more practical structures with nonuniform frames are to be studied from an optimization viewpoint or if an optimal design capability for such structures is to become practical.

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18. SUPPLEMENTARY NOTES

19. KEY WORDS (Continue on reverse side if necessary and identify by block number)

stress, design synthesis, buckling, shell synthesis aspects.

20. ABSTRACT (Continue on reverse side if necessary and identify by block number)

This report describes the solution to the singly reinforced discrete stiffener problem by direct use of an optimization scheme employing mathematical programming. The optimization formulation and procedure used here admit a large number of simultaneous buckling modes thus allowing optimization under conditions of mode coalescence, which are usual in this type of problem. This is a preliminary study, the purposes of which are; to develop and evaluate methodology for the treatment of this problem, to develop preliminary insights into how multiple frame sizes may efficiently be employed in submersible structures, to examine the nature of design improvement resulting from use of multiple size frames, and to investigate the nature of such optimal designs.

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