25 METER - MILLIMETER WAVE TELESCOPE

10/23/74

REPORT # 5

NRAO Millimeter Wave
Telescope Report ~~No. 7~~ #5
October 15, 1974

Design and Performance of Panel B

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Summary

The design of Panel Structure B (see figure 1) is finished. The weight of one Panel B is 639 lb, and the total contribution to the gravitational telescope deformation is $\Delta z = 0.51$ mil (≤ 0.69 mil is demanded). This includes (a) the sag of the panel structure and (b) of its surface bars, as resulting from the dead load and the surface plates; (c) the panel deformation normal to the surface, from compression or tension forces parallel to the surface, as imposed from the tilting backup structure upon the holding points of the panel; and (d) manufacturing tolerances. Panel B is supported by the backup structure in four homologous holding points, and it gives support to the corners of 12 surface plates in 21 support points along its surface bars. The telescope has 16 Panels B. If all panels of the telescope had the same weight per area, their total weight would be 34,600 lb.

12.75 μm

I. Demanded Accuracy

For $\lambda = 1.2$ mm wavelength, we demand a total deviation, of the surface from the best-fit paraboloid, of $\Delta z \leq \lambda/16 = 2.95$ mil (1 mil $= 10^{-3}$ inch $= 0.0254$ mm). In Report 6 we listed 6 major contributions, and each one then should be $\leq 2.95/\sqrt{6} = 1.20$ mil. Since gravitational deformations split up into three parts (backup, panels, plates), the panels should fulfill

$$\Delta z \leq 1.20/\sqrt{3} = 0.69 \text{ mil.} \tag{1}$$

This total then splits up again into three parts:

1. Sag of panel structure }
2. Sag of surface bars } dead weight + surf. plates (zenith position)
3. Deformation from forces (rms of zenith and horizontal position)

The first item is the sag of the structural surface joints of the panel (points 1 - 9 in figure 2). The second item describes the bending of the surface bars in beam action, in between these surface joints, from their own dead weight plus the weight of the plates they support. The third item results from the forces which are imposed by the telescope structure when tilted, through the holding points, onto the panel structure. Each item then should have $\Delta z \leq 0.69/\sqrt{3} = 0.40$ mil. In addition, we must investigate the influence of manufacturing tolerances, of bar areas and joint coordinates.

The sag is always down, and a parallel sag would be homologous and unimportant. We thus subtract the average sag (now of panel B, but later we must use the average sag of <u>all</u> panels on the telescope). Calling $s = \Delta z$ of the sag, and s its average, we then demand for the first two items:

$$\mathrm{rms}(s - \bar{s}) = 0.40 \text{ mil}. \tag{2}$$

But the telescope forces may change sign during tilt, and the resulting deformations can go either way. Calling ϕ the force deformation, we cannot subtract an average and demand

$$\mathrm{rms}(\phi) = 0.40 \text{ mil}. \tag{3}$$

We investigated four different forces on Panel B, see figure 6. Their amount in pounds comes from a telescope computer analysis made by W. Y. Wong. Panels B may have four different locations on the telescope (figure 1), and the telescope was analyzed in two positions, zenith and horizon. We thus get eight values for force F_2, for example, and the value actually used is their <u>rms</u>;

since we always have either one of them; they do not add up. But the forces F_1 - F_4 in the single panel do add up. Calling f_i the rms deformation of the 9 surface points resulting from force F_i, and since F_1 and F_4 actually occur twice on the panel, we finally use

$$\mathrm{rms}(\phi) = \sqrt{2f_1^2 + f_2^2 + f_3^2 + 2f_4^2}\ . \tag{4}$$

We may, of course, violate equation (2) or (3), as long as equation (1) is fulfilled for the total.

II. Design Features

Figure 1 shows the position of Panel B on the telescope. Figures 2, 3, and 4 show the internal panel structure; note that the four panel corners are supported by a soft beam action for allowing them the same sag as the interior points have. Figure 5 gives the surface bars and their loading condition, figure 6 the external forces F_i. Panel B has a total of

26 joints,
61 bars, (5)
7 groups of bar areas.

Bar areas are identical within each group, and different from one group to the other. Except for the surface bars, all bars are round pipes, with areas in $0.177 \leq A \leq 0.455$ inch. All stresses are so small, ≤ 0.3 ksi (no survival loads in radome), that all pipes are completely determined by the maximum allowed slenderness ratio (against buckling) of

$$l/r = 200 \tag{6}$$

and by the minimum allowed wall thickness (for welding) of

$$t \geq 0.1 \text{ inch}, \quad ? \tag{7}$$

which makes all bars heavily overdesigned for their loads.

