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General Panel Sizing Computer Code and Its Application to Composite Structural Panels

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A computer code for obtaining the dimensions of optimum (least mass) stiffened composite structural panels is described. The procedure, which is based on nonlinear mathematical programming and a rigorous buckling analysis, is applicable to general cross sections under general loading conditions causing buckling. A simplified method of accounting for bow-type imperfections is also included. Design studies in the form of structural efficiency charts for axial compression loading are made with the code for blade and hat stiffened panels. The effects on panel mass of imperfections, material strength limitations, and panel stiffness requirements are also examined. Comparisons with previously published experimental data show that accounting for imperfections improves correlation between theory and experiment.

Nomenclature

A	= planform area of stiffened panel
b_k	= element lengths defined in Figs. 3 and 4
E_1, E_2	= Young's modulus of composite material in fiber direction and normal to fiber direction, respectively
ET	= longitudinal extensional stiffness of panel
e	= overall bow in panel, measured at panel midlength (see Fig. 1)
G_{12}	= shear stiffness of composite material in coordinate system defined by fiber direction
GT	= shear stiffness of panel
L	= panel length (see Fig. 1)
M	= bending moment caused by bow in panel and/or lateral pressure
N_x, N_y, N_{xy}	= applied longitudinal compression, transverse compression, and shear loading, respectively, per unit width of panel (see Fig. 1)
$N_{x_{cr}}$	= value of N_x that causes buckling
N_{x_E}	= Euler buckling load of panel
$N_{x_{cr}}(e=0)$	= theoretical value of N_x that causes buckling when no initial bow is present
$(N_x)_{\text{design}}$	= value of N_x for which panel is designed
P	= pressure loading on panel
t_k	= lamina thickness defined in Figs. 3 and 4
W	= mass of stiffened panel
$\frac{W/A}{L}$	= mass index
X, Y, Z	= coordinate axes defined in Fig. 1
z_k	= offsets defined in Fig. 6
$\epsilon_{\max}$	= maximum allowable value of longitudinal or transverse strain in each lamina
μ_{12}	= Poisson's ratio of composite material in coordinate system defined by fiber direction
λ	= buckling half-wavelength
γ	= N_x/N_{x_E}
ρ	= density

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Introduction

THE introduction of composite materials has greatly expanded the options for obtaining efficient structural design. Because of the number of options, the task of finding the least-mass configuration of even a simple composite structure, such as a flat panel, is greater than for a similar structure fabricated from isotropic material. In recent years studies have resulted in computer codes for design of specific stiffened panel configurations subject to simple loadings (usually axial compression) which have provided needed insight into characteristics of efficient panel design.[1-4] Often these methods are based on simple buckling equations similar to those employed with metal panels. Without established criteria to rule out undesirable proportions, simple methods sometimes fail to predict panel behavior accurately. For example, an accurate analysis of corrugated panels which had been designed on the basis of simple buckling equations showed that buckling occurred in a twisting mode at a load much lower than the design load.[5] These results indicate the desirability of using in the design process an accurate analysis code that can account for all possible buckling modes. The feasibility of using an accurate buckling analysis in an optimization code was also established for specific configurations.[5]

Another aspect of panel behavior is that geometric imperfections can substantially reduce the strength of optimized panels. Several studies of the effect of imperfections on buckling of optimized structures have been made.[6] A simplified procedure for accounting for an overall bow-type imperfection and effects of lateral pressure has been developed which appears to correlate with experimental results for metal panels.[7]

In the present paper, an automated code suitable for final panel sizing, which includes the effects of an overall imperfection, is presented and discussed. Capabilities of the code are described and various applications given. Design studies are presented for blade and hat-stiffened panels subject to compressive loadings. The studies show the effect of imperfections, material strength, and axial and shear stiffness requirements. Finally, comparisons are made between predicted results and previously published test results for hat-stiffened panels with imperfections. The computer code, denoted PASCO (panel analysis and sizing code) will be made available through COSMIC.

