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Report

INVESTIGATION OF A HYDROSTATIC AZIMUTH THRUST BEARING FOR A LARGE STEERABLE ANTENNA

by

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FOREWORD

This report was prepared by the Franklin Institute Research Laboratories under JPL Contract No. 953457 (FIRL Project Number 32G-C3388-01). This report covers work conducted from 17 May 1972 - 27 October 1972.

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The principal investigators at FIRL were Mr. H. C. Rippel and Mr. J. H. Rumbarger (FIRL Program Manager). Other investigators who contributed to the study at FIRL were Mr. James Dunfee and Mr. Frank Kramberger. Prof. V. Castelli, Columbia University, acted as a consultant to FIRL and made significant contributions to the program as a principal investigator.

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1. ABSTRACT

The problems inherent in the design and construction of a hydrostatic azimuth thrust bearing for a tracking antenna of very large size were studied. For a load of 48,000,000 lbs., it is concluded that the hydrostatic bearing concept is feasible, provided that a particular multiple pad arrangement, high oil viscosity, and a particular load spreading arrangement are used. Presently available computer programs and techniques are deemed to be adequate for a good portion of the design job but new integrated programs will have to be developed in the area of the computation of the deflections of the supporting bearing structure. Experimental studies might also be indicated to ascertain the life characteristics of grouting under cyclic loading, and the optimization of hydraulic circuits and pipe sizes to insure the long life operation of pumps with high viscosity oil while avoiding cavitation.

2. INTRODUCTION

The specification of a 48,000,000 lb. dead weight load for the antenna under discussion was obtained by doubling the dimensions of the already existing AAS antenna of 210-feet diameter. A first estimate shows that the load should increase by a factor of $2^3 = 8$. When investigating the feasibility of carrying such an enormous load it is very important to predict whether the estimated load of 48,000,000 lbs. is likely to increase or decrease when the actual design calculations are carried out. Antennae are structures composed of beams. These beams are loaded in either bending or compression and tension. Given the material and type of load distribution, the deflection due to being loaded in bending is given by

$$d \propto \frac{W\ell^3}{I} \qquad (1)$$

where

W = total load
ℓ = beam length
I = section modulus

If the load is produced by the weight of the beam itself, and if we are comparing the deflection of beams similar in shape, then formula (1) reduces to

$$d \propto \ell^2 \qquad (2)$$

(Deflection is proportional to square of length)

For beams loaded in either tension or compression the actual deflections are given by

$$d \propto \frac{W\ell}{A} \qquad (3)$$

where

A = cross sect. area

If again we are comparing the effect of the beam load on beams of similar geometry, formula (3) becomes

$$d \propto \ell^2 \qquad (4)$$

Equation (4) shows that for tension and compression also, the deflections are proportional to the square of element length. This result can also be obtained from dimensional analysis. In similar structures where the value of Young's Modulus, E, is sufficient the elastic properties:

$$d = f(E, \ell, \gamma) \qquad (5)$$

where

γ is the weight density of the material.

This leads to:

$$\frac{d}{\ell} = g\left(\frac{\gamma\ell}{E}\right) \qquad (6)$$

and since the problem is linear in, E,:

$$\frac{d}{\ell} \propto \frac{\gamma\ell}{E} \qquad (7)$$

This indicates that to scale the 210-foot diameter AAS antenna geometry up to a 420-foot diameter dish, the deflection would not simply be doubled, but quadrupled. Therefore, even if the specifications on the accuracy of the 420-foot diameter dishes were relaxed in the future, it is unlikely that a dramatic reduction in weight would result, unless the quality of the dish itself were to be degraded by a large factor compared to the 210-foot dishes.

Further reasoning indicates that no great optimism should be used when estimating the feasibility of the antenna bearing. Scaling of the hydrostatic bearing from the 210-foot diameter antenna is not possible. The load has increased by a factor of 8, but the area of the bearing has only increased by a factor of 4. Simple scaling would therefore cause operating pump pressures twice as high as those at the present site. Grout and foundation concrete loadings would be double that presently incurred by the existing structures. Moreover, even if the doubled pressures were acceptable, the ensuing flows would be 16 times higher than those of the present bearings, based on a doubling of the film thickness. The present configuration of 6 pumps delivering 8 GPM each would become a system of 6 pumps delivering 128 GPM each.

It is obvious from the preliminary statements above that some re-thinking is necessary in the design philosophy and approach to the hydrostatic bearing for the 420-foot diameter dishes. Other considerations also indicate the desirability of improving upon the design concepts used for the 210-foot diameter dishes. Grout disintegration and leaking from the oil reservoir are chronic problems presently being experienced.

Whatever the design approach, the dimensions of the bearings for the new antenna must be greatly increased. This requires review of the analytical methods which were utilized for the calculation of runner deflections and pad deflections in the case of the 210-foot antenna. Further study may be necessary in order to improve the design in other areas for the future antenna, although they have not given any trouble in the already existing configuration. An example of this is the runner-joint design, another is the hydraulic system, yet another is the aligning mechanism, etc.

3. CONFIGURATION SELECTION

The main consideration in designing the bearings is that of carrying the required load while providing adequate stiffness of the system. In the case of an antenna, these stiffness requirements are dictated by wind loads, imperfections in the runner geometry, etc., but mostly by the resonant frequency of the structure having to be above a certain pre-established limit. For the purpose of establishing orders of magnitude and approximate values of the stiffness, two main modes of vibrations can be considered to be affected by the hydrostatic bearing assembly. The first, a mode of vertical translation; the second, a mode of rocking about an axis on the plane of the bearing.

For the translation modes, the stiffness requirements are given by:

$$K_T = 4\pi^2 f^2 m \quad \text{(lbs/ft)} \tag{8}$$

where

K_T = vertical stiffness of the azimuth bearing (lbs/ft.)
f = frequency requirement in cycles per unit time
m = antenna mass (lbs-sec^2/ft.)

The stiffness requirements dictated by the rocking mode are given by:

$$K_r = 4\pi^2 f^2 H^2 m \quad \text{(ft.-lbs/radian)} \tag{9}$$

where

H = effective height of the center of mass above the plane of the bearing (ft.)

R = radius of the hydrostatic bearing (ft.)

