STABILITY LIMITS OF ORTHOTROPIC SHALLOW SANDWICH SHELLS - COMPUTATION AND TEST RESULTS -*

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Abstract

This paper deals with a computer programme for analysing bifurcation buckling and natural vibrations of orthotropic shallow sandwich shells. The programme is discussed briefly, and tests are described which were initialised to check the computed buckling loads. From the first test results conclusions are drawn concerning the significance of computed buckling loads and the continuation of the test programme.

I. Introduction

Many of the modern finite element computer programmes are capable of computing bifurcation buckling loads of complex structures. As these tasks lead to eigenvalue problems, their solution requires much more computation time than the calculation of stresses and deformations. Therefore, only characteristic components of complex structures such as columns, plates, or shells are usually analyzed with respect to their buckling behaviour. It might be doubted, that in this case general-purpose computer programmes are best suited. Special-purpose programmes might be more economic tools.

It is well known that buckling loads of thin shells under certain loading conditions, e.g. isotropic cylinders subjected to axial load, may exhibit a large amount of scatter, with average values far below the theoretical buckling loads. Theoretical procedures must therefore be checked and qualified by tests.

This paper deals with a computer programme that was particularly designed for bifurcation buckling, and natural vibrations, of orthotropic shallow shells. It is believed that it requires a minimum of data preparation. The programme was named BEOS, which is an abbreviation for Buckling of Eccentrically Orthotropic Sandwich Shells. Moreover, the paper will outline a test programme initiated to give experimental evidence to the validity of the results of BEOS. Only a few tests could be performed so far. The results and consequences of these will be discussed.

II. The Computer Programme BEOS

Scope of the Programme

The most general structure to which the programme BEOS can be applied is a shallow sandwich shell with two different orthotropic faces and an orthotropic core (See Fig 1).

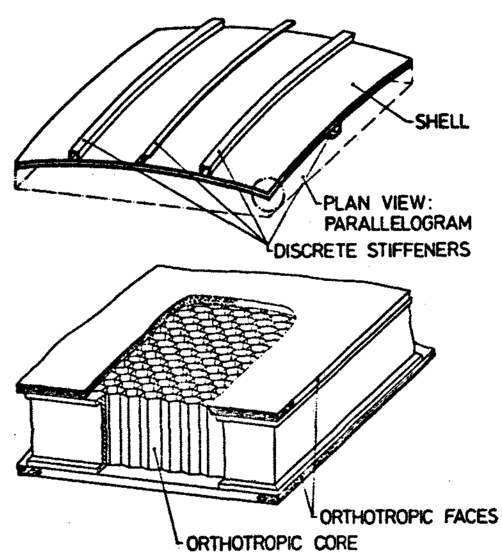


FIG.1: SHELL CONFIGURATION FOR COMPUTER PROGRAMME BEOS.

* Dedicated to Prof. Dr.-Ing. W. Thielemann on the occasion of his 70th birthday on Aug. 20th, 1978

The plan view of the shell must be a parallelogram. Widely spaced stiffeners running parallel to the edges may be fixed to either of the shell faces. The shell may be clamped or supported in different directions along its edges and/or at discrete points. Thus a variety of boundary conditions can be regarded.

It is assumed that the prebuckling equilibrium state, i.e. the distribution of membrane forces within the shell, has been determined independently. Any distribution can be dealt with. The programme will compute the factor by which this distribution must be multiplied in order to get to a bifurcation of equilibrium. Another option will cause the programme to analyze natural vibrations about the specified equilibrium state. Several bifurcation points, or natural frequencies, can be determined along with the corresponding deformation modes.

Theoretical Background. Shells are three-dimensional structures that are mentally reduced, for theoretical considerations and for ease of computation, to two-dimensional ones. This aim is achieved by means of certain assumptions concerning the variation of deflections along the normal direction. Most widely used are the Kirchhoff-Love assumptions, the most significant of which specifies that the normals to the shell remain normal at any deformation. The deformation of the shell can then be described completely by three functions of the coordinates (x,y). These coordinates span the reference surface of the shell, for which usually the median surface will be taken. The three functions are the displacements u,v in the tangential directions and the normal displacement w. Thus the Kirchhoff-Love shells have three functional degrees of freedom.