The surface bars are rectangular tubing, of 5 x 3 inch, with A = 2.74 inch^2; this size is needed regarding their sag in beam action under dead loads and surface plates in zenith position. The 65-m design had only 4 x 2 inch with A = 2.02 inch^2, but now we need a higher accuracy for λ = 1.2 mm.

In general, we have two degrees of freedom in our design for fulfilling demands (2) and (3): changes of geometry and of bar areas. Fortunately, the latter was not needed during the optimization of Panel B. It thus has the smallest possible weight compatible with conditions (6) and (7). The weight is for one panel

$$W = 639 \text{ lb.} \tag{8}$$

The area of one panel is

$$A = 108.3 \text{ ft}^2 \tag{9}$$

which gives

$$w = 5.92 \text{ lb/ft}^2 \tag{10}$$

as compared to 7.5 lb/ft^2 for the panels of the 65-m design. Result (10) may also be compared to 3.8 lb/ft^2 for the surface plates, and to 11.05 lb/ft^2 for the backup structure. If all other panels would have the same weight/area as Panel B, then the whole telescope (5845 ft^2 of curved area) would have 34.6 kip for all its panel structures. Actually, Panel A must be heavier (longer bars, same l/r), while Panels C and D may be lighter. As a conservative estimate, one may use 39 kip for the total.

A further condition for the panel design is a limit on their depth (normal to the surface), such that none of the downward panel bars will interfere with any of the members of the telescope backup structure. This is checked with a special program written by W. Y. Wong, and the present final design of Panel B fulfills this demand.

III. The Performance

The structural sag in zenith position, from dead loads and surface plates, is $\bar{s}$ = 2.47 mil in the average for the nine surface joints, with

$$\mathrm{rms}(s - \bar{s}) = 0.316 \text{ mil}$$

fulfilling demand (2) by a good margin. Weighted average and rms are taken with the area supported by each point as the weight.

The additional sag of the surface bars under dead loads and plates (figure 5) was calculated according to the formulas given in the Steel Construction Manual, pages 2-123 and 124, for distributed (dead) loads and for concentrated (plate) loads. Results are given in table 1.

Using the supported area as weight, the weighted average is $\bar{s}$ = 0.405 mil, with

$$\mathrm{rms}(s - \bar{s}) = 0.351 \text{ mil} \tag{12}$$

again fulfilling demand (3) by a nice margin.

Table 1. Sag of Surface Bars Under Beam Action

line	points	length l (inch)	load P (lb)	sag Δz (mil) Δz_1	Δz_2	supported area (ft^2)
back	7-8	76	19.3	.670	.957	10.2
middle	4-5	65	34.3	.551	.776	18.1
front	1-2	54	15.0	.180	.255	7.9
supporting structural joints				0.0		36.1

The force deformations, f_i, of the surface points in z-direction, as explained before equation (4), are given in table 2 together with the forces F_i. Each f_i is again the weighted value, $f_i = \text{rms}(\Delta z)$, over the nine surface points. Their quadratic addition yields a nice low value as compared to demand (3):

$$\text{rms}(\phi) = \sqrt{2f_1^2 + f_2^2 + f_3^2 + 2f_4^2} = 0.199 \text{ mil.} \tag{13}$$

Table 2. Deformations f_i from Telescope Forces F_i.

i	line	points	force F_i (lb)	deformation f_i (mil)
1	diagonal	11-19	331	.058
2	front	11-13	296	.057
3	back	17-19	424	.074
4	side	11-17	347	.110

Finally, the combined total deformation, of structural sag, surface bar sag, and force deformation is

$$\Delta z = \sqrt{0.316^2 + 0.351^2 + 0.199^2} = 0.513 \text{ mil,} \tag{14}$$

which fulfills the combined demand (1), $\Delta z \leq 0.69$ mil, by a good margin.