Formula (9) is based on the assumption that the moment of inertia of the antenna about an axis on the plane of the bearing is mostly due to the mass of the antenna moving as a point mass at the distance H from the axis of rotation. This assumption improves with the lengthening of the distance H. The rotating mass of the antenna can be roughly distributed as one-half of the mass acting at the elevation axis and one-half of the mass acting at 30% of the height to the elevation axis above the plane of the bearing. The height of the elevation axis is approximately equal to the diameter of the azimuth bearing. H can then be estimated in terms of the bearing radius as:

$$mH^2 \approx \frac{m}{2}(2R)^2 + \frac{m}{2}(0.6R)^2 \qquad (10)$$

$$H \approx \sqrt{2R^2 + .18R^2} \approx 1.48R$$

The rocking mode stiffness, K_r, can be expressed in terms of the vertical pad stiffness, K_T, and the bearing radius, R, for the three-pad geometry as:

$$K_r = \frac{K_T R^2}{2} \quad \text{(ft-lbs/radian)} \qquad (11)$$

The ratio of the rocking mode stiffness, K_r of EQ (9) for a specified frequency, f, to the equivalent rocking mode stiffness (EQ 11) using the vertical translation stiffness, K_T of EQ 8 for the same frequency will give an indication of the relative values of the two modes

$$\frac{2K_r}{K_T R^2} = \frac{2H^2}{R^2} \approx 4.38 \qquad (12)$$

This means that the vertical stiffness requirement of the azimuth hydrostatic bearing pad for a specified frequency is 4.38 times greater when considering the rocking mode than when considering the vertical translation mode.

A rocking mode stiffness of 2.53×10^{12} ft-lbs/radian (equivalent vertical pad stiffness of 1.03×10^9 lb/ft.) would be required to meet a 2 cps frequency requirement and with R = 70 feet. This can be considered to be an upper limit of the required stiffness since it is based upon high estimates for H and f. The required vertical pad stiffness of 1.03×10^9 lbs/ft is much lower (by at least one order of magnitude) than can be normally obtained from oil film hydrostatic bearings. Therefore the stiffness requirement determines the limiting deflections of the load distributing and equalizing mechanisms which normally surround the oil film bearings. Total allowable deflection based on the 1.03×10^9 lbs/ft stiffness is 0.56 inches. In the evaluation of this total deflection budget, it should be considered that the runner supports should offer a compliance such as to account for 0.1 inches of deflection. This leaves about 0.46 inches available for the alignment mechanism.

3.1 PADS ON THE ROTOR VS. PADS ON THE STATOR

The bearing configuration consists of pads of finite extent which ride on a 360° runner. To put the runner on the moving part and the pads on the static part (directly on the foundation) has been suggested as a means of maintaining the load on each of the pads and each of the segments of the foundation constant regardless of the motion of the antenna. This is indeed desirable since it would eliminate some of the problems of deterioration of the grout attributable to the cyclic variation of load level. One possible configuration utilizing this concept is illustrated in Figure 1. In this case, three support points establish a plane assuring that for any aligning condition the loads on the foundation remain constant at all times except for the effect of wind. However, the load of the antenna dish is supported essentially at the two elevation bearing points and cannot be easily spread around a 360° circle. The runner surface cannot be much closer to the elevation bearing than as shown in Figure 1 so it is almost impossible to conceive of an economic structure which would provide the 360° runner with the stiffness necessary when the bottom of the runner is at the required elevation. In addition the runner leveling procedures and bearing oil containment would create quite considerable problems.

This situation would not be relieved by increasing the number of load carrying pads since the stiffness distribution of the runner would have to be made more uniform in order to keep all these pads uniformly and constantly loaded.

In summary, the approach of mounting the runner on the moving part of the antenna results in difficult technical problems of load distribution, which in general require that the static pedestal weight is now added to the moving part. The additional drawbacks of difficulty of oil containment and runner leveling make this design approach unreasonable so that it will not be pursued any further.

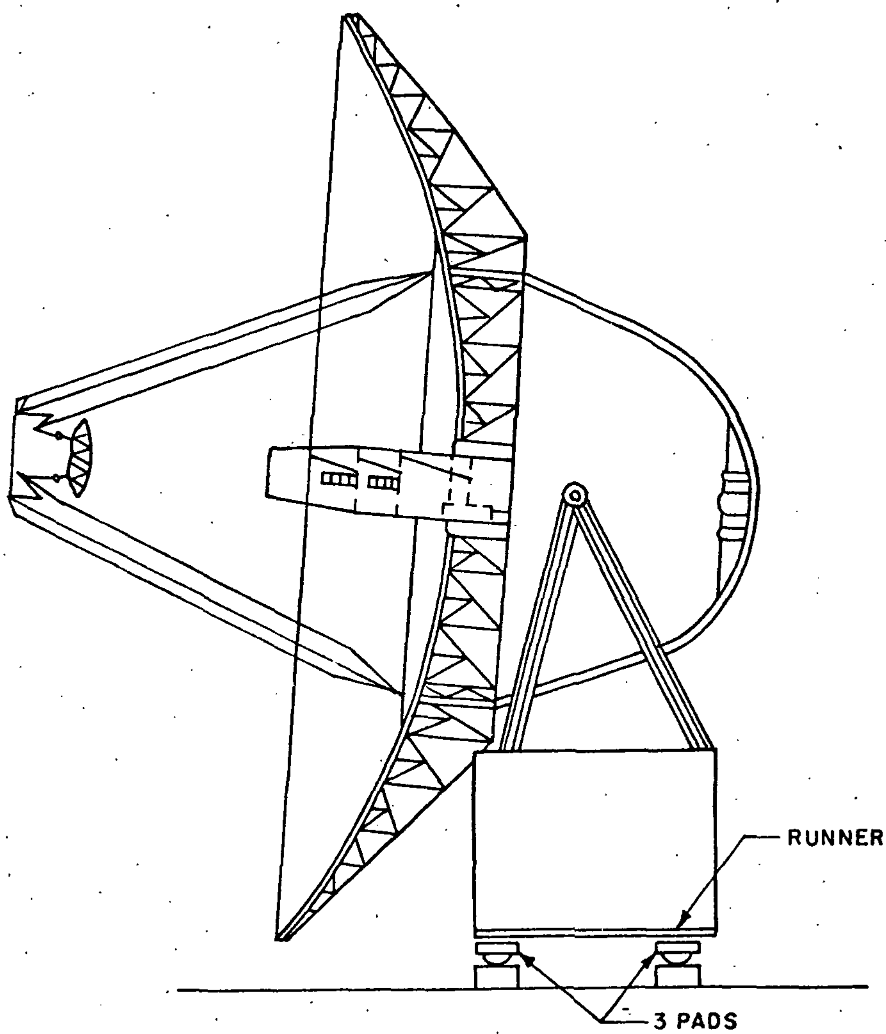

Figure 1. Runner on the Rotor Configuration

3.2 WHEELS VS. OIL PADS

A wheel and rail design, although appealing, is quite impossible due to the fact that the maximum allowable loads on a wheel running on a rail is of the order of 150,000 lbs. Considering that there are 3 supports, each of which has to carry 16 million lbs., the 420 ft. antenna would require approximately 100 wheels for each alidade support. The 100 wheels could be arranged on several trucks running on multiple rails. Equalizing and distributing the load amongst all the wheels and all the rails without exceeding the allocation of 0.3 in. static deflection in the alignment mechanism is a formidable task and most probably quite impossible to realize with certainty. One arrangement that could be considered for a wheel-rail design is illustrated by Figures 2 and 3. Each alidade support is on 96 wheels, subdivided into 3 major trucks, each of which contains 2 minor trucks. Alignment is provided by 3 grease joints per truck. Most of the stiffness is obtained through the center truck but is obtained only if the stiffness beam carrying the 2 minor trucks is stiff enough. The load is shared among the 3 trucks by preloading the beam which extends from the alidade support approximately 12 ft. in each direction. The depth of this beam should be in the order of 6 to 10 feet, allowing it to carry 5 million lbs. at each end, though the deflection is of the order of 1/2 inch to a one inch. These lateral trucks are therefore preloaded by means of the accurate prestressing of the preloading beam, yet the total stiffness is much inferior to that demanded by the frequency considerations.