For sandwich shells this theory might be unsufficient. Therefore, the programme BEOS is based on a more general theory utilizing five functional degrees of freedom. The two additional freedoms are the shear deformations of the sandwich core. Alternatively they can be considered as functions specifying the relative movements of the face sheets. The deformation model is illustrated in Fig. 2. For the face layers of the sandwich, the thickness of which may be of the same order as the core height, the Kirchhoff-Love assumptions are used. Thus the core couples the deformations of two separate orthotropic shells that may have different properties. In performing this action the core height is preserved, but its deformation due to transverse shear may be considerable. The theory also takes into account the membrane forces and bending moments transferred by the core. The deformation model described admits that the normals to the shell reference surface no longer remain straight lines at deformation. They may become broken lines connected at the interfaces between core and face plies.

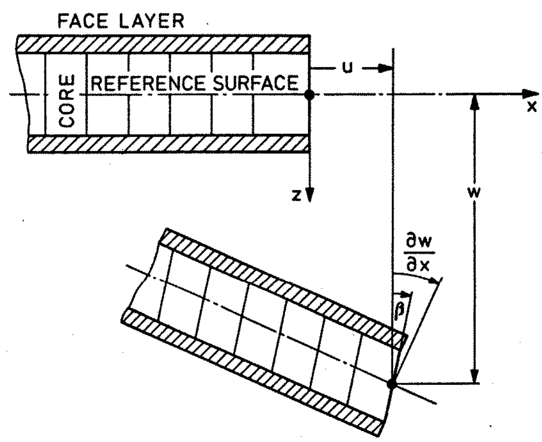


FIG. 2
DEFORMATION MODEL OF A SANDWICH SHELL WITH THICK FACES

The computation method is based on an energy formulation. The second variation of the total potential energy Π of a shell can be expressed as

$$\delta^2\Pi = \iint_{(S)} \delta\boldsymbol{\varepsilon}^T \mathbf{E}\, \delta\boldsymbol{\varepsilon}\, dS - \qquad (1)$$
$$-\lambda \iint_{(S)} \left[\bar{N}_x \left(\frac{\partial \delta w}{\partial x}\right)^2 + 2\bar{N}_{xy} \frac{\partial \delta w}{\partial x} \frac{\partial \delta w}{\partial y} + \bar{N}_y \left(\frac{\partial \delta w}{\partial y}\right)^2 \right] dS,$$

S being the reference surface.
This expression is twice the change of internal energy due to a small increment of deformation superimposed on an equilibrium state.

The membrane forces of this equilibrium state are $\bar{N}_x$, $\bar{N}_{xy}$, $\bar{N}_y$, with $\bar{N}_x$ and $\bar{N}_y$ positiv in compression. Their contribution to the second variation of internal energy is given in the second term of Equ. (1). The first term reflects the contribution of the stress increments, which arise from the strain increments $\delta\boldsymbol{\varepsilon}$. The matrix $\mathbf{E}$ is the stiffness matrix transforming strains into stresses:

$$\boldsymbol{\sigma} = \mathbf{E}\,\boldsymbol{\varepsilon} \qquad (2)$$

May it suffice here to mention that $\boldsymbol{\varepsilon}$ is a generalized strain vector the components of which are the in-plane strains of the reference surface, curvature changes, and transverse shear strains of the core. Correspondingly, $\boldsymbol{\sigma}$ is a generalized stress vector the components of which are membrane forces, bending and twisting moments, and transverse shear forces. For more detailed information see references [1, 2, 5].

In order to find a numerical solution of the variational problem

$$\delta(\delta^2\,\Pi) = 0 \qquad (3)$$

the buckling or vibration mode, i.e. the unknown increments of the five functional degrees of freedom, which are continous functions of (x,y), must be expressed in terms of a finite number of discrete unknowns. In accordance with the terminology used in the Rayleigh-Ritz method, these unknownswill be called "generalized co-ordinates". In the finite element method the term "degrees of freedom" is used alternatively. The discretisation process is as follows:

A grid of lines running parallel to the edges is used to divide the shell into subregions. The values of the five deflection functions, their first derivates in x- and y-direction, and the mixed second derivative, at the corners of the subregions, are used as generalized co-ordinates. Bicubic polynomial interpolation is applied to express the unkown functions in the interior of each subregion. For each subregion the integrand in Equ.1 can now be evaluated, and the integrations can be performed. The contribution of stiffeners to the functional $\delta^2\,\Pi$ is established by one-dimensional cubic interpolation. Summation over all subregions yields the total value of $\delta^2\,\Pi$ as a quadratic form in the generalized co-ordinates. In performing these operations, the generalized co-ordinates known already due to boundary conditions, are eliminated from the set of unknowns.