Capabilities of the Design Code

The code has been designed to have sufficient generality in terms of panel configuration, loading, and practical constraints so that it can be used for final sizing of panels in a

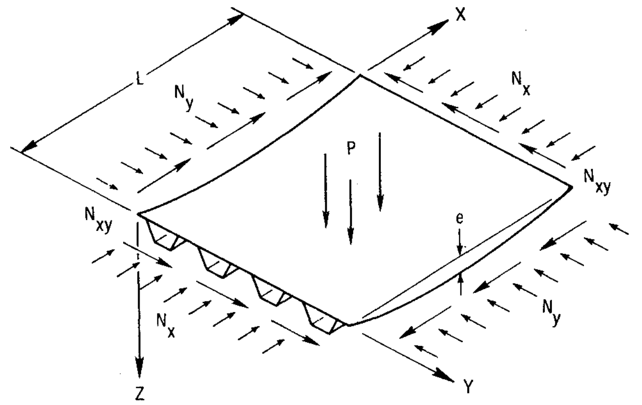


Fig. 1 Panel loadings and imperfection treated by sizing code.

realistic design situation. The panel may have an arbitrary cross section composed of an assembly of plate elements with each plate element consisting of a balanced symmetric laminate of any number of layers. Each layer can have orthotropic material properties and can be oriented in any direction resulting in anisotropic bending stiffness. Any group of dimensions, including ply angles, may be selected as design variables; the remaining dimensions can be held fixed or related in a linear fashion to the design variables.

The panel may be loaded by any combination of in-plane loadings (tension, compression, and shear) and lateral pressure as indicated in Fig. 1. Multiple load cases, that is, requiring the panel to sustain several sets of loading conditions, can be treated easily. Thermal stresses resulting from temperature changes are calculated. The material properties corresponding to the temperature of each plate element may be changed for different load cases. The effect of an overall panel imperfection as shown in Fig. 1 is included in a simplified manner.[7]

Realistic design constraints such as minimum ply thickness, fixed stiffener spacing, upper and/or lower bounds on extensional and shear stiffness may be prescribed. In addition, the natural vibration frequency of the panel may be specified to exceed a given value. Buckling loads and natural frequencies are calculated by the VIPASA (vibration and instability of plate assemblies including shear and anisotropy) computer code.[8] In addition, stress and strains in each layer of each plate element are calculated and margins against material failure based on an assumed failure criterion are assured. At present the user has the option of a maximum strain or maximum stress criterion. Other criteria can be easily installed once the equations in terms of stress and/or strain are established. Only a portion of the broad capability of the code is illustrated in this report. Emphasis is placed on panels with an initial bow under uniaxial compression subject to material strength and stiffness constraints.

The limitations of the design code are basically those contained in the buckling analysis that is discussed in the following section. In addition, the temperatures are constant within each plate element and are not functions of the design variables.

Analysis and Optimization Method

Buckling Analysis

The buckling analysis used in the computer program is performed by an efficient stiffened-panel buckling analysis code denoted VIPASA.[8] The VIPASA analysis treats an arbitrary assemblage of plates with each plate loaded by N_x, N_y, and N_{xy}. The analysis connects these individual plate elements and maintains continuity of the buckle pattern across the intersection of neighboring plate elements. The solution is exact in the sense that it is the exact solution of the plate equations satisfying the Kirchoff-Love hypothesis subject to the following assumption on behavior in the longitudinal direction. The stiffened panel cross section, loading, and temperature are assumed to be uniform in the x direction (Fig. 1) and the buckle pattern is sinusoidal along the length. For orthotropic panels with no shear loading, node lines are straight and perpendicular to the longitudinal panel axis conforming to the classical simple support boundary conditions at the ends of the panel. However, if shear and/or anisotropy is present, node lines become skewed and the boundary condition of simple support along the panel ends is not satisfied. For buckle lengths small compared to panel length the results are reasonably accurate. However, for buckle lengths equal to panel length results are often quite conservative.