The major problem with this design is the load distribution amongst the 8 wheels used on each axle. The distribution would have to rely on a certain amount of compliance on the rail mounting, and on a very good leveling of the rails themselves.

It should be realized that the above concept is based on an upper limit of performance of the wheels and any error in the load distributing schemes causes an overload.

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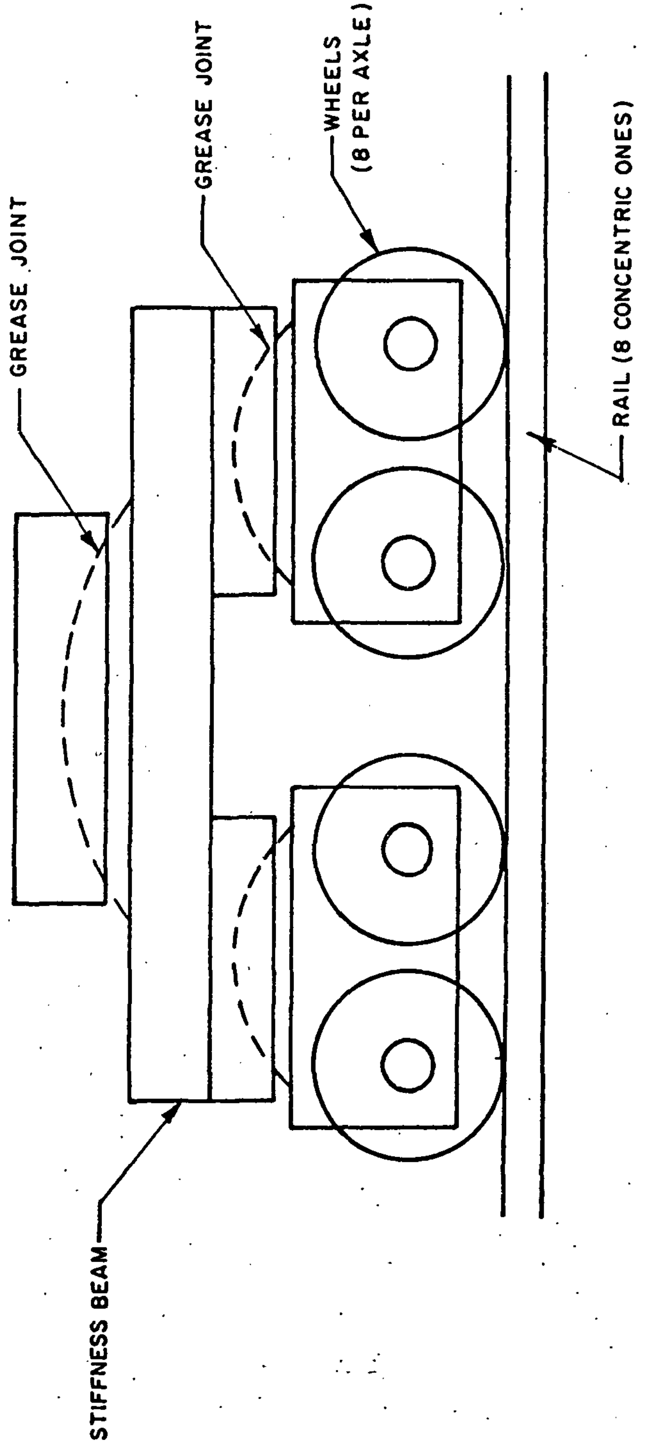

Figure 2. One of Three 32-Wheel Trucks at Each Alidade Foot

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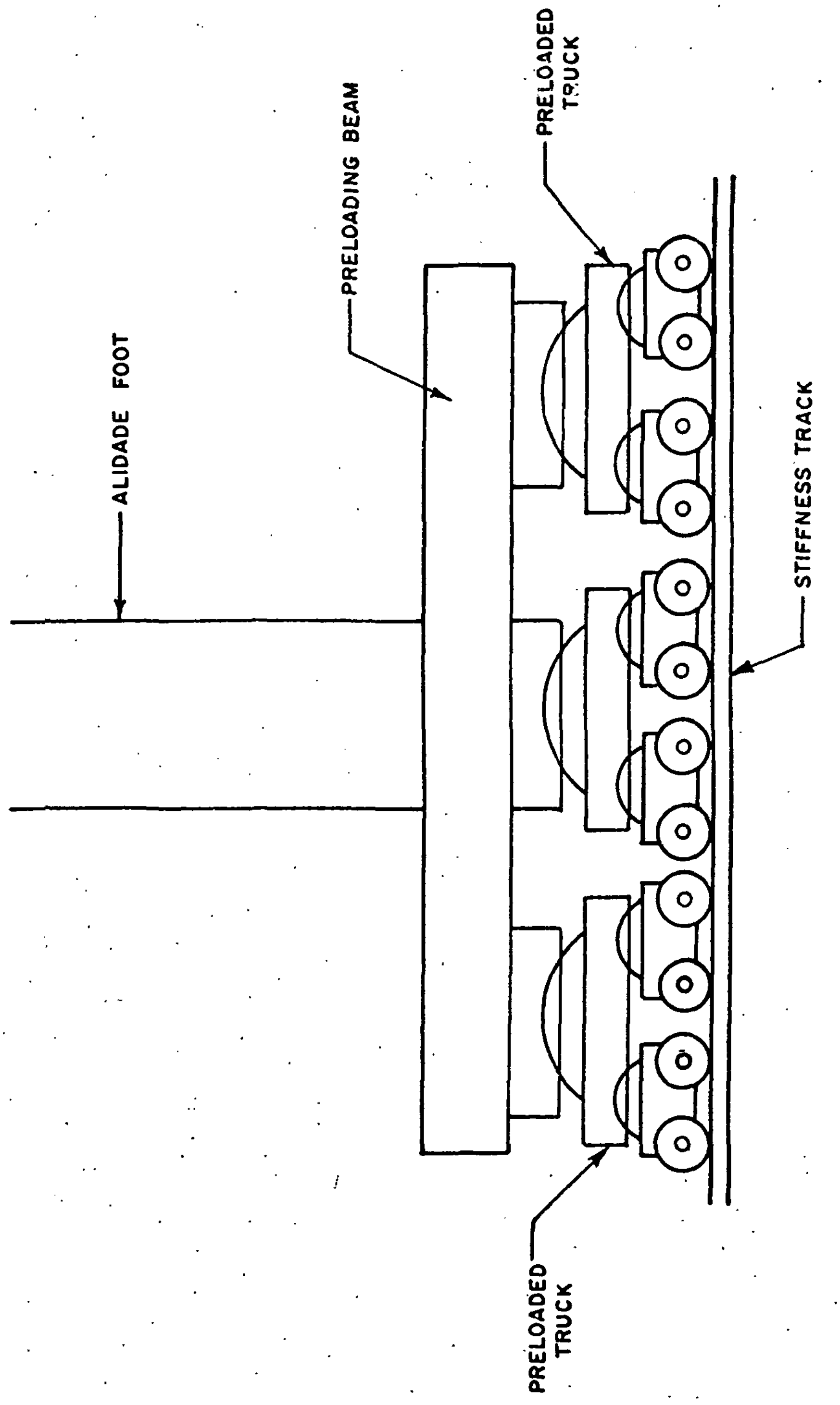