The operation of variation, Equ. 3, is equivalent to differentation with respect to all these variables. It leads to a general matrix eigenvalue problem from which the eigenvalues and the corresponding eigenvectors can be determined. In the programme BEOS simultaneous vector iteration, as described by Schwarz et.al.[4], is used for that purpose. Clearly the procedure described is a special variant of the finite element method. The finite element implied would have as many as 80 degrees of freedom, but establishing it explicitely is avoided in the programme.

Examples. The application of BEOS will be illustrated by means of two examples. The first one, taken from ref. [5], is a rectangular cylindrical shallow sandwich shell with isotropic faces and an orthotropic core, loaded by longitudinal compression and shear.

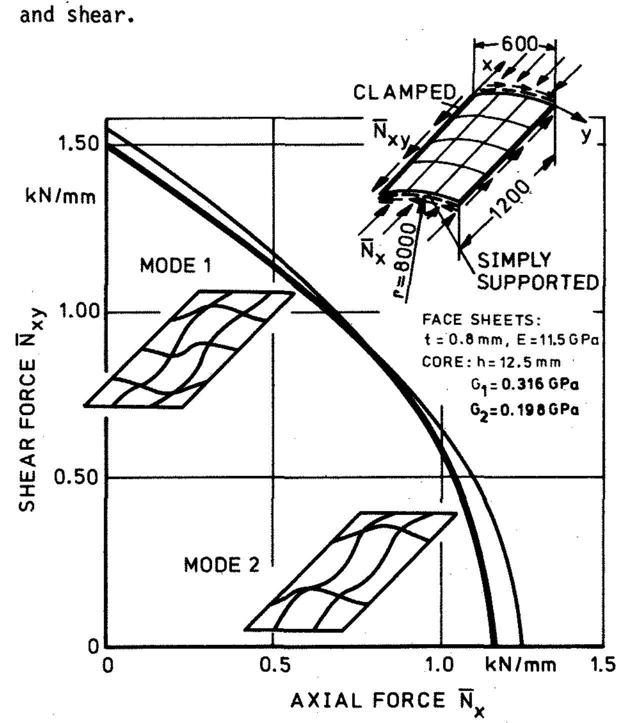
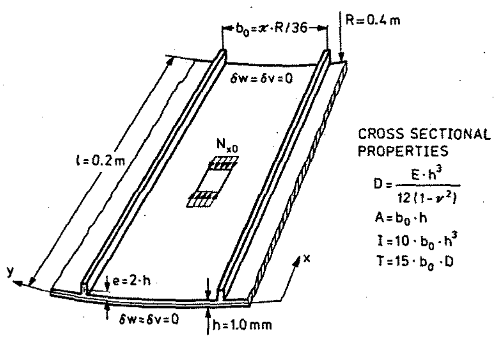
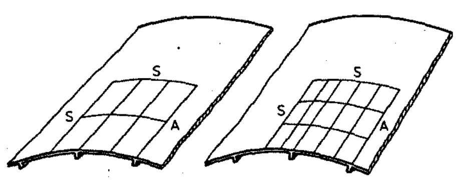


FIG. 3 BUCKLING OF A SANDWICH SHELL SUBJECT TO COMPRESSION AND SHEAR

Fig 3 shows the dimensions of the shell, and the results in the form of an interaction diagram. A close inspection of the results reveals that the interaction curve consists of two separate lines, one connected with the two-lobe buckling mode 1 and the other connected with a three-lobe buckling mode 2. Only the innermost portions of the curves are of interest. At low values of $\bar{N}_x$ mode 1 is relevant. At larger values of $\bar{N}_x$ a switch to mode 2 occurs.

The second example, Fig 4, was taken from ref (3). It served to check the programme with respect to the effects of widely spaced stiffeners. Buckling of an eccentrically stiffened full cylindrical shell subject to axial compression was considered by applying BEOS to a wave panel of the shell. From an analysis of Singer and Haftka(6), the circumferential wave number was known to be n = 12. The panel considered is formed by a part of the shell extending between two consecutive longitudinal nodal lines of the buckling mode. Classical simple support was assumed along the curved edges. As the shell is not of sandwich construction, the option for a Kirchhoff-Love shell analysis with only three functional degrees of freedom was used. Because of symmetry only a quarter of the wave panel was modelled for BEOS with symmetry conditions (S) applied at the midlines, and antisymmetry conditions (A) along the nodal line.