Geometric Imperfections and Lateral Pressure

The first-order effect of an overall bow-type geometric imperfection or lateral pressure is assumed to be the additional stress resultants produced by bending. The maximum bending moment can be calculated by the classical beam column formula[9]:

$$M=\frac{N_x e}{1-\gamma}+\frac{PL^2}{\pi^2}\frac{1}{\gamma}\left[\sec\left(\frac{\pi}{2}\sqrt{\gamma}\right)-1\right] \tag{1}$$

where

$$\gamma = N_x / N_{x_E} \tag{2}$$

and N_x is the applied axial load per unit width, N_{x_E} is the Euler buckling load for the panel, and L is the panel length. The stress resultant distribution caused by this bending moment is added to that for the in-plane loading to calculate stresses and buckling loads. For $\lambda = L$, this bending moment is omitted consistent with the observation that the Euler load is the asymptote of the strength of a column with an initial imperfection. However, the bending moment does affect other failure modes. The additional stresses caused by the bending moment superimposed on those caused by the applied in-plane loading can result in stresses or strains exceeding the allowable values, or in buckling with a buckle length some fraction of the panel length. In the VIPASA analysis, the stress must be constant along the length. Therefore, the conservative assumption of taking the stress associated with the maximum bending moment is made. For the shorter buckle lengths the assumption is quite accurate. The stress must also be constant in each plate element so that it is necessary to subdivide plates that have the Z coordinate varying in order to approximate the bending behavior with reasonable accuracy.

Optimization

The details of the optimization procedure are essentially those developed in Ref. 5 for specific panel configurations and loadings. The program is divided into three separate modules. In the analysis module, all constraints (buckling, stiffness, stress, or strain) are calculated with the VIPASA program and supporting subroutines. The program automatically calculates constraints for the critical buckling modes and the modes that could become critical during the optimization. Derivatives of all constraints are calculated numerically by making separate small increments to each design variable. (Any width, thickness, or ply angle may be a design variable. Ply angles were not, however, used in the present studies.) In the second module, a first-order Taylor series representing all significant constraints is developed using the value of the constraints and their derivatives from the analysis module. This approximate analysis technique is similar to that used by Schmit and Miura.[10] The Taylor series

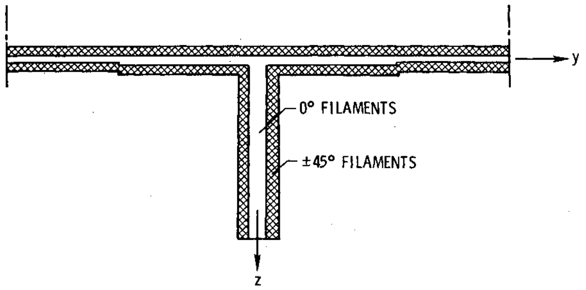


Fig. 2 Blade-stiffened panel configuration.

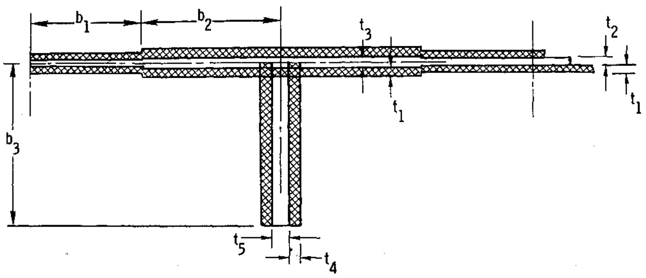


Fig. 3 Mathematical model of blade-stiffened panel, including design variables.

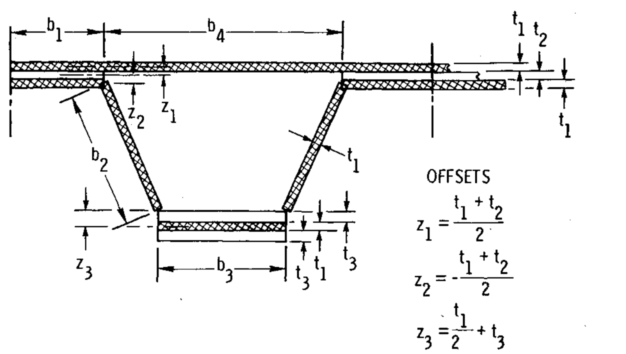


Fig. 4 Mathematical model of hat-stiffened panel configuration, including offsets and element width and thickness design variables.