Figure 3. Total Assembly for the Foot of the Alidade with Two Preload Trucks and One Stiffness Truck

The design approach using wheels is therefore considered marginal and probably quite expensive to implement. Other features make pressurized oil bearings preferable to wheels, such as the smoothness with which they can move at extremely low speeds. Indeed it would be difficult to support an oil film in the wheel bearings when the speed of rotation is very, very small. Thus, high torque and possibly stick-slip conditions would be produced.

3.3 NUMBER OF PADS

The number of pads used is dependent on the working level of oil pressure selected for the system. An average projected pad pressure of 1000 psi results in approximately 1500 psi of recess pressure. This low pressure level allows overload due to malfunction of parts of the system, and the option to increase the value of working pressure up to 3 times without causing major damage to the system, while continuing operation.

Table 1 shows what a load of 16 million lbs. per leg, with 1000 psi average projected load requires in terms of pad sizing. Three pad configurations, namely, 1 x 1, 3 x 4 and 2 x 3 and either 1 pad per leg or 3 pads per leg are shown. The advantage of rectangular pads is a saving in runner width. At the same time, however, a great difference between width and length of the pad would cause problems both in finish machining and in the pad bending. The proportion of 2 x 3 could be considered a good compromise.

Table 1

PAD SIZE BASED ON 1000 PSI AVERAGE LOAD AND 16,000,000 LBS. PER LEG

Pad Proportions / Pads Per Leg	1 x 1	3 x 4	2 x 3
1	10.54' x 10.54'	9.12' x 12.17'	8.6' x 12.91'
3	6.09' x 6.09'	5.27' x 7.03'	4.97' x 7.45'

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From the last column of Table 1, observe that the minimum width of the runner for the single pad configuration is greater than 9 feet. In the 3-pad per leg configuration, the track width is approximately 5 1/2 feet. The 60 x 90 in. pad dictated by the 3-pad-per-leg configuration on a 2 x 3 aspect ratio is definitely feasible in terms of available machining equipment. The same can be said for the runner dimensions that it entails. Grave doubts have to be cast on the capability of manufacturing a single pad configuration of 102 in. by 155 in.

Splitting the bearing into 3 pads per leg introduces the added problem of distributing the load amongst the pads. It would be a very difficult task to accomplish this without sacrificing stiffness. It is, however, evident that the oil films have a tendency to exhibit more stiffness than that required of the bearing system. Even with a film thickness of 20 mils the complete maximum stiffness of the hydrostatic bearing system carrying 48 million lbs, would be 86.0 x 10^9 lbs. per foot.

It is possible to sacrifice part of this stiffness in order to accommodate the multipad configuration. This stiffness sacrifice consists of relegating the task of providing the suspension stiffness to a single pad, while letting the other pads sustain part of the load without providing any stiffness whatsoever resulting in 28.7 x 10^9 lbs./ft. overall stiffness. This concept is illustrated in Figures 4 and 5. In Figure 4 the main pad provides the stiffness while the two lateral pads are loaded by means of the elastic deflection of a preloading beam. This beam is made soft enough so that it does not contribute to or interfere with the stiffness of the central pad. The configuration is quite simple and would be rather easy to manufacture, but it has the major disadvantage of always maintaining a load on the two satellite pads. If any failure mode is encountered in these side pads, the antenna aximuth motion would have to be immediately arrested. The configuration of Figure 5, instead, is predicated on having the load imposed on the satellite pad by means of hydraulic jacks. The oil is provided to the jack cavities at high pressures and from there is passes through a restrictor to the recesses in