RESULTS	SINGER/ HAFTKA	BEOS 2 × 3	BEOS 3 × 5
$P_{cr} \cdot R/(\pi \cdot D)$	17 809	17 643	17 636

FIG. 4 BUCKLING OF A DISCRETELY STIFFENED CYLINDRICAL SHELL

In the table at the bottom of Fig 4 the eigenvalues computed by BEOS for two different partitions are compared to Singer's and Haftka's results, obtained with a Fourier expansion using 20 Terms for each of the three deflections $\delta u, \delta v, \delta w$. Practically there is no difference between the three results.

III. The Test Programme

Preliminary Considerations

The scatter observed in buckling tests is due to imperfections in shape, boundary support, and load distribution. Shape imperfections can be kept small in relation to the dimensions of the test shell by using not too small test specimens. Many years of experience with buckling tests on thin-walled shells tought us that well-defined unique edge conditions can best be obtained by stiff clamping. But then some uncertainty arises as to the distribution of membrane forces. This distribution must therefore be determined by strain measurements at a large number of points.

The Test Specimens

The test specimens have a quadratic plan view with the fixed dimensions 800 x 800 mm². They are cylindrically curved with the straight generators in the direction of the main compression load. Three-layer sandwich construction was chosen, (see Fig 5). The core consists of 8 mm thick PVC-foam with a density of 50 kg/m³. The face layers are built-up of three thin plies of glass fabric reinforced epoxy. For the inner and outer ply of each face layer a nearly unidirectional fabric was used, with 93% of the fibres oriented parallel to the generators whereas for the intermediate plies bidirectional twill fabric was oriented at 45 degrees.

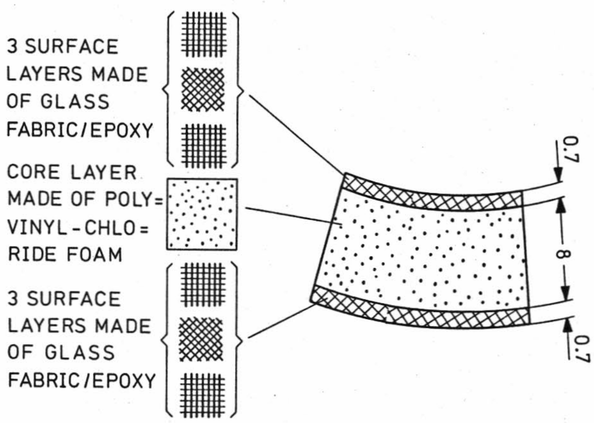


FIG. 5 THE CONSTRUCTION OF THE PANEL

The panels are surrounded by 100 mm wide frames. The face plies are continued into these frames, but the foam core is replaced by chopped fibre mats. At the curved edges mats are added below and above such that the frame thickness is equal to the shell rise.

The test specimens are molded in a die (See Fig 6) the lower and upper parts of which are compressed against each other by clamps. The die is put into an oven for curing at a temparature of 323 K for 15 hours.

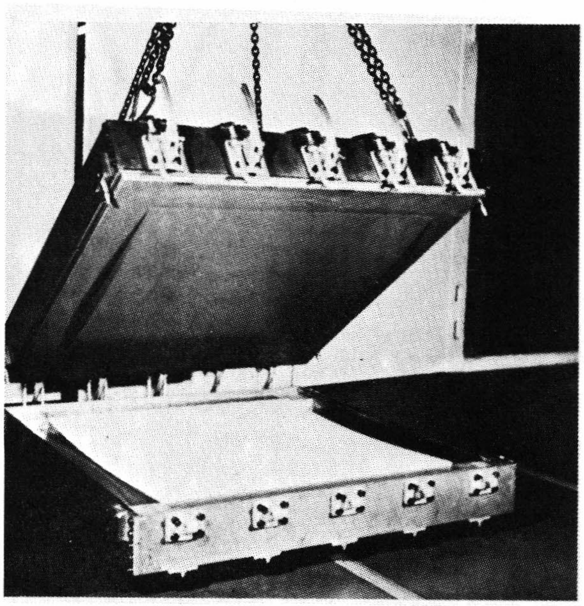


FIG. 6 DIE FOR MOLDING THE PANELS

Measuring of Elastic Properties

The computation of buckling loads requires knowledge of the stiffness matrix **E** in Equ. 1. In principle the components of this matrix can be computed from the geometry and the constituent properties of the sandwich. However, we preferred to determine experimentally the properties of the face layers, the shear stiffness of the core, and the bending stiffnesses of the sandwich in 0°- and 90°-direction. Thus only a few components of the stiffness matrix **E** were not measured directly.