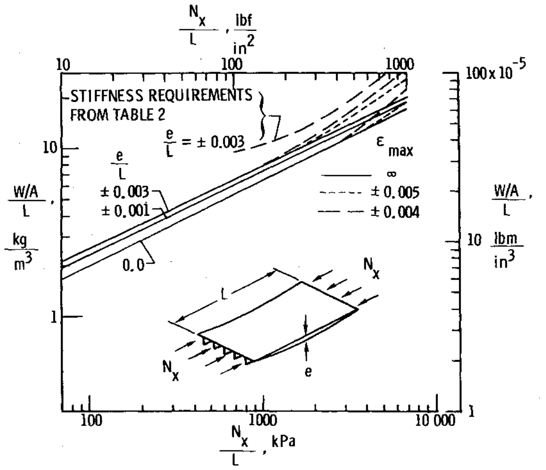


Fig. 5 Structural efficiency of graphite-epoxy blade-stiffened panels.

is a reasonably accurate but relatively simple local representation of the total problem. The objective function (panel mass) along with the Taylor series expansion of the constraints is passed to the third module which is the optimizer CONMIN.[11] The program CONMIN is a general optimizer which will minimize an objective function subject to inequality constraint relations involving the design variables. It is based on a feasible direction algorithm. If the input design violates any constraint, the initial portion of the optimization develops a feasible design while the later portion minimizes the mass. During the sizing in CONMIN, the maximum changes in the design variables are controlled to prevent serious errors in the Taylor series approximation. New values of the design variables are obtained and this configuration is returned to the analysis module for a complete analysis by VIPASA (including derivitives); the whole process is repeated as many times as required to achieve a converged design that satisfies all constraints when subject to accurate analysis. Typically 10 cycles are adequate to produce a converged design. Local optima are an occasional problem but are usually easy to recognize in a series of similar cases or by changing starting points. When the optimization is initiated at different starting points or when various control parameters are used to force the optimization to follow different paths, the resulting designs may have slightly different proportions. But the masses of these designs are very nearly the same.

This design process usually leads to configurations which have more than one buckling mode critical so that little postbuckling strength is shown by each panel in test. To design efficient panels that utilized significant postbuckling strength would require a major improvement in analysis capability to treat this effect properly.

Design Studies for Composite Panels with a Bow-Type Imperfection

It is known that imperfections can have a significant effect on panel strength if panel dimensions have been optimized for minimum weight. The usual practice in design is to require margins of safety developed through experience and testing to insure adequate strength for panels manufactured within prescribed tolerances. The new capability in the present code allows a more systematic approach to design in the presence of expected imperfections. The effect of bow-type imperfection on panel mass is investigated by conducting a number of design studies. Both blade- and hat-stiffened panels fabricated from graphite-epoxy and subjected to longitudinal compressive loadings were studied. The following material properties were used. $E_l = 131$ GPa, $E_2 = 13.0$ GPa, $G_{l2} = 6.41$ GPa, $\mu_{l2} = 0.31$, and $\rho = 1589$ kg/m^3. Results and comparisons are presented using structural efficiency diagrams in which the mass index W/AL is given as a function of the loading index N_x/L. Additional studies were also made showing the effect of material strength and in-plane stiffness requirements on panels with and without imperfections. All results are for a panel length L equal to 76.2 cm (30 in.).

Panel Configurations

Blade-Stiffened Panel

The panel cross section is shown in Fig. 2. The offsets in the actual configuration were neglected so that the mathematical model of the blade-stiffened panel is as shown in Fig. 3. The configuration is defined in terms of the elements widths b_l, b_2, b_3, the thicknesses t_l and t_4 of ±45-deg layers, and the thicknesses t_2, t_3, and t_5 of 0-deg layers. The filament orientation is measured with respect to the X axis. All laminates are balanced and symmetric. The three widths and the five thicknesses shown in Fig. 4 are the design variables in the structural synthesis. No upper or lower bounds were placed on the design variables. Except for the cases involving longitudinal extensional stiffness requirements, the lightest designs were designs with t_2 equal to zero.