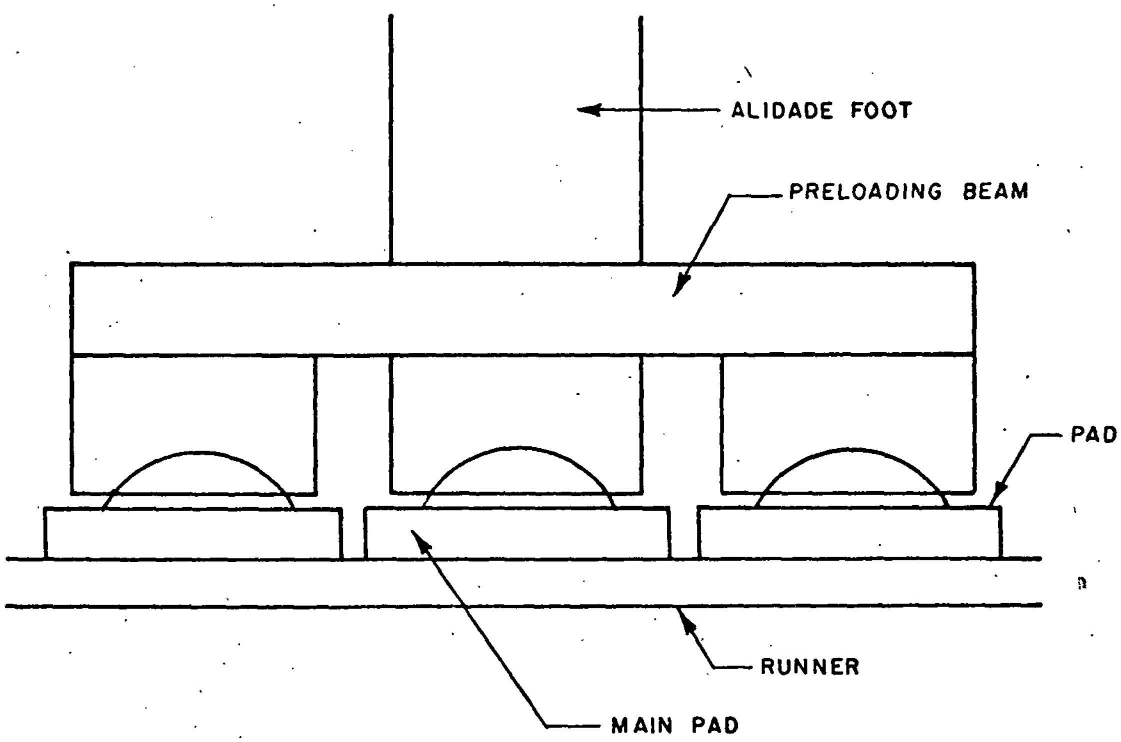

Figure 4. Three Pad Arrangement with One and Two Elastically Preloaded Pads

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the film of the pad itself. This configuration insures that if failure in the feeding system should occur in the satellite pad, both the film and the loading pressures would be lost simultaneously, thus not causing a direct pad failure. In the event of such a malfunction both satellite pads would have to be shut off at the same time in order to prevent serious damage to the structure, so the feed lines to the 2 satellite pads would have to be in common. The central pad then, is designed to accommodate a third of the load with its consequent stiffness, and in the case of malfunction of the satellite pads, this central pad would carry the entire load of that leg. Based on an even split of the load, each of these 3 pads would carry 5 1/3 million pounds. In case of malfunction of the satellite pads, the 16 million lbs. would all have to be carried by the central pad, raising the average projected pressure on it to 3000 psi, and reducing the oil film thickness to 0.014" which is still acceptable.

The sketch of the plan view of this three pad configuration on a single runner is shown in Figure 6. Note that the transverse loading beam does not have to be located exactly at the center of the main pad nor does it have to be exceedingly stiff since its deflection is compensated by the jacks.

Added benefits of the configuration of Figure 5 are the ease with which, by proper load control, the pads can be removed from the antenna foot. The removal of the central pad can, for example, be carried out by feeding higher than normal pressure to the hydraulic jacks while the removal of any one of the two side pads can be carried out by simply cutting the pressure from the hydraulic jacks thus loading the antenna completely on the main pad. The oil feed system in the secondary pad is schematically indicated by Figure 7.

Following is a simplified model of the oil pressure governing equations. The difference between the jack pressure and recess pressure is proportional to the flow rate q through the constant c