The sandwich faces have to be considered as orthotropic layers. The in-plane elastic properties of such a layer are defined in the law

$$\begin{Bmatrix} \varepsilon_x \\ \varepsilon_y \\ \gamma_{xy} \end{Bmatrix} = \begin{bmatrix} A_{11} & A_{12} & 0 \\ A_{12} & A_{22} & 0 \\ 0 & 0 & A_{33} \end{bmatrix} \begin{Bmatrix} \sigma_x \\ \sigma_y \\ \tau_{xy} \end{Bmatrix}$$

There are four independent coefficients A_{ij} in the compliance matrix, so at least four independent measurements have to be taken. We measured six quantities in four types of tests, getting an opportunity for cross-checking the results. The test types were

1) Tension test (DIN 53455) with load in 0°-direction, strain measurements in 0° and 90° direction
2) Tension test (DIN 53455) with load in 90°-direction, strain measurements in 0° and 90°
3) Shear test (DIN 53399) with a 0°/90°-panel
4) Shear test (DIN 53399) with a ± 45°-panel.

Five complete samples were produced at five different days, each sample consisting of 3 specimens for test types 1 and 2, and 4 specimens for test types 3 and 4. In addition to the tests on the face layers we performed the test types

5) Core shear sitffness test (DIN 53294)
6) Bending test (DIN 53293) in 0°-direction
7) Bending test (DIN 52293) in 90°-direction.

Within each sample 6 tests were performed of type 5, and 3 of each of the types 6 and 7. The scatter observed in these tests was smaller than expected. A detailed report on the tests will be published in the next time.

The Test Facility

The test facility shown in Fig 7 consists of a cantilever box beam subjected to bending and twist by means of two transverse shear forces applied at its free end. The test specimen forms part of the compression flange of the box beam.

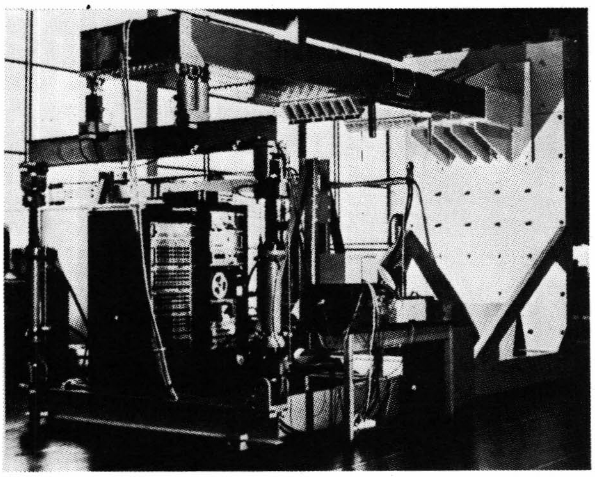


FIG. 7 TEST FACILITY

As such it is loaded in a similar manner as a panel in the wing of an airplane (Fig 8).

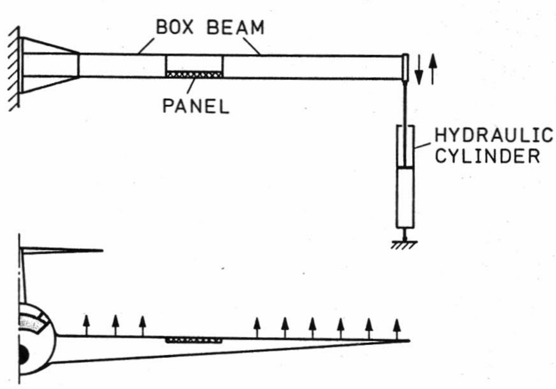


FIG. 8
LOAD APPLICATION TO THE PANEL IN A WING AND IN THE TEST

The test specimen is mounted on the box beam by 120 bolts screwed through the surrounding frame. Moreover, the frame is embedded in a mixture of epoxy and a filler. This mixture hardens after the panel has been bolted. Fig 9 shows the test specimen mounted to the box beam.

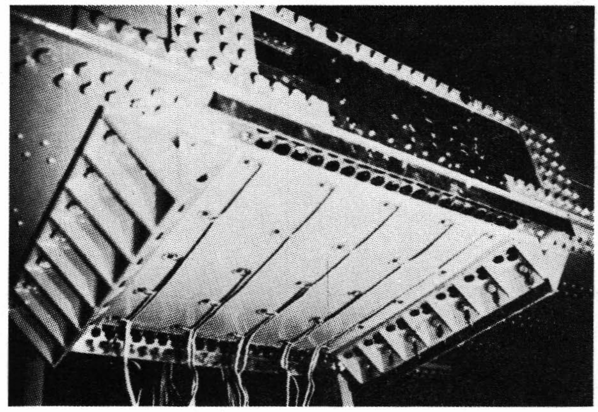


FIG. 9 TEST SPECIMEN AFTER MOUNTING

For load application two hydraulic cylinders with a maximum load capacity of 100 MN are used. They will be controlled in the manner of a master-slave relation. The pressure in the master cylinder is controlled by the operator until it exerts a certain force or deflection, and the slave cylinder is controlled automatically so that a prescribed angle of twist is maintained at the free end of the cantilever box beam.