Manufacturing tolerances should be considered, too. Bar areas of commercially available pipes may be off their nominal value by

$$A \pm 5\%, \text{ max.} \tag{15}$$

And during the manufacture of a panel structure, we may specify that each coordinate (x, y, z) of each of the 26 joints shall not deviate from the design value by more than

$$x, y, z \pm 1/8 \text{ inch, max.} \tag{16}$$

The effect of these deviations on the resulting performance was investigated in an additional computer run. Using equal-distributed random numbers, all bar areas and joint coordinates of our final design were changed according to (15) and (16), and this changed structure then was analyzed the same way as described above. The result is, instead of equation (14),

$$\Delta z = 0.514 \text{ mil.} \tag{17}$$

ONE QUARTER OF TELESCOPE SURFACE WITH PANELS

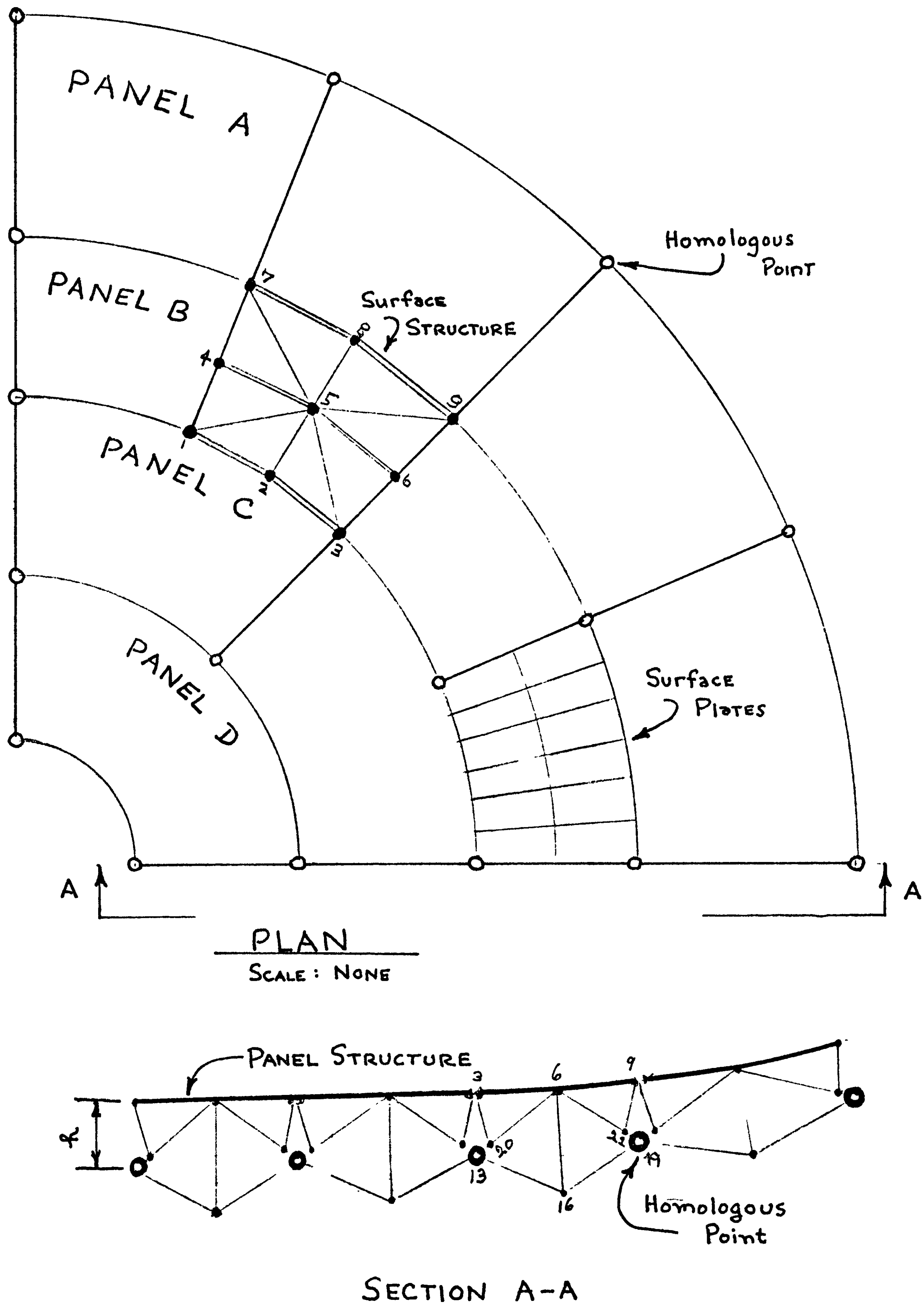


Fig. 1 Location of panels A, B, C, D, on one quarter of the telescope.