Panels were considered to be many bays wide, usually 8 or 16, with symmetry boundary conditions. This arrangement was found to allow for all possible mode shapes in the y direction that would occur for a wide panel. Thus, optimum stiffener spacing is readily determined. In the case of shear

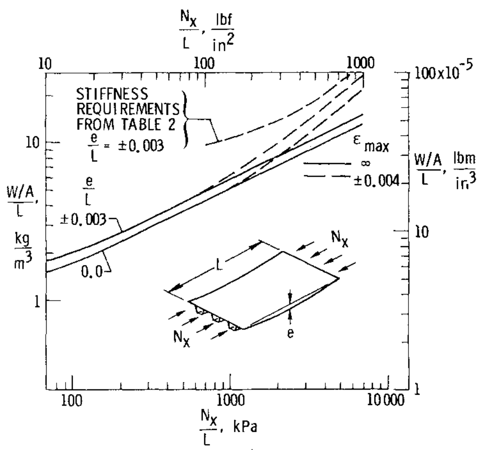


Fig. 6 Structural efficiency of graphite-epoxy hat-stiffened panels.

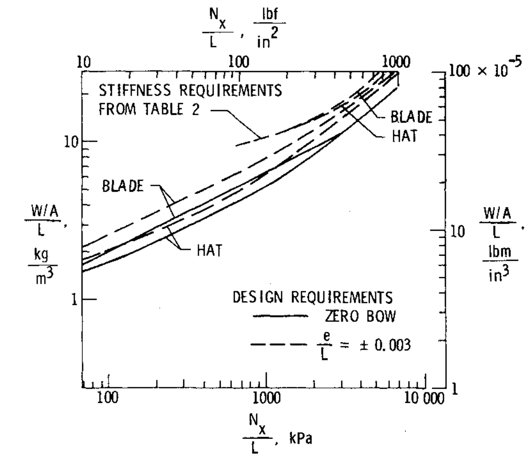


Fig. 7 Structural efficiency of graphite-epoxy blade-stiffened panels and hat-stiffened panels. All panels have ϵ_{max} = ±0.004.

Table 1 Minimum stiffness requirements representative of commercial aircraft wing construction

Loading intensity N_x/L*, kPa	Longitudinal stiffness ET, MN/m	Shear stiffness GT, MN/m
689	386	79
1720	508	105
3440	613	133
5510	701	165

*L = 76.2 cm (30 in.)

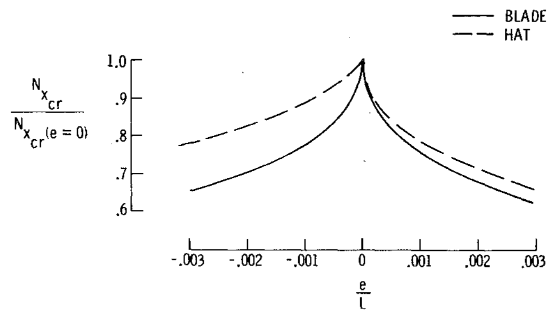


Fig. 8 Effect of bow on buckling load of the blade-stiffened panel and hat-stiffened panel designed for zero bow.

(N_{xy}) or transverse (N_y) loadings, the results are sensitive to panel width. For these loadings, it is best to use the actual panel width with the correct boundary conditions. In this case, several design runs may be made holding stiffener spacing to different integral fractions of the width.

Hat-Stiffened Panel

The panel configuration is shown in Fig. 4. In this case the more detailed modeling capability of the program was utilized in accounting for the offsets that are present because of the finite thicknesses of each plate element. The configuration is defined in terms of the four element widths b_1, b_2, b_3, b_4, the thickness t_1 of 45-deg layers, and the thicknesses t_2 and t_3 of 0-deg layers. These dimensions are the design variables in the structural synthesis. In addition the following upper and lower bounds were placed on some of the design variables to control the design to reasonable dimensions.

$$\left.\begin{matrix} b_3 \\ b_4 \end{matrix}\right\} \geq 2.03 \text{ cm (0.8 in.)}$$

$$t_3 \leq 1.27 \text{ cm (0.5 in.)}$$

Structural Efficiency Results

The structural efficiency diagram for the blade-stiffened panel studied is shown in Fig. 5. The solid lines represent models that have only a buckling requirement. Each curve was generated by calculating the panel mass at four to six values of N_x/L. By allowing continuous variation of all design variables (as opposed to step changes in ply thickness) the smooth curves result. The panels are designed for three sets of values of the bow parameter: e/L = 0.0, ±0.001, and ±0.003. The curves for which e/L = ±0.003 represent acceptable designs for two conditions: 1) the design loading with a positive value of e/L and 2) the design loading with a negative value of e/L. If only one condition were used, the panel could buckle if the bow were not present—a situation which would be unacceptable. It was found that panels designed for equal bows in both directions would also carry the design load for any value of bow between the two extremes. The increase in mass required to design for the bow varies with the loading. Lightly loaded panels [N_x/L = 68.9 kPa (10 lbf/in.^{2})] with e/L = ±0.003 are about 27% heavier than panels with no bow. Heavily loaded panels [N_x/L = 6890 kPa (1000 lbf/in.^{2})] are about 17% heavier.

