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C.A.M. Gray

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STRUCTURAL MODEL of CLOSED CIRCULAR "CYLINDRICAL" SHELL for POLUTION-FREE TRANSPORTATION of MATERIALS.

PROFESSOR C.A.M. GRAY
and
DR. G. SINGH

DEPARTMENT of STRUCTURAL ENGINEERING
SCHOOL of CIVIL, MECHANICAL and MINING ENGINEERING.

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SYNOPSIS

1/7.82 scale model of one of the spans of a multispan shell structure, designed to carry, within,a material conveyor and a walkway, was analysed experimentally. This analysis was undertaken on behalf of Australian Iron and Steel Collieries to check the adequacy of their design. This position paper describes the nature of the problem of mathematical analysis, the need for model analysis, the geometric and load simulation, measurement/calculation of strain/stresses and some of the details of further research into the behaviour of this type of structures which is intended in the future. This will include the study of the prototype behaviour.

INTRODUCTION

The basic shell of the prototype structure is to be fabricated with $\frac{1}{8}$ plate Aus-ten 50 rolled to 2440 mm diameter. This closed cylindrical shell is to be stiffened by externally placed and welded diaphragms - 2 x 3" x 3" $\frac{5}{16}$ angles mounted back to back and spaced at 1370 mm intervals. Excepting wind and self load all the design loads due to conveyor and walkway have been arranged to act on stiffening diaphragms; through cross beams placed horizontally inside the thin shell and to lie in the plane of the diaphragms. Internal arrangement of the space is shown in the Figure I. Each of the 19.2 m long shell units is to be simply supported on columns through end diaphragms fabricated from 2 x 6" x 3" channels mounted back to back and welded to the shell through a 12" wide flat ring made from $\frac{1}{2}$" plate. The belt tension in the conveyor is to be taken externally.

For further geometric details and for details of the various initial design criteria like standard sizes of the readily available conveyor equipment, the size of the steel plate as manufactured, the expertise for fabrication and erection available locally, relative costs, corrosion, demountability and transportability etc. see reference I.

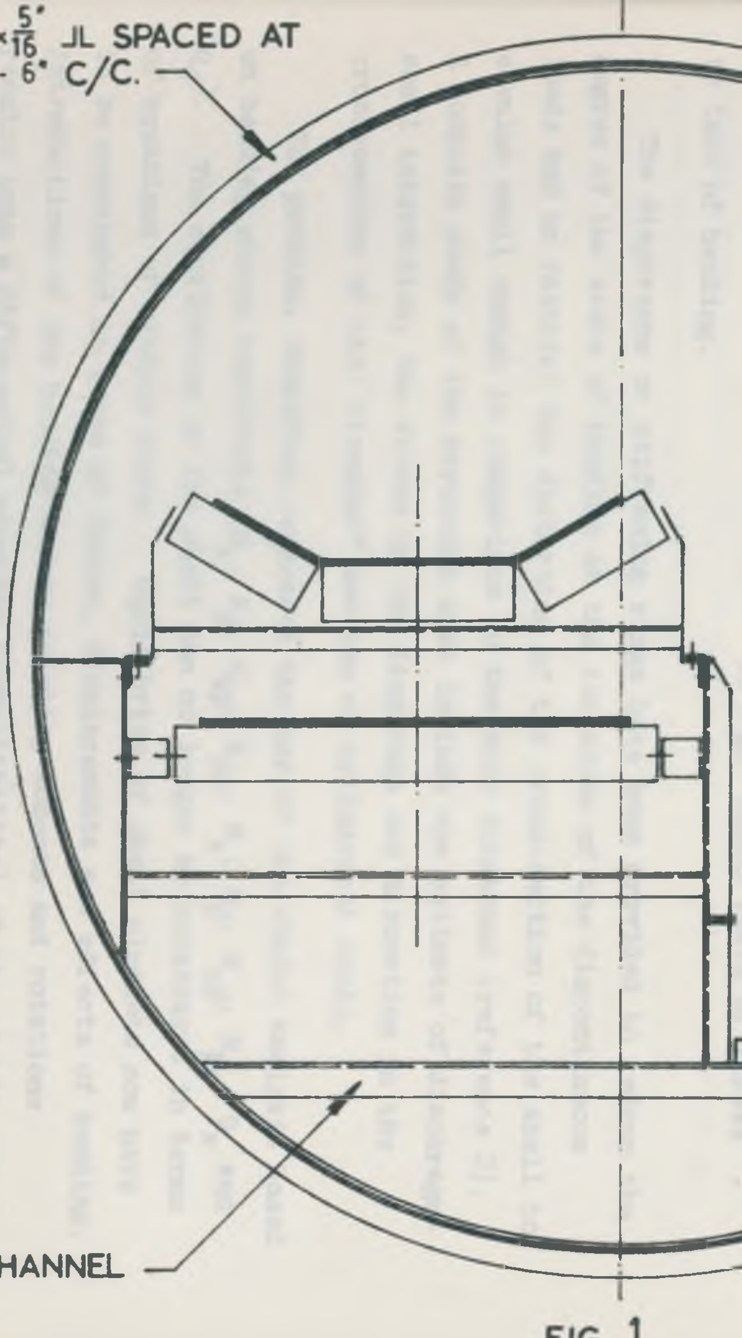
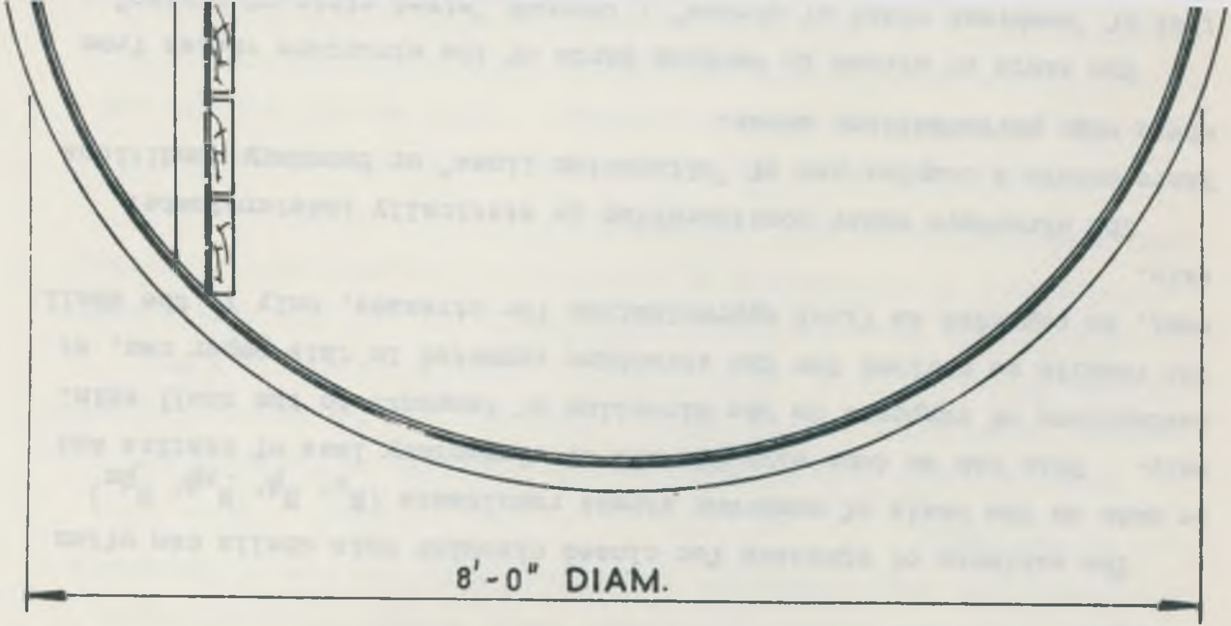


FIG. 1


8'-0" DIAM.

COMPLEXITY OF MATHEMATICAL ANALYSIS AND NEED FOR SCALE MODEL TEST.

The estimate of stresses for closed circular thin shells can often be made on the basis of membrane stress resultants (N_x, N_ϕ, $N_{x\phi}$, $N_{\phi x}$) only. This can be done with the aid of elementary laws of statics and assumptions of supports in the direction of tangents to the shell skin. The results so arrived for the structure reported in this paper can, at best, be regarded as first approximation for stresses, only in the shell skin.

The structure under consideration is statically indeterminate. There exists a complex set of "alteration lines" or boundary conditions where edge perturbations occur.

The state of stress in various parts of the structure varies from that of "membrane state of stress" - through "mixed state of stress" - to that of bending.

The diaphragms or stiffening rings have been provided to reduce the degree of the state of bending at the locations of the discontinuous loads and to restrict the distortion of the cross-section of the shell to a value small enough in comparison to the skin thickness (reference 2). A complete study of the structure must include the estimate of diaphragm-shell interaction, the stress in the diaphragm and distortion in the cross-section of the "circular" section of cylindrical shell.

The problem, therefore, warrants the use of the strict analysis based on bending stress resultants (N_x, N_ϕ, $N_{x\phi}$, $N_{\phi x}$, M_x, M_ϕ, $M_{x\phi}$, $M_{\phi x}$, Q_x and Q_ϕ). The equilibrium of the shell can no longer be considered in terms of equations of statics alone. Equilibrium of shell elements now have to be considered in terms of forces, displacements and effects of bending. Introductions of the bending theory relating moments and rotations results into a differential equation (compatibility) of the eighth order; like that of Schorer's (reference 3)

$$\frac{\partial^8 M_\phi}{\partial \phi^8} + 4a^2\, b^2\, M_\phi = 0$$

Even with the application of the above, simplifying assumptions will have to be made which may produce results where accuracy would be difficult or even impossible to judge without the aid of the model test of the shell.

Model analysis "can be employed for a new design of which no practical experience or test data is available and for which a rigorous mathematical analysis is impracticable. The result of the model analysis in such cases can, most effectively, be used for verification (or otherwise) of assumptions in mathematical analysis and to develop these mathematical methods" (reference 4).

Mathematical study of the state of instability for this type of structure presents even greater problems. This is well brought out through a consideration of even a simple circular cylinder which is assumed to buckle symmetrically under the action of a uniform axial membrane stress (reference 5). The buckling stress for this case as derived mathematically is well known to be

$$\sigma_{cr} = \frac{Et}{r\sqrt{3(1-\mu^2)}} \quad .$$

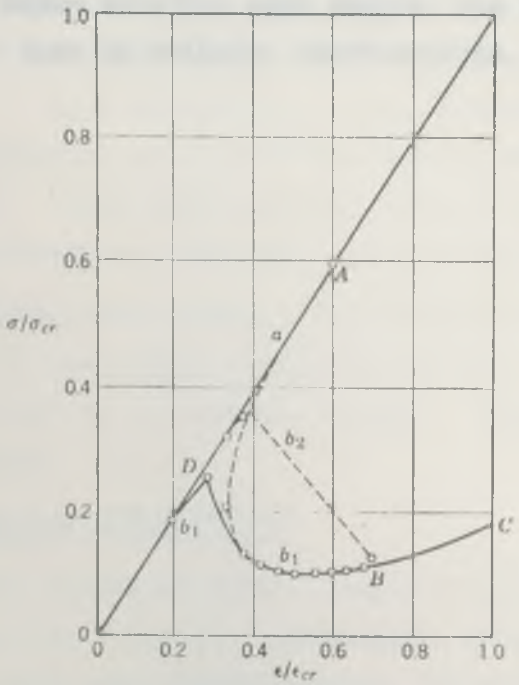


Figure 2. (From Haas)

Figure 2 shows a plot of e/e_{cr} against σ/σ_{cr} where e is axial strain of the cylinder and σ is the stress applied. σ_{cr} and e_{cr} are the critical buckling stress and strain.

The straight line indicates the elastic behaviour, whereas the various relations as observed experimentally are indicated by dotted lines. In all cases the ratio (σ/σ_{cr}) / (e/e_{cr}) becomes considerably smaller than it was before the incidence of buckling. In the past, values of 0.1 have been measured for this ratio. When initial imperfections exist the snap-through buckle occurs earlier than for "perfect" cylinders. The observed ralationship deviates at B and begins to follow the concave curve. The minimum value of the ratio is about the same.

From extensive tests on metal cylinders, quoted in the literature, it appears that the ratio of the actual buckling stress to the theoretical buckling stress varies from 0.1 to 0.6. There seems to be no relationship between buckling load and the buckling pattern which may be used to determine the buckling load from the buckling shape. Since some of the parameters entering the analysis of the buckling loads are less well known even for such shells, the factor of safety used is usually higher than in ordinary constructions.

STRUCTURAL SIMILITUDE

The value and accuracy of the model analysis depends on the degree of simulation achieved for the physical phenomenon being studied.

All the recognized geometric and the considered load variables were carefully simulated to obtain, experimentally, the functions which describe the phenomenon. Buckingham's π - theorem of similarity has been utilised to obtain the indication of the manner in which the structural behaviour depends on each of the independent variables.

A single scale factor of $^{1}/_{7.82}$ was adopted for fabrication of the model. The main criteria affecting the selection of the scale reduction factor were the availability of standard fabrication materials and their workability. The material used has the samv values of E and μ as those of the prototype material. The support conditions have been simulated similarly.