K-KERN

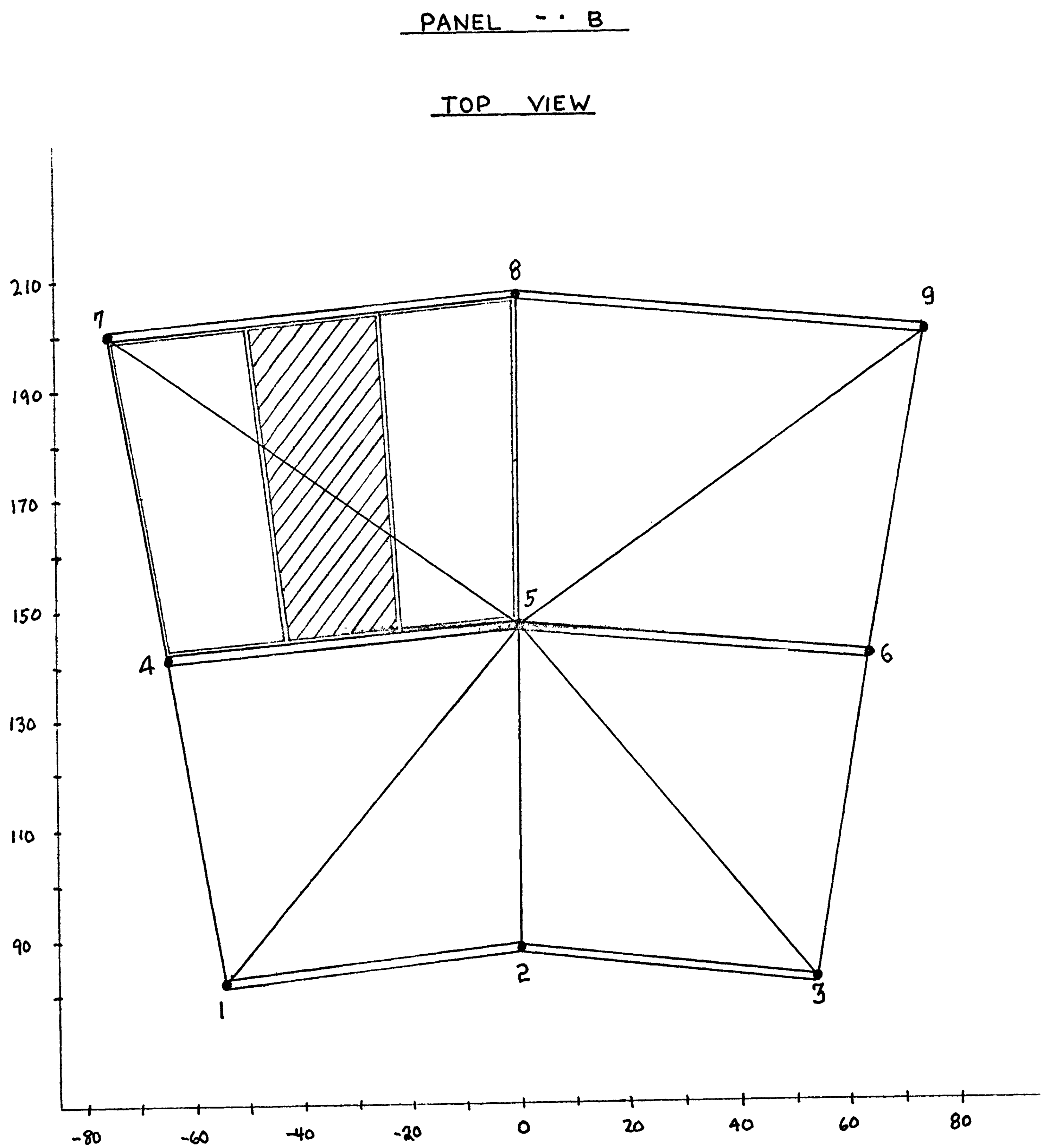