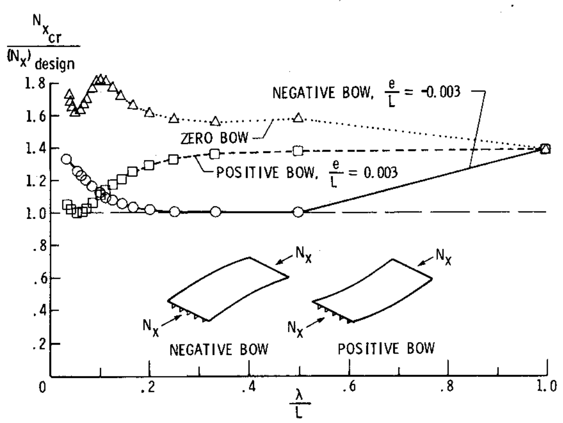
In Ref. 7 a value of e/L = 0.001 was shown to correlate with test results for metal panels. However, the results of Fig. 5 show that the allowable bow could be triple this value with only a modest mass increase compared to the mass increase in going from e/L = 0.0 to e/L = ±0.001.

The structural efficiency diagram for the hat-stiffened panel is shown in Fig. 6. It is seen that the mass increase due to the bow is much less than that observed for the blade-stiffened panel. A more detailed comparison of the blade- and hat-stiffened panel will be given in a later section.

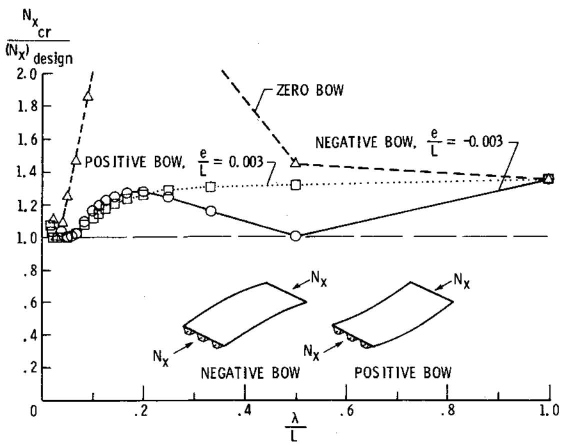
Effect of Material Strength

The dashed curves in Figs. 5 and 6 represent the effect of material strength requirements which, for these cases, are maximum allowable lamina strains. The ±0.004 designation means that for these panels the longitudinal and transverse strains in any lamina do not exceed ±0.004. This value of maximum strain is approximately the strain level that can be imposed on panels without suffering significant strength loss due to local damage according to the experimental data of Ref. 11. An indication of the sensitivity to the value used for ϵ_{max} is indicated in Fig. 5 where results for ϵ_{max} = ±0.005 are shown for the blade-stiffened panel. The allowable shear strain in all cases was sufficiently high so that it was not an active design requirement.

An initial bow has a large effect on the load at which material strength considerations become important. Consider, for example, the blade-stiffened panel with a maximum



a)



b)

Fig. 9 Ratio of buckling load to design load as a function of buckling half-wavelength for a blade-stiffened panel and for a hat-stiffened panel designed for a loading of $N_x/L = 689$ kPa (100 lbf/in.^{2}) and for a bow of $e/L = \pm 0.003$; a) blade-stiffened panel, b) hat-stiffened panel.

strain requirement of ± 0.004. For $e/L = 0.0$ the material strength requirement begins to have an effect at N_x/L equal to about 2760 kPa (400 lbf/in.^{2}). For $e/L = \pm 0.003$ the material strength requirement begins to have an effect at N_x/L equal to about 689 kPa (100 lbf/in.^{2}). For clarity, design curves incorporating material strength requirements are not shown for the $e/L = \pm 0.001$ case.