Major part of this investigation has been done for static load conditions. The loading arrangement on the model was designed to simulate the loading on the prototype. Figure 3 shows the internal details of the model. While lead ingots were used as loads, particular care was taken to eliminate shear transfer between the lead ingots and the structure. This was achieved, very satisfactorily, by using grease impregnated synthetic rubber foam as contact medium between the structure and the ingots. These uniformly distributed loads were increased in steps up to the equivalent of five times the working loads (considered).

Further tests were carried out with externally applied concentrated dynamic loads on the central diaphragm and with concentrated static loads on the shell skin in the midspan region. Figure 4 shows the general setup of the test.

INSTRUMENTATION AND COMPUTATIONS

Electrical resistance strain gauges of type "T.M.L. - polyester" were used to measure the strains. Rectangular rosette gauges (R) were bonded to the shell and uniaxial gauges (S) were bonded to the rings at preselected locations. Additional gauges were employed to study the locations which were experimentally found to be of significance. These

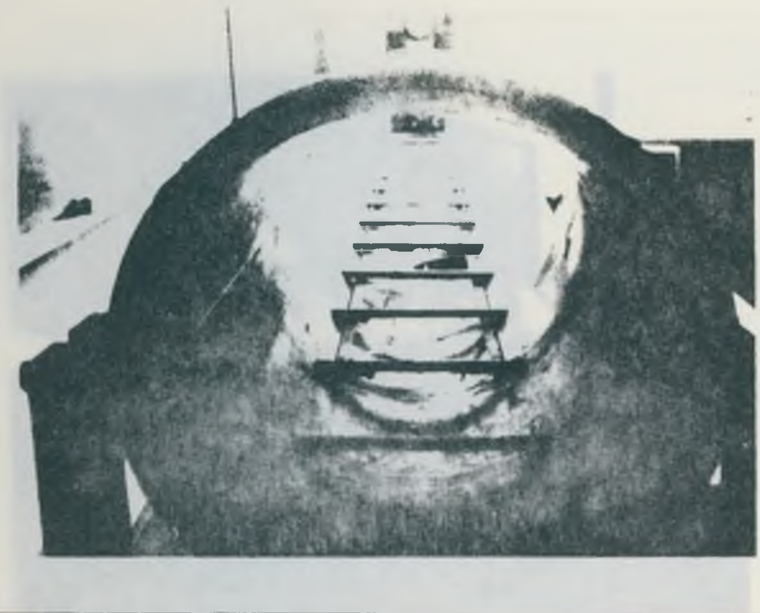


Figure 3.

Figure 4.

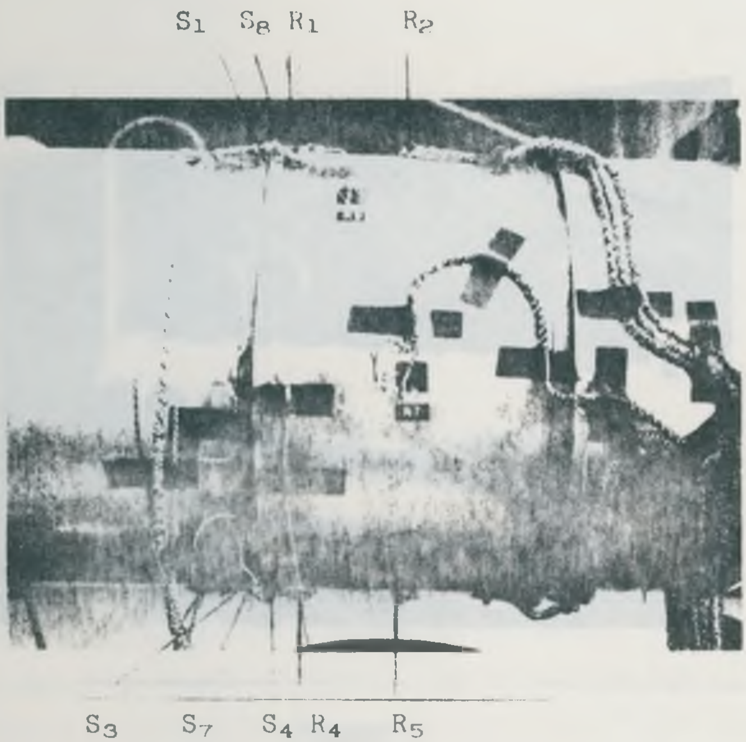


Figure 6.

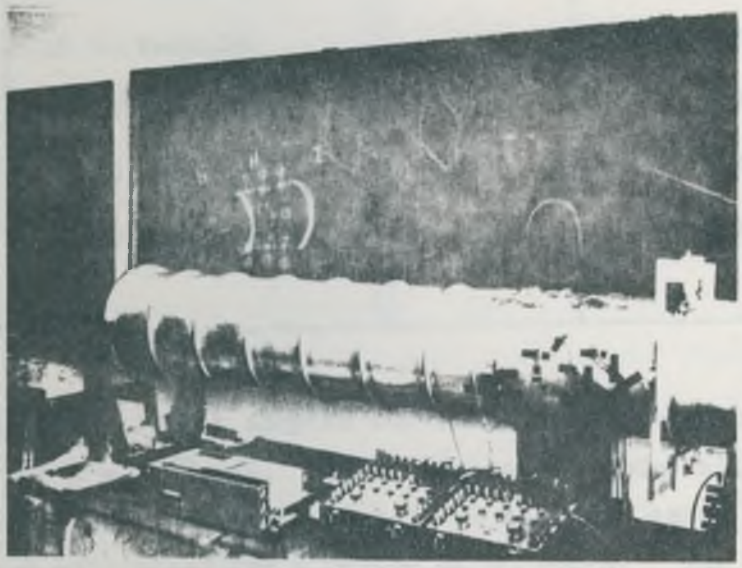


Figure 7.

locations are shown in the Figures 5, 6 and 7.

TABLE I.

LOCATION	STRESS x 10^{-6}(Pa)	LOCATION	STRESS x 10^{-6} (Pa)
R_1	16.53 Compressive	S_1	1.37 Compressive
R_2	20.30 Compressive	S_6	15.15 Tensile
R_4	15.15 Tensile	S_2	3.79 Tensile
R_5	18.60 Tensile	S_5	15.85 Compressive
R_7	3.50 Compressive	S_3	2.76 Compressive
R_8	2.33 Compressive	S_7	12.40 Tensile
R_9	15.50 Compressive	S_4	0.69 Tensile

The measuring circuit adopted was sensitive enough to detect strain of the order of 1 x 10^{-6} m/m. High order of reproducibility was achieved in repeated load tests as well as in those cases where loads on the model were maintained for a period of 60 hours. The drift in the circuit and change in contact resistances in the switching unit (manual) was insignificant. Principal stresses were calculated manually. Table I shows the values of the estimated maximum principal stresses at various locations at an equivalent working load.

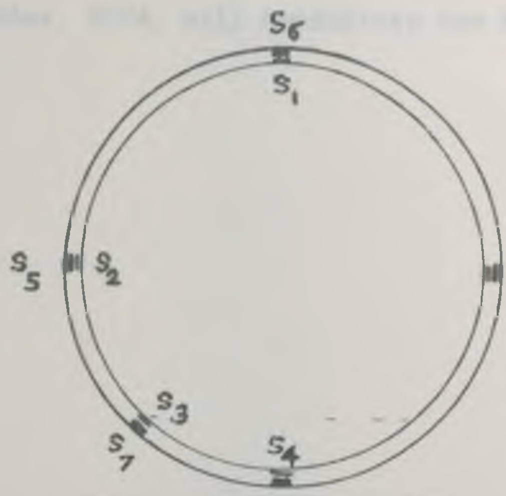


Figure 5.

For further details of maximum/minimum principal stresses computed for up to five times the working load see reference 6.

Very significant changes in the cross-section of the shell and the resulting redistribution of stresses were observed, at higher load levels.

FUTURE RESEARCH

Further research into this type of structure is intended to be carried out in three phases.

A Instrumentation of the first prototype at homologous points to measure strains at various loads considered and to include the effect of self weight and the wind loads. This will be done to quantify the accuracy of the $^{1}/_{7.82}$ scale model.

B. Further instrumentation of the model and other models to measure deflections and strains at frequent intervals (space) to quantify the perturbations and their distribution ("rate of vanish") with a view to develop a mathematical technique of analysis which should make use of models unnecessary for future designs of this kind of structure. This work is thought to be particularly necessary for estimates of the instability of the structure.

C. Study the effects of further such "alteration lines" as openings cut in the skin of the shell.

In the future data acquisition and analysis is intended to be carried out automatically. In the system to be adopted a general purpose mini computer, NOVA, will constitute the heart.

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