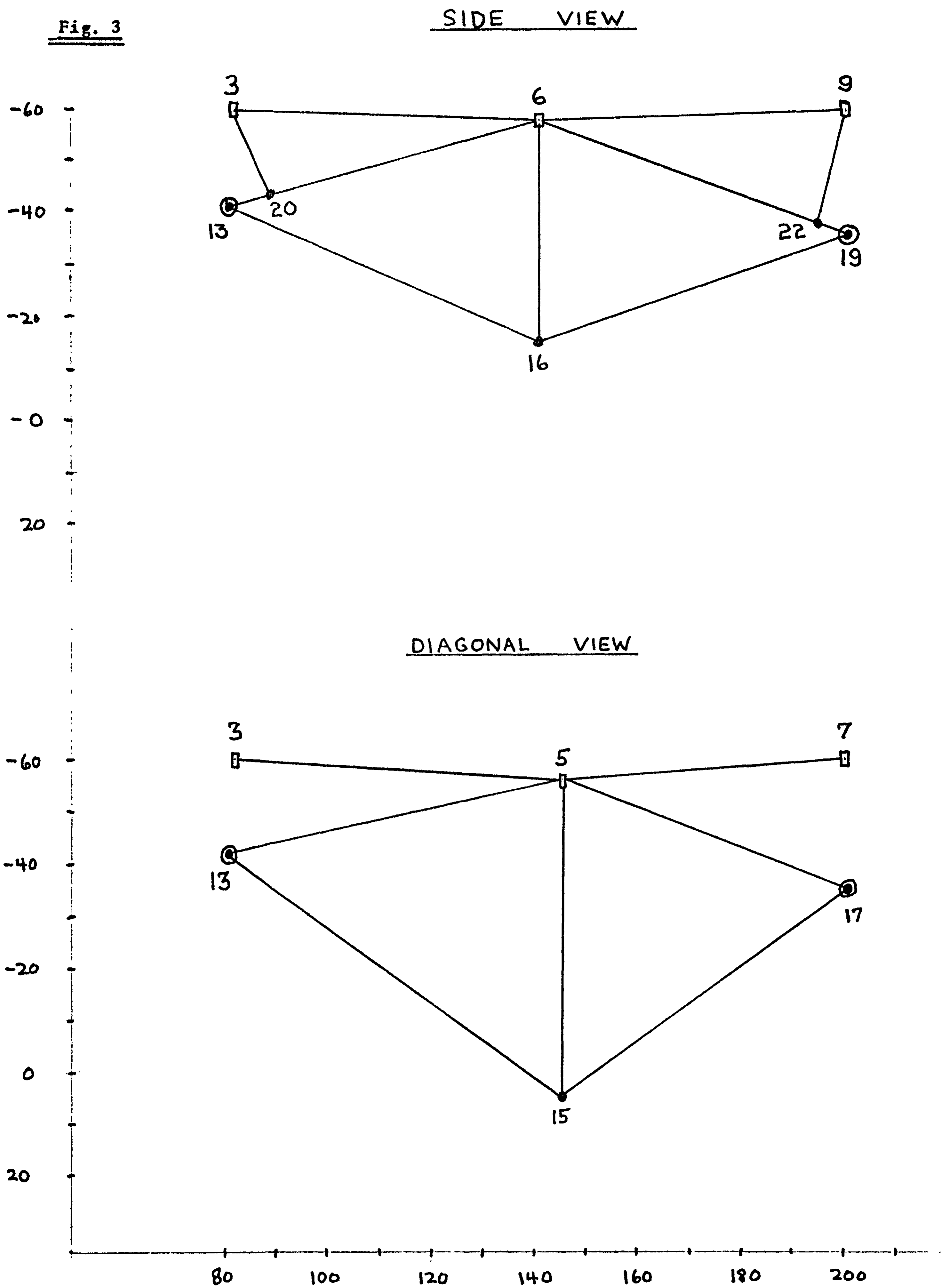