During the test the load has to be applied in steps in order to have sufficient time to take the strain measurements at constant loading. The resulting transverse load, the torque moment, the transverse deflection and the angle of twist at the free end can be plotted during the test on an x-y-recorder. These values are also digitized together with the strain gauge readings and put out serially on perforated paper tape. Presently 50 rosette strain gauges 0°/45°/90° are applied to each panel, 25 on either side. The first strain measurements are taken before and after mounting of the panel. Thus strains that may arise during mounting can be taken into account in the test evaluation.

Evaluation of Test Data

For evaluation of the test data a computer programme was written. It reads the paper tape with the raw data obtained during testing. Additional information, as stiffness coefficients, calibration factors, and location of strain gauges, is read from a separate input file. The programme computes, for each load step, the resulting transverse load, the transverse deflection, the torque, and the angle of twist at the free end of the box beam, as well as the strains taking into account their initial values due to mounting. Having determined the strains, the programme computes the stresses in the face layers[7].

A complete test log with all measured forces, displacement stresses and strains is compiled. Moreover the programme creates a plot file for graphic representation of the stresses as functions of the deflection of the master hydraulic cylinder. The two stresses determined on opposite face layers at the same location and direction are plotted in one frame. It was expected that this graphic presentation would facilitate the detection of incipient buckling, because the two stress curves should deviate from each other due to increasing bending action, as shown in Fig 10.

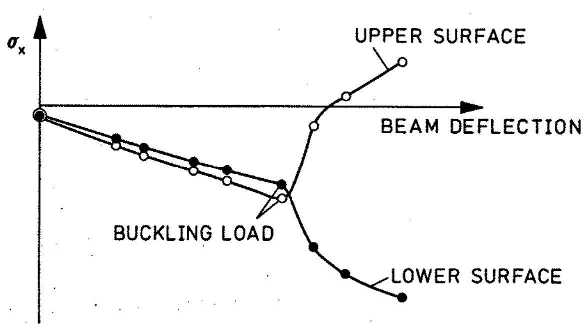


FIG. 10
EXAMPLE OF CORRESPONDING STRESSES AT THE UPPER AND THE LOWER SURFACE

IV. Test Report

Observations and Results

Two specimens could be tested so far. They only differed with respect to their rise, or radius of curvature. The values are given in the following table:

Shell No.	Shell rise	Radius of Curvature
1	25 mm	3212 mm
2	12,5 mm	6406 mm

The cantilever box beam was loaded such that the angle of twist was zero. Thus pure compressive loading of the test specimen was to be expected. The stress measurements revealed, however, that a considerable account of shear was present in addition to compression.

No distinct buckling process with appearance of deep buckles could be observed. The most significant sign was an acoustic one, a slight "bang". Later on, buckles became visible. On the stress plots a buckling load could be identified only with the first shell (See Fig 10). The stress plots of the second test showed only slight irregularities at the point where the "bang" occured. The loading applied to the box beam increased far beyond the buckling load in both tests. Of course, this was to be expected, since stress can be shifted from the buckled shells to the surrounding frame. However, even the buckled panels themselves were able to transmit increased forces. This can easily be concluded from the stress measurements, showing that the mean stresses, or membrane forces, continued to increase.

The following values of the membrane forces just prior to the observed points of buckling were determined from the strain gauge readings:

Shell No 1, Membrane forces in N/mm, x,y in mm.

x \ y	-360	-180	0	+180	+360
-360	N_x = 95,9 N_{xy} = 43,4 N_y = 0.0	92,4 42,7 0,0	91,7 42,0 0,0	90,3 44,1 0,0	91,0 45,5 0,0
-180	63,0 35,0 0,0	84,0 42,0 0,0	85,4 42,0 0,0	81,9 40,6 0,0	71,4 37,1 0,0
0	62,3 31,5 0,0	70,0 37,1 0,0	72,8 40,6 0,0	74,2 37,1 0,0	71,4 29,4 0,0
+180	55,3 30,8 0,0	73,5 37,1 0,0	76,3 40,6 0,0	72,1 33,6 0,0	70,0 26,6 0,0
+360	50,4 17,5 0,0	70,7 25,9 0,0	75,6 26,6 0,0	73,5 28,0 0,0	73,5 28,0 0,0

Shell No 2

x \ y	-360	-180	0	+180	+360
-360	N_x = 44,5	41,6	36,1	37,8	40,3
	N_{xy} = 22,3	17,8	14,4	15,8	17,7
	N_y = 6,9	4,7	5,3	5,0	- 0,7
-180	29,9	33,1	33,8	36,5	30,1
	20,0	19,9	17,3	18,3	13,3
	- 0,6	-1,4	0,1	-0,5	-1,1
0	26,3	32,0	31,7	31,4	24,3
	18,5	21,0	19,0	19,3	12,6
	-1,7	-4,1	-3,3	-4,9	-4,0
+180	22,6	28,6	28,6	30,1	23,9
	15,1	18,6	18,0	17,3	12,9
	-3,5	-4,0	-3,7	-3,1	-1,7
+360	32,4	28,1	28,7	31,5	27,5
	16,8	14,2	13,7	13,8	13,4
	0,5	1,4	1,6	3,2	2,5

Comparison with Theory

The membrane forces measured prior to buckling, were input to the programme BEOS. From the stiffness measurements, the following values of extensional stiffnesses B_{ij} and bending stiffness K_{ij} were determined

B_{11} = 39 743 N/mm B_{12} = 8 577 N/mm

B_{22} = 20 512 " B_{33} = 8 347 "

K_{11} = 657 600 N mm K_{12} = 141 912 N mm

K_{22} = 339 398 " K_{33} = 138 112 "

and used in the computation. The three lowest eigenvalues were computed as factors to be applied to the given distribution of membrane stresses.

For the first specimen the eigenvalues were

$$\lambda_1 = 0.61\ ,\quad \lambda_2 = 0.63\ ,\quad \lambda_3 = 0.84\ .$$

Consequently, due to theoretical predictions, buckling should have occured already at about sixty per cent of the observed buckling load.

After the test the panel was removed from the test facility, but later it was mounted again and tested once more. The stress distribution measured prior to buckling was different from that determined in the first test (and presented in the table). With the new distribution the eigenvalues computed by BEOS were

$$\lambda_1 = 1.09\ ,\quad \lambda_2 = 1.11\ ,\quad \lambda_3 = 1.48\ .$$

This time the agreement between theory and test was nearly perfect.

For the second test specimen the computed eigenvalues were

$$\lambda_1 = 1.22\ ,\quad \lambda_2 = 1.24\ ,\quad \lambda_3 = 1.80.$$

Comparison with the eigenvalues determined for the stress distribution of the first test with test specimen No. 1 reveals the astonishing result that they differ by a factor 2. It has to be suspected that the stress measurements in the first test were in error by this factor. Unfortunately the measurements could not be traced back so far, that this suspicion could be proved, or denied, conclusively.

For the further discussion of the results we dare to be optimistic, assuming that the membrane forces measured in the first test should be halved, so that the computed eigenvalues have to be doubled. We then have the result, that in the computations the buckling load is over-estimated by 9 to 22 per cent, which is very satisfactory in shell buckling.

V. Conclusions

The buckling tests performed so far were satisfactory with respect to the aim of providing experimental support to the theoretical results obtained with the computer programme BEOS. However, they were unsatisfactory with respect to the significance of buckling as a failure phenomenon. This is mainly due to the mild buckling of the tested panels. In Fig 11 the conceptual buckling behaviour of two panels with different curvature is depicted. Their buckling behaviour is totally different. The flat panel has a low buckling load, but it can carry loads far above its bifurcation buckling load. Imperfections will round-off the "knee" so that for realistic shells the theoretical bifurcation buckling load is of no significance at all. The panel will fail eventually by exceeding the material strength or by secondary buckling. In any

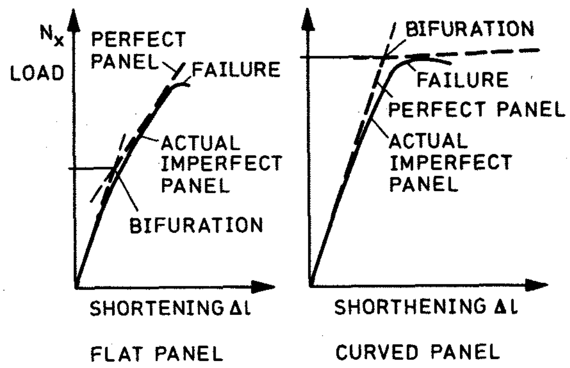


FIG. 11 CONCEPTUAL BUCKLING BEHAVIOUR OF PANELS WITH DIFFERENT CURVATURE

case a theoretical treatment would have to go far into the non-linear postbuckling region, and the prediction of failure would be very uncertain. Tests should therefore be made with specimens the dimensions of which are close to the dimensions of actual structures.

On the other hand, for curved panels that cannot carry considerable loads in the postbuckling region, the bifurcation buckling loads might be useful for estimating the load carrying capacity. Model tests for selected parameters will be sufficient, since a computer programme can be used for the practical structures. But here another phenomenon may limit the applicability of computed buckling loads for failure prediction. If the postbuckling curve descends to steeply, the imperfections may cause the panels to fail at loads considerably below the bifurcation buckling load.

We may conclude, therefore, that a computer programme for bifurcation buckling loads is useful only if these loads may be used to estimate closely the load carrying capacity of the structures. This can be done if the postbuckling equilibrium states do not admit increase of loads far beyond the bifurcation load. It would be worth while to add programme features providing information about the initial postbuckling path by evaluating Koiter's[8] theoryof initial postbuckling behaviour. A comprehensive review of this theory was presented by Budiansky[9]. This theory will yield valuable information as long as the postbuckling paths do not ascend or descend steeply. In this latter case methods for analyzing highly developed postbuckling states would be required.

For the continuation of our test programme it will be necessary to search for shell geometries, which will exhibit a proper postbuckling behaviour. Additional analytical studies will be required for this purpose. The postbuckling behaviour of curved panels[9], and even of sandwich panels[10] has already been analyzed. However, these investigations were disregarded so far, because the boundary conditions and stiffness properties considered in these papers were different from those prevailing in our test specimens.

The tests have confirmed again, that modern mechanics know the means for analyzing properly the structural behaviour of orthotropic sandwich shells. This will be valid also for our next steps, viz. the determination of their behaviour in the postbuckling regime.

VI. References

1. Geier, B., "Die Anwendung des Verfahrens von Rayleigh-Ritz mit maschenweiser Interpolation auf die Berechnung der Beullasten flacher Sandwichschalen". Dt. Luft- und Raumfahrt, DLR-FB 72-05,1972.

2. Geier, B., "Ermittlung der Eigenschwingungen flacher Sandwichschalen mit dem Programm BEOS". Zeitschr. f. Flugwissenschaften Vol.21 (1973) No 12,pp 454-465.

3. Rohwer, K., "Bifurcation Buckling of Shells Using Rayleigh-Ritz Method With Meshwise Hermite Interpolation". Proc. of the Symposium on Applications of Computer Methods in Engineering, Los Angeles, Calif., Aug 23-26,1977.

4. Schwarz, H.R., Rutishauser, H. and Stiefel, E., "Numerik symmetrischer Matrizen". B.G. Teubner, Stuttgart, 1968.

5. Geier, B., "Eine Diskretisierung der Gleichung für die Berechnung der Stabilität orthotroper Sandwichschalen über rechteckigem Grundriß" in "Finite Elemente in der Statik" edited by

Buck, K.E., Scharpf, W., Stein, E, and Wunderlich, W., Verlag Wilhelm Ernst & Sohn, Berlin-München-Düsseldorf, 1973.

6. Singer, J. and Haftka, R., "Buckling of Discretely Stringer-Stiffened Cylindrical Shells and Elastically Restrained Panels" Technion - Israel Inst. of Technol., Deptmt. of Aeron. Engng., Haifa, T.A.E Report No 91, 1974.

7. Klein, H. "Spannungsanalyse mit Dehnungsmeßstreifen an orthotropen Strukturen". DFVLR Internal Report IB 152-77/31, 1977.

8. Koiter, W.T., "On the Stability of Elastic Equilibrium," NASA TT-F10, 833, 1967.

9. Budiansky, B. "Theory of Buckling and Postbuckling Behavior of Elastic Structures". Advances in Applied Mechanics edited by Chia-Shun Yih, Vol. 14, 1974, pp 1-65.

10. Koiter, W.T. "Buckling and Postbuckling Behavior of a Cylindrical Panel under Axial Compression". NLR-Rep. No. S476.

11. Pope, G.G. "The Behaviour in Axial Compression of Slightly-Curved Panels, Including the Effect of Shear Deformability". Int.J. Solids Structures, 1968, Vol. 4, pp 323-340.

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