Effect of Stiffness Requirements

Wing panels may have extensional and shear stiffness requirements as well as buckling and material strength. The impact of stiffness requirements is shown in Figs. 5 and 6 also. The stiffness requirements chosen, which are typical of current aluminum wing construction, are given in Table 1.[12] A bow imperfection of $e/L = \pm 0.003$ was also included. Although not shown, other calculations without the effect of the bow resulted in designs only a few percentage points lighter. However, if these designs were analyzed with a bow, buckling occurred as low as 50% of the design load.

Comparison of Blade-and Hat-Stiffened Panel

Structural Efficiency

A structural efficiency diagram is shown in Fig. 7 for the two configurations with two values of bow ($e/L = 0.0$ and $e/L = 0.003$), material strength requirement of $\epsilon_{max} = \pm 0.004$, and the stiffness requirements given in Table 1. The hat configuration is lighter than the blade and is not as adversely affected by the bow. When material strength or stiffness requirements are controlling, there is little difference in the mass; however, the presence of a bow always gives an advantage to the hat-stiffened panel when material strength or buckling is controlling the design.

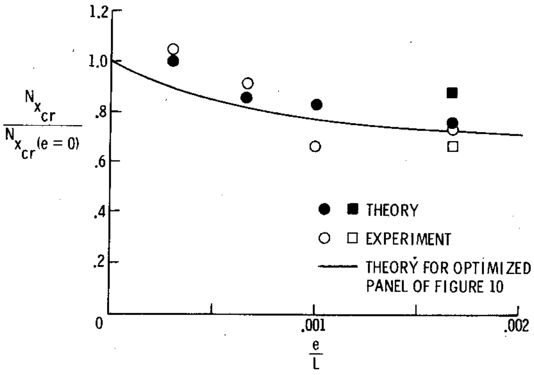


Fig. 10 Comparison of theory and experiment for hat-stiffened panels with an initial bow.

Effect of Bow on Panels Designed for Zero Bow

Both a blade- and hat-stiffened panel designed to support a loading of $N_x/L = 689$ kPa (100 lbf/in.^{2}) with zero bow were analyzed for various amounts of bow. The results in terms of nondimensional quantities are presented in Fig. 8. Positive values of e/L produce a moment that puts the tip of the blade or hat in tension and the skin in compression. For a value of e/L of only 0.001, the buckling load is reduced by about 24% for the blade-stiffened panel and by about 20% for the hat-stiffened panel. The hat-stiffened panel is a more efficient configuration for resisting bending, which is probably why it suffers less of a loss in load carrying ability in the presence of a bow. This same characteristic is responsible for the hat-stiffened panel requiring less of a mass increase than the blade-stiffened panel when designed for a bow.

Although the reduction in the buckling load shown in Fig. 8 is almost symmetric with respect to $e/L = 0.0$ for the blade, this is not true for the hat. In some designs for which extensional and shear stiffness requirements were imposed, the buckling load curves were very asymmetric with respect to $e/L = 0.0$ for both configurations.

Buckling Response of Panels Designed for a Bow

Examination of the buckling response as a function of buckle length gives some indication of the possible failure modes of the panel. Examples are shown in Fig. 9 for the blade and hat configurations. These examples are for panels designed for N_x/L = 689 kPa (100 lbf/in.^{2}) and e/L = ± 0.003. The mass-strength data for these cases are included in Figs. 5 and 6. As stated earlier, the panel carries the design load with either a positive or negative bow. The buckling load as a function of buckling half-wave length is shown in Fig. 9 for the positive bow (0.003), negative bow (-0.003), and for zero bow.

The circular symbols and square symbols represent the buckling response of the panel when it is analyzed with e/L = -0.003 and e/L = 0.003, respectively. For a negative bow the stiffener is in additional compression due to bending. This results in the longer buckle length stiffener rolling or twisting modes being critical. For a positive bow (skin in compression) the local skin buckling of the blade configuration becomes critical. For the hat-stiffened panel, buckling at the smaller buckle lengths occurs in the web of the hat and the effect of the bow on this mode is small.

The bow does not directly affect the buckling load for λ = L. For this reason, the panel has the same buckling load at λ = L for both the positive, negative, and zero bow. As seen in Fig. 9, the buckling load at λ = L is substantially higher than the design load. The synthesis procedure automatically leads to this result to prevent the bending moment predicted by Eq. (1) from becoming excessive. The triangular symbols represent the buckling response of the panel when it is analyzed with $e/L = 0$. For this case the blade-stiffened panel has a substantial margin over the design load at all buckle lengths; however, the short wave length buckling load for the hat is very close to the design load even if the bow is not present.

The fact that several buckling loads are critical in Fig. 9 demonstrates the importance of considering several buckling constraints simultaneously. In addition to considering buckling loads for several values of λ, the program can also consider several eigenvalues for a given value of λ. This latter capability is necessary to prevent mode switching of the type encountered in Ref. 5.

Comparison With Experiment

Inclusion of imperfections in the method of the present paper is an attempt to explain the discrepancy which often exists between theory and experiment for panels loaded in compression. The degree to which the proposed simple approach succeeds in quantitatively defining the effect is illustrated in Fig. 10. The experimental panel compressive buckling load, normalized by the predicted load assuming no imperfection, is shown as a function of the measured imperfection for the 152.4-cm (60-in.) long hat-stiffened panels.[3] Complete details on panel construction, dimensions, and material properties are given in Ref. 3. These panels were designed utilizing an earlier sizing code based on a simplified buckling analysis. In some cases empirical allowances were made for such things as imperfections, overall beam transverse shear, and anisotropic effects. In general, the local and column buckling loads were close together so that interaction could be expected in the presence of an imperfection, and buckling and collapse were usually close together. There is no single theoretical buckling curve as a function of imperfection; therefore each calculation is shown by a single symbol. (To avoid confusion where there are two results at the same value of e/L, two separate symbols are used.) As a reference, the curve from Fig. 8 for an optimized hat-stiffened panel is also shown.

In calculating the theoretical loads for the test panels, it was assumed that the flat end test conditions produced nearly clamped boundary conditions, so that the effective simple support length, which must be used in the theory, was taken as one half the actual length. It can be shown[9] that the imperfection should also be reduced by one half to produce the same bending moment at the center of the simply supported columns for a given axial load. Thus, the same value of e/L is appropriate for both the clamped experimental panels and for the simply supported panels of the theory.

Not all of the test panels were found to be sensitive to the imperfections. The one plotted at the smallest value of e/L had a critical local web buckling mode that was fairly insensitive to imperfection. Since skin buckling was at a significantly higher load, web buckling could occur without catastrophic collapse. In fact, this test result is about 5% higher than a prediction based on initial buckling as failure.

Considering the rather formidable difficulties in determining the proper thickness and material properties for accurate prediction of composite panel strength, the correlation between theory and experiment is considered good. The nominal material properties and thicknesses[3] were used in the calculations. The maximum difference of about 25% is for a panel fabricated from fabric rather than tape. The material properties given for the fabric did not include any anisotropic bending effects, which, if included in the theory, would result in closer agreement with experiment for this panel. In addition, no special effort was made to account for the second-order effects introduced by the clamped boundary conditions as discussed by Williams and Stein.[13] Considering all of the factors affecting the results, it is concluded from Fig. 10 that accounting for imperfections is an important consideration in the design of stiffened panels.

Concluding Remarks

A computer code denoted PASCO (panel analysis and sizing code) is described for sizing composite structural panels. A number of design studies were made on blade and hat-stiffened panels subject to compressive loads. The effect of imperfections, material strength, and in-plane stiffness requirements were investigated. A small imperfection was found to have a significant effect on panel mass. If panels were optimized without consideration of imperfections, the buckling load with imperfections present was reduced significantly. The studies show the hat-stiffened panel to be more efficient than the blade-stiffened panel and not as adversely affected by imperfections. However, where stiffness and/or strength were controlling, the mass of the two configurations was more nearly the same.

Comparisons made between the present theory and experiment for hat-stiffened panels having bow-type geometric imperfections show the importance of accounting for imperfections. The computer code described herein provides a rational means for including the effect of a bow-type imperfection and should be a considerable aid in the design of stiffened composite panels. The code will be made available through COSMIC.

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