Fig. 2 Top view of panel B, with joints numbered. Double lines indicate the surface bars which support the plates. One plate is shown.

Fig. 3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DIAGONAL VIEW
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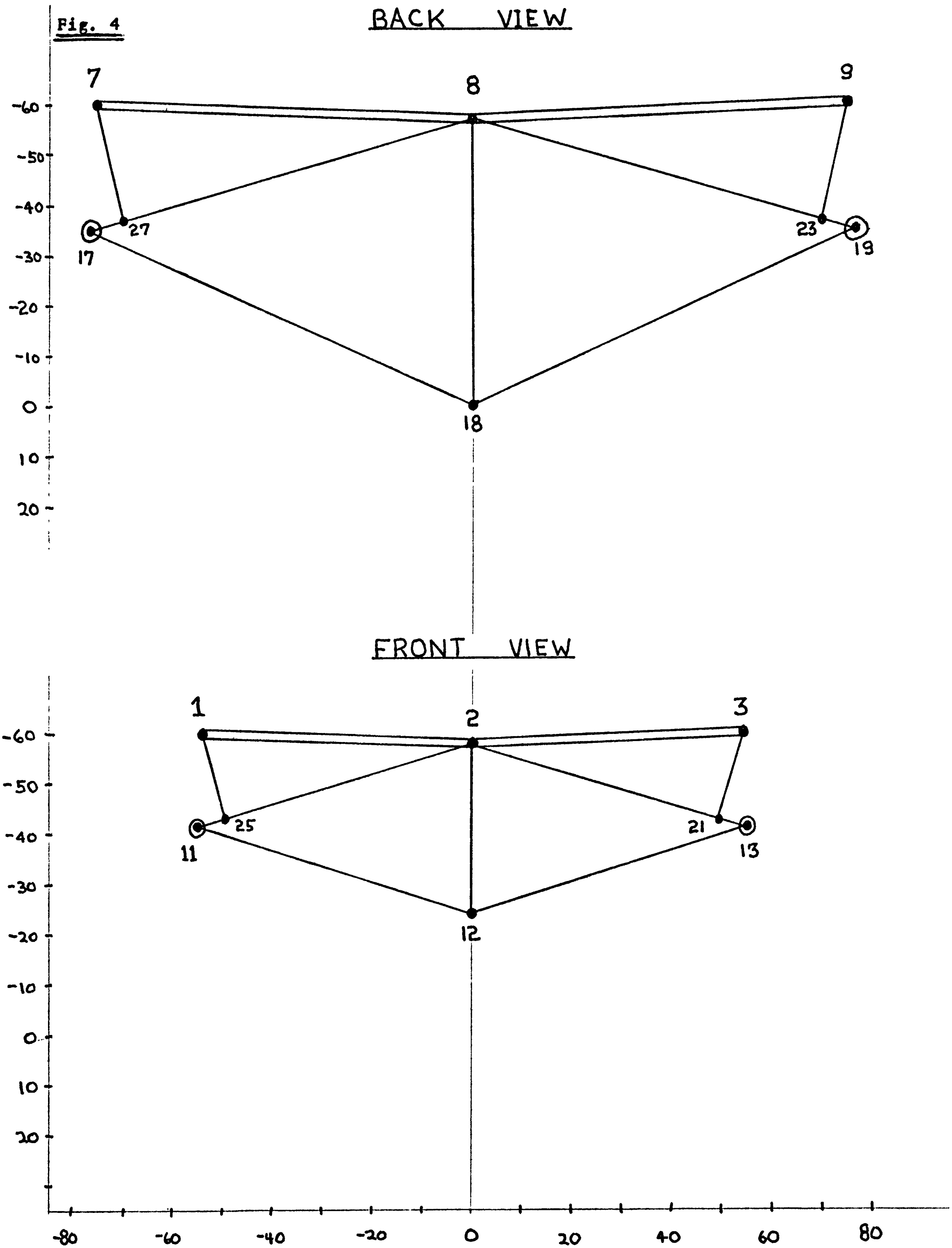
Fig. 4


BACK VIEW
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FRONT VIEW
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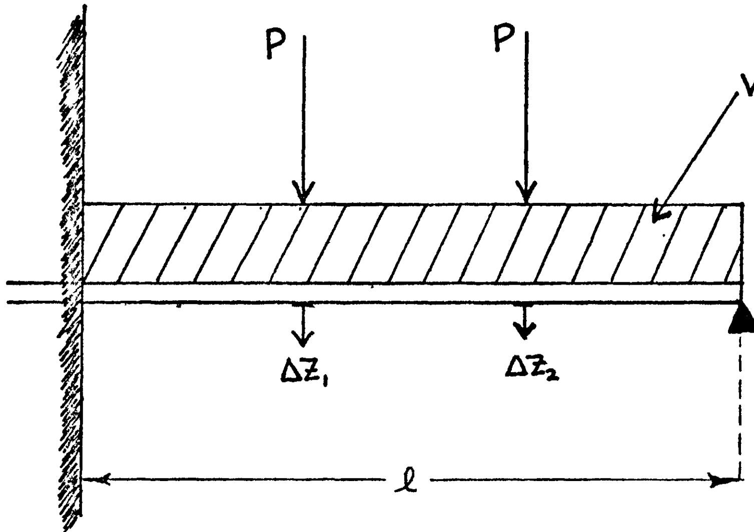

Fig. 5 The surface bars (5 x 3 inch rectangular tubing) and their loading condition; w = dead weight of the bars, P = weight of supported plate corners, ΔZ = resulting sag (table 1).