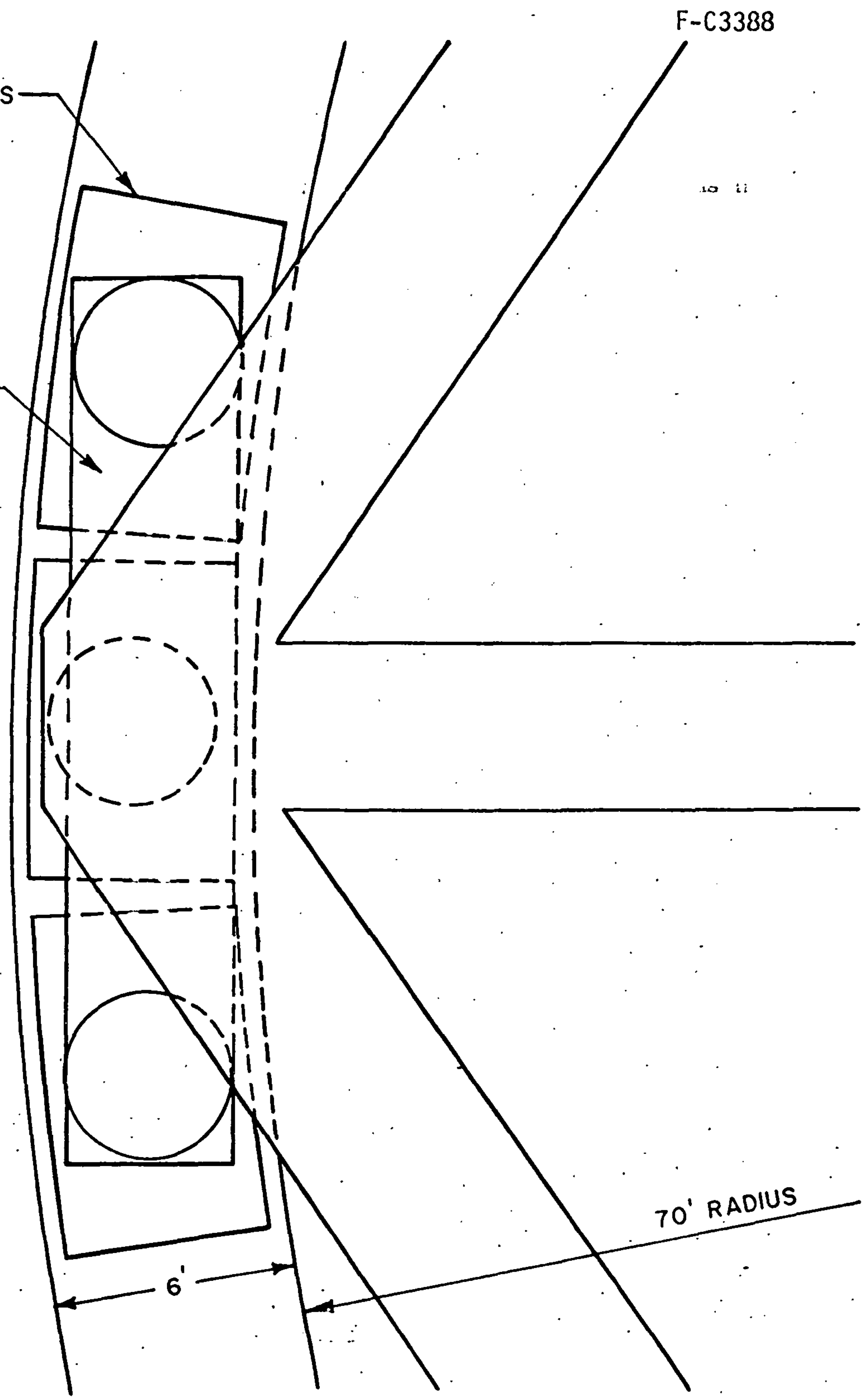

Figure 6. Pad View of Pad Arrangement on a Single Runner

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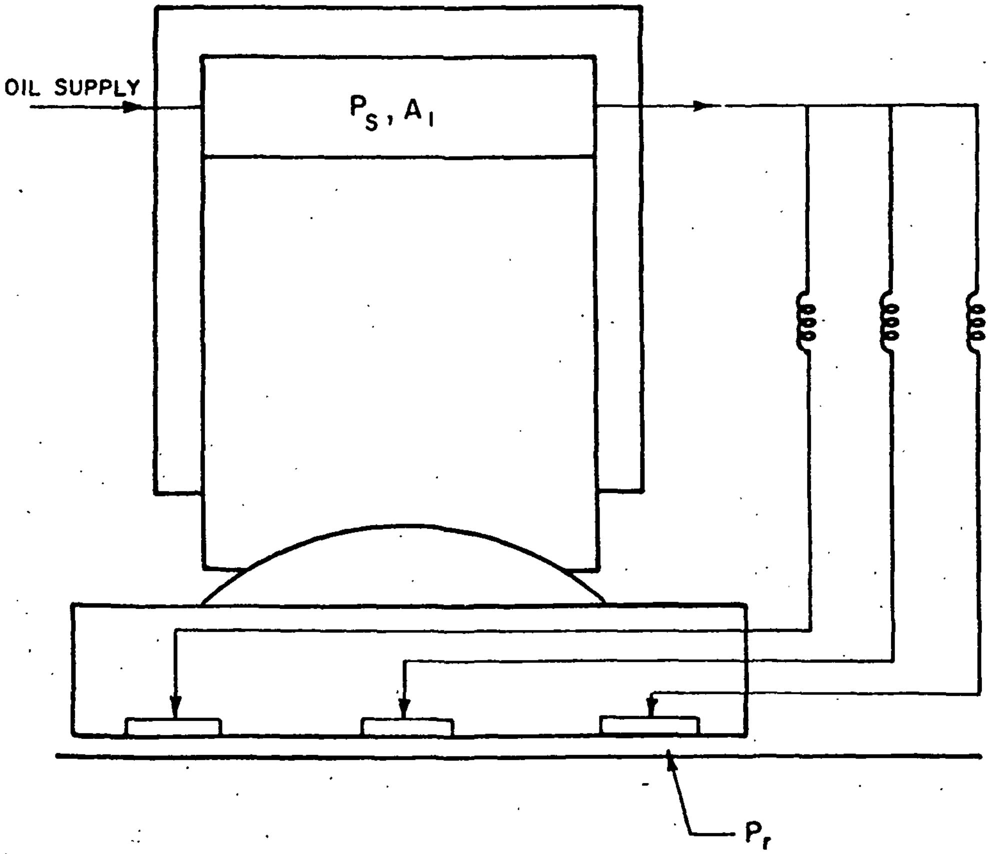

Figure 7. Secondary Pad Oil Circulation

$$P_s - P_r = cQ \tag{13}$$

Furthermore, the load imposed by the oil pressure in the jack is equal to the product of the pressure in the jack cavity and its cross-sectional area.

$$W_{jack} = P_s\, A_1 \tag{14}$$

As far as the bearing is concerned, the load that it will carry is proportional to the recess pressure p_r, its total area A and the area constant a_f, which depends on the distribution of recesses and clearances in the pad design.

$$W_{brg} = a_f\, A\, p_r \tag{15}$$

The relation between the flow and the load generated by the bearing film is given by

$$Q = q_f \frac{W_{brg}}{A} \frac{h^3}{\mu} \tag{16}$$

where

h = film thickness

μ = equals oil viscosity

q_f = flow coefficient depending on the bearing geometry

Combining these equations to obtain the value of the film thickness we obtain

$$h^3 = \frac{\mu}{q_f c}\left(\frac{A}{A_1} - \frac{1}{a_f}\right) = \frac{\mu}{q_f c a_f}\left(\frac{P_s}{P_r} - 1\right) \tag{17}$$

where we see that at equilibrium, that is when the load generated by the bearing is identical to the load generated by the jack, the clearance in the bearing is independent of the value of the flow passing through it and therefore of the load that the bearing carried. On the other hand, the load carried by such a unit is directly proportional to the flow rate q.

$$W = \frac{cQ}{\left(\frac{1}{A_1} - \frac{1}{a_f A}\right)} \tag{18}$$

It is then feasible to feed the jack cavity by means of either a controlled pressure or controlled flow rate and thus establish the amount of load carried by the secondary pad. As an example, a flow of 60 GPM of an oil with viscosity equal to 10^{-4} lbs/sec^2 would sustain a load of 5 million lbs. while maintaining a clearance of 20 mils, for bearing proportions which are quite feasible. These proportions are

$$\left.\begin{aligned} A/A_1 &= 2.4 \\ a_f &= 0.6 \\ q_f &= 2.7 \end{aligned}\right\} \tag{19}$$

In this case the value of the restrictor coefficient to be built into the capillaries is equal to C = 3.8 lb-sec/in.^{2}

From the above discussion, it is felt that splitting the load at each alidade leg into three equal portions each carried by a pad, results in physical pads sized at 60" x 90". The load distribution problem is solved by sacrificing 2/3 of the maximum achievable stiffness. This configuration is a good compromise between manageability of pad size, average pressure of the film, practical dimensions of the runner and offers reasonably reliable operation under mechanical or hydraulic type of malfunction.

3.4 MECHANICAL LOAD EQUALIZERS AND MULTIPLE RUNNERS

Three pads of each alidade leg are shown in the previous section to be practical. At this point questions arise whether or not it would be convenient to use mechanical leveling or load equalizing systems as shown in Figure 8 or even multiple parallel runners as shown in Figure 9 to distribute the loads.

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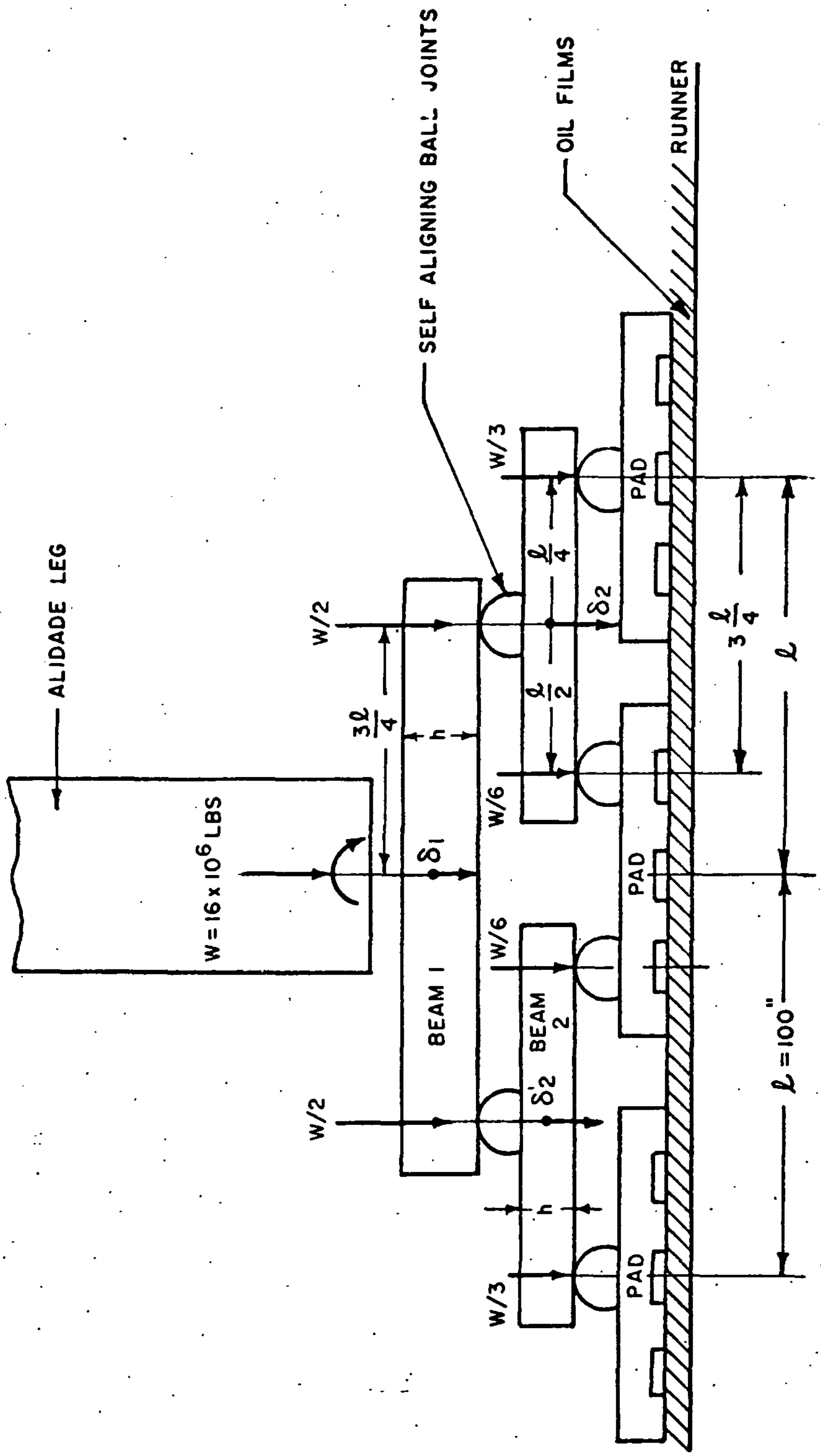

Figure 8. Mechanical Load Equalizers

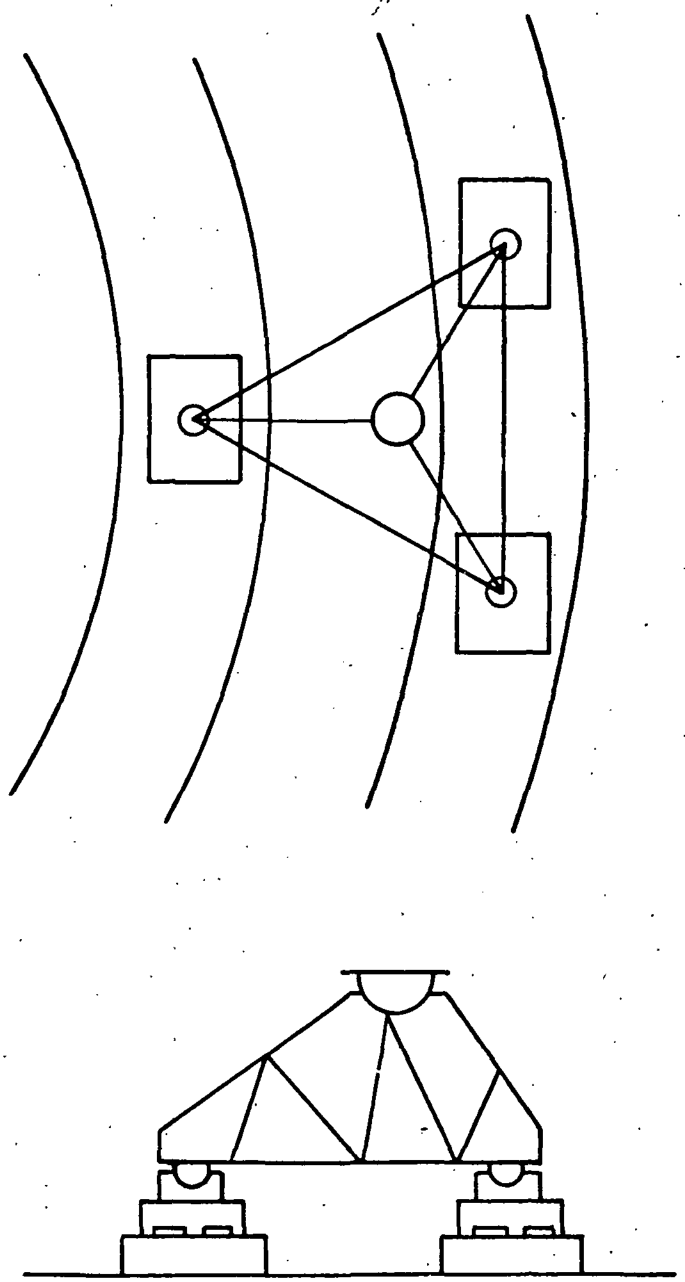

Figure 9. Multiple Parallel Runners

First consider the overall stiffness requirements that total deflection not exceed 0.3 in. for the load equalizing scheme of Figure 8. The distance between the three hydrostatic pads is denoted by ℓ, and the equalizer beam thicknesses by h. δ_1 and δ_2 are the deflections of equalizer beams 1 and 2 respectively. Assuming no deflection in various aligning ball joints stiffness requirements give:

$$\delta_1 + \delta_2 \leq 0.3'' \text{ for } W = 16 \times 10^6 \text{ lbs.} \tag{20}$$

From standard beam formulas (for bending only) the beam deflections are:

$$\begin{aligned} \delta_1 &= \frac{W(1.5\ell)^3}{48EI} = \frac{9W\ell^3}{128EI} \\ \delta_2 &= \frac{(\frac{W}{2})\ (\frac{\ell}{2})^2\ (\frac{\ell}{4})^2}{3EI(\frac{3}{4}\ell)} = \frac{W\ell^3}{288EI} \end{aligned} \tag{21}$$

Then from Equation (20):

$$0.3 \leqq \frac{W\ell^3}{EI}\left(\frac{9}{128} + \frac{1}{288}\right) \leqq \frac{0.074W\ell^3}{EI} \tag{22}$$

or:

$$I > \frac{0.246W^{\ 3}}{E} \tag{23}$$

Assuming:

$W = 16 \times 10^6$ lbs
$\ell = 100$ in.
$E = 30 \times 10^6$ psi

$I > 131{,}000$ in.^{4} based upon deflection

For a solid rectangular beam

$$I = \frac{bh^3}{12}$$

Then for b = 30"

$$h = \left[\frac{(12)(131{,}000)}{30}\right]^{1/3} = 37.4\text{" depth} \qquad (24)$$

The bending stress in the upper solid rectangular beam 30" x 37.4" deep (1.9 tons/ft) would be 85,650 psi which is far too high for A-36 or similar structural steel. Shear deflections would also need to be considered and an even thicker beam required to limit overall deflection to 0.3". Assuming a safe stress of 20,000 psi would result in a beam depth of 77.5" (3.95 tons/ft or 49.4 tons for the top beam) under 16×10^6 lbs. applied load. Total beam weights for the three equalizer beams of Figure 8 would be approximately 100 tons (not including the self-aligning ball joints). Total height would exceed 20 ft. from runner to alidade leg. An arrangement of equalizer beams as shown in Figure 8 is unwieldy and impractical when compared to the three-pad piston concept of Figure 5. The arrangement of Figure 8 also has problems of load centering in the direction radial to the runner. Maintenance would require complete removal of the entire beam arrangement. The alidade leg would need an external means of temporary support to enable such removal and maintenance. Hydraulic failure of any one of the three hydrostatic pads would have to be immediately arrested (similar to Figure 4). The mechanical leveling or load equalizing scheme of Figure 8 is considered impractical.

The use of more than one runner as illustrated in Figure 9 offers some advantages which are outweighed by obvious disadvantages. Advantages include automatic load spreading (three points determine a plane) on the foundation in the radial direction. As will be seen later, this is quite desirable. The list of disadvantages is much longer. Multiple runners are of course much more expensive than a single one. Moreover, oil distribution schemes would have to be implemented so as to avoid overflow of one bearing system. The geometrical arrangements of the pad would make it cumbersome to locate the azimuth drive gear in a convenient place and might also crowd the radial bearing to some extent. The foundation due to its much increased width would become much bulkier and more cumbersome. Failure of one hydrostatic pad would require immediate stoppage of the antenna pad replacement and maintenance would be very difficult. The same arguments advanced for the unwieldy and bulky load equalizer beams would also apply to the multi-runner concept of Figure 9. The trusswork connecting the pads would need to be designed to safe stress limits and would be impractical when compared to the three pad hydraulic jack configuration of Figure 5.

Due to all of its disadvantages, neither a triple runner or a double runner system will be pursued any further.

4. OIL BEARING DESIGNS

This discussion assumes the following: The 3-pad configuration has been selected and the pad dimensions are 60 in. by 90 in. A single runner accommodating all pads is being used. There is no restriction in locating the pads very close to one another in the circumferential direction. The loading beam splits the load at each leg into three equal portions. A sufficient amount of stiffness can be provided by the center pad alone at essentially any practical film thickness.

4.1 COMPLIANT SURFACE VS. RIGID SURFACE

In an application such as this, where the flatness of the runner and pad surface have to be so high, with respect to their other dimensions, it is always attractive to think of a compliant surface geometry in order to relieve some of the requirements. Thus, coating the surface of the pad with a layer of elastomer would definitely be quite forgiving in terms of asperities, lack of flatness, and grooving of the runner surface while at the same time saving a great deal of oil flow and providing the required stiffness. It is however, unfortunate that no well-tested elastomer is known that could be put on the pad surface and exhibit sufficient shearing and ripping strength in order to offer a foolproof application. However, this type of solution should be kept in mind in case the geometrical dimensions of the antenna remain the same while for some reason the load decreases by a factor of 2 or more. It has been well established that compliant surface bearings having rubber as the elastomer can be used very reliably for pressures on the order of 500 psi average. Not much experience is available at higher pressures but it is well known that the ripping strength of most elastomers is in the order of 1000 psi and that at these load levels failure could easily occur.

4.2 OIL VISCOSITY

For the purpose of preliminary discussion, we can assume that the oil film under the pads will be of uniform thickness. It is then easy to show that the load-carrying capacity is given by

$$W = a_f\, A_p\, p_r \quad \text{(lbs.)} \tag{25}$$

where

p_r = recess pressure

A_p = pad area

a_f = load coefficient (dependent upon the pad and recess geometry)

The flow rate needed to feed such a bearing is given by

$$Q = q_f \frac{W}{A_p} \frac{h^3}{\mu} \quad (\text{in.}^3/\text{sec}) \tag{26}$$

where

h = film thickness

μ = oil viscosity

q_f = flow coefficient (dependent upon the pad and recess geometry)

The power dissipated by such a bearing from the recess to the ambient is given by

$$H_B = \frac{H_f}{q_f} \frac{W}{A_p} Q \quad \text{(in.-lb/sec)} \tag{27}$$

where

H_f = flow coefficient (dependent on geometry only)

The behavior of hydrostatic oil pads of many common configurations has been described by Rippel in the Cast Bronze Hydrostatic Bearing Design Manual. Figure 10 presents data on the variation of pad coefficients for flat, rectangular, 4-recess pads.

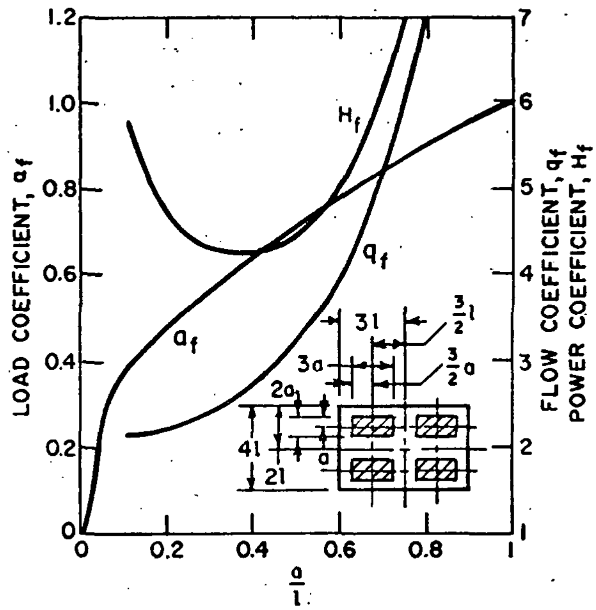

Figure 10. Pad Coefficients for a Rectangular Bearing with Four Rectangular Recesses. $A_p = 24\ell^2$

In this figure, we may select for reference purposes a value of $\frac{a}{\ell}$ of 0.4. Note that the configuration treated here is a 4 recess pad whereas the actual one most probably would be either a 6 or an 8 recess configuration. However, this is not going to make too much difference in the case of uniform film thickness. The reason for the multi-recess configuration is simply to give local stiffness to all parts of the pad.

For the selected value of $\frac{a}{\ell}$ of 0.4 we can read that $a_f = 0.64$, $q_f = 2.75$ and $H_f = 4.3$.

The selected unit pressure of 1000 psi results in a recess pressure of 1560 psi. This is deemed to be quite a reasonable pressure level and that it is well within a factor of 3 of the capability of most high pressure pumps. The various tradeoffs which are due to a selection of oil viscosity and the operating film clearance are indicated in Table 2.

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Table 2

FLOW RATE, PUMP POWER, AND DISSIPATION FOR VARIