Minimum-Weight Design of Stiffened Cylindrical Panels under Combined Loads

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Theme

THE increasing demand for lightweight structures has made the structural engineer more conscious of minimum-weight design. Since stiffened cylindrical shells have been used extensively during the past thirty years in underwater, surface, and aerospace vehicles, a tremendous effort has been exerted in designing such a configuration for minimum weight.

A methodology was developed and demonstrated by the first author and his collaborators[1-4] for designing a stiffened cylinder, subjected to various load conditions with minimum weight when at least one of the active modes of failure is known a priori. The nonmenclature employed herein is identical to that of Refs. 1-4. Since the structural geometry of the above-mentioned vehicles (aircraft fuselage, submarine hull, etc.) is best represented by a combination of stiffened cylindrical panels, the methodology is, herein, applied to panels and extended to accommodate the combined application of loads.

The precise statement of the problem considered, in this extension, is as follows: Given a stiffened thin cylindrical panel of specified material, radius of curvature, length and width, find the realistic size, shape, and spacing of the stiffeners, and the realistic thickness of the skin, such that the resulting configuration can safely carry a given set of surface loads with minimum weight.

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The design objective is minimum weight. General instability is taken to be the equality constant. This means that in this particular case, the methodology is applicable to that part of the fuselage for which general instability is the active or one of the active failure modes (design the fuselage with minimum weight against general instability). For other parts of the fuselage or other uses of stiffened panels, the equality constraint can be a different failure mode. Panel instability, which is also a catastrophic failure mode, is considered as an inequality constraint. Other behavioral inequality constraints are local instability of the skin and stiffeners. When all of these constraints are met, the resulting configuration can carry the applied loads safely with minimum weight. In addition, geometric inequality constraints are used in order to ensure realistic dimensions for the design variables (thicknesses, shape, and spacings of the stiffeners).

The methodology of Refs. 1-4 is modified and extended so as to be applicable to a simply supported (classical), stiffened, rectangular, cylindrical panel under combined loads. The details may be found in Ref. 5, which serves as a backup paper. The solution is accomplished in two phases. First, by a proper grouping of the design variables, the number of parameters that optimize the weight is minimized; an unconstrained minimization (with general instability as the equality constraint) is performed and design charts are prepared, which clearly show the effect of the design parameters on the shell weight; this is accomplished by employing the irregular mathematical search method of Nelder and Mead,[6] in combination with a search and pattern waves method.[7,8] Second, on the basis of the design charts the design space is scanned to arrive at the minimum weight configuration, which satisfies all other constraints.

Both phases, herein (backup paper), are fully automated. Phase two is automated through small computer programs that use the generated design chart data, from phase one, as input data. All programs are listed in the backup paper.[5]

In addition to the automation another important feature of the methodology is that it permits the designer to deviate from the minimum weight solution with minimum penalty in weight (which in some cases is zero), in order to avoid simultaneous occurrence of failure modes (whenever such occurrence makes the resulting configuration more imperfection-sensitive) and/or avoid unrealistic values for the design variables.

Three design examples are presented on Table 1. Two of the examples correspond to uniaxial compression (along the direction of zero curvature) and the third to a combined uniaxial compression with shear. In addition, results are presented graphically as average (smeared) thickness vs skin thickness on Figs. 1 and 2.

The common data used for all three cases (examples) are: $R=85$ in.; $L=100$ in.; $b=100$ in.; $E=E_x=E_y=10.5\times10^6$ psi; $\rho=\rho_x=\rho_y=0.101$ lb/in.^{3}; $\nu=0.33$; $\sigma_y=50{,}000$ psi.

The three cases are:

Case 1: $\bar{N}_{xx}=2700$ lb/in.; $\bar{N}_{xy}=\bar{N}_{yy}=0$; RSRR.

Case 2: $\bar{N}_{xx}=2700$ lb/in.; $\bar{N}_{xy}=420$ lb/in.; $\bar{N}_{yy}=0$; RSRR.

Case 3: $\bar{N}_{xy}=2700$ lb/in.; $\bar{N}_{xy}=\bar{N}_{yy}=0$; TSRR.

Case 1 provides a basis of comparison for the other two cases. Case 3 represents the optimum stiffener shapes for

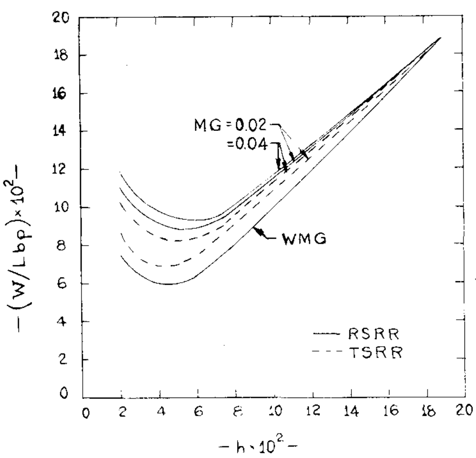

Fig. 1 Minimum smeared thickness vs skin thickness; $\bar{N}_{xx}=2700$ lb/in.

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Index categories: Structural Design (including Loads); Structural Stability; Subsystem Design.