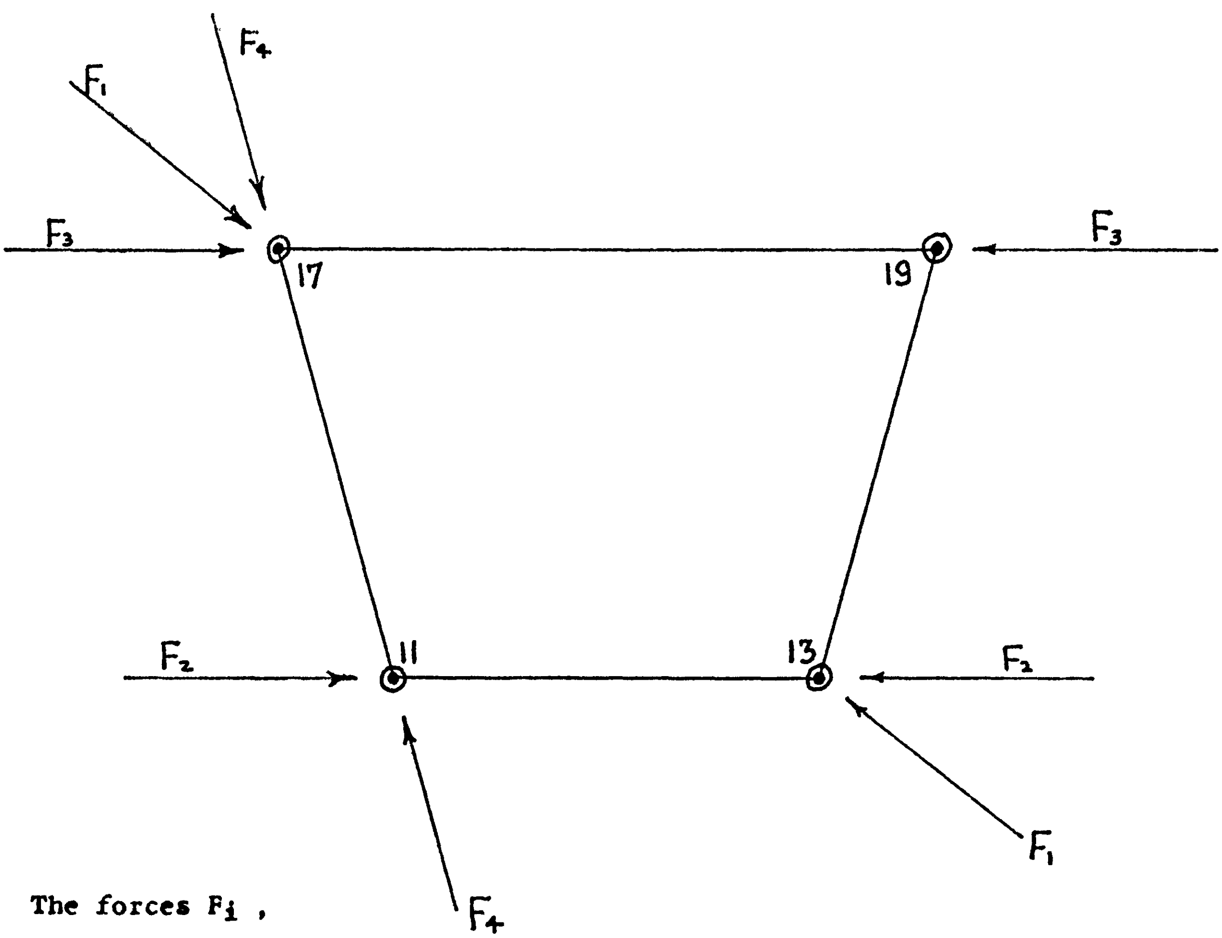

Fig. 6 The forces F_i, imposed by the tilting telescope onto the four holding points of the panel (table 2).