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Table 1 Design results

	Case One (RSRR) $\bar{N}_{xx}$ = 2700 lb/in $\bar{N}_{xy} = \bar{N}_{yy}$ = 0						Case Two (RSRR) $\bar{N}_{xx}$ = 2700 lb/in $\bar{N}_{xy}$ = 420 lb/in $\bar{N}_{yy}$ = 0						Case Three (TSRR) $\bar{N}_{xx}$ = 2700 lb/in $\bar{N}_{xy} = \bar{N}_{yy}$ = 0			
	WMG		MG = 0.02 in.		MG = 0.04 in.		WMG		MG = 0.02 in.		MG = 0.04 in.		MG = 0.02 in.		MG = 0.04 in.	
h	0.04	0.067	0.04	0.067	0.04	0.067	0.04	0.067	0.04	0.067	0.04	0.067	0.04	0.067	0.04	0.067
W/Lbp	0.0598	0.0677	0.0902	0.0909	0.0956	0.0930	0.0679	0.0682	0.09960	0.0945	0.1064	0.0974	0.0683	0.0805	0.0824	0.0853
α_x	80.	120.	20.	10.	20.	15.	85.	120.	20.	10.	20.	10.	20.	10.	20.	10.
α_y	20.	60.	50.	30.	45.	25.	25.	60.	50.	35.	45.	30.	70.	60.	50.	40.
$\bar{\lambda}_{xx}$	0.3323	0.0029	0.9110	0.1878	1.0160	0.3070	0.4155	0.0031	1.0640	0.2093	1.1720	0.2380	0.3520	0.1495	0.7132	0.1508
$\bar{\lambda}_{yy}$	0.1190	0.0071	0.2314	0.1363	0.2527	0.046	0.2199	0.0131	0.2934	0.1639	0.3382	0.1746	0.2918	0.0339	0.2535	0.0982
c_x	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.1087	1.1087	1.1087	1.1087
dw_x	3.2	8.04	0.8	0.67	0.8	1.005	3.4	8.04	0.8	0.67	0.8	0.67	0.695	0.582	0.695	0.582
dw_y	0.8	4.02	2.0	2.01	1.8	1.675	1.00	4.02	2.0	2.345	1.8	2.01	2.8	4.02	2.0	2.68
ℓ_x	1.121	2.080	1.355	2.280	1.393	2.400	1.157	2.080	1.005	2.306	1.369	2.246	1.128	2.245	1.279	2.245
ℓ_y	0.074	*	4.697	1.390	6.330	1.300	0.093	*	2.199	2.322	10.670	9.187	4.270	31.000	7.022	14.500
tw_x	0.0052	*	0.0690	0.0400	0.0790	0.0550	0.0060	*	0.0600	0.0540	0.0900	0.0600	0.0190	0.0310	0.0430	0.0400
bf_x	-	-	-	-	-	-	-	-	-	-	-	-	0.264	0.220	0.264	0.2200
tw_y	0.0005	*	0.0200	0.0200	0.0400	0.0400	0.0010	*	0.0200	0.0200	0.0900	0.0600	0.0200	0.0200	0.0400	0.0400
t_{fx}	-	-	-	-	-	-	-	-	-	-	-	-	0.0190	0.0200	0.0430	0.0400

*Very small number.

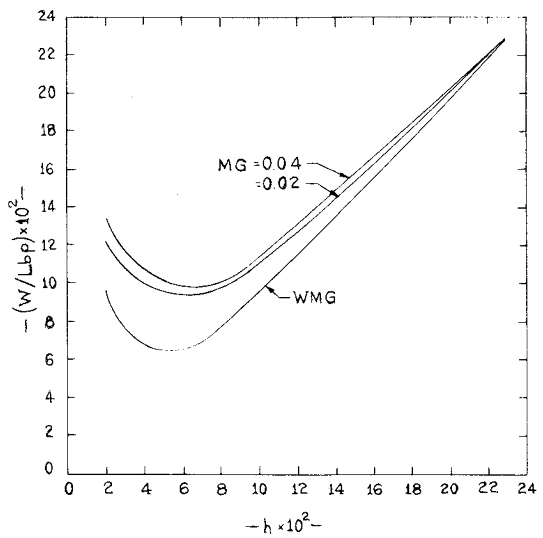

Fig. 2 Minimum smeared thickness vs skin thickness; RSRR; $\bar{N}_{xx}$ = 2700 lb/in.; $\bar{N}_{xy}$ = 420 lbs/in.

uniaxial compression. In all cases, designs are generated without a geometric minimum gage constraint, and with a minimum gage constraint of 0.02 in. and 0.04 in. for all geometric components. It is clearly seen from Figs. 1 and 2 that the actual minimum weight design for all three cases corresponds to a skin thickness between 0.04 in. and 0.067 in. The corresponding smeared thickness for case 3 and MG = 0.04 in. is 0.082 in., while for MG = 0.02 in. it is 0.070. These observations seem to be in agreement with the complete cylinder results of Ref. 3 (Table 6, Case 2) for this geometry, which yield an optimum smeared thickness of 0.088 in. when the MG is 0.05 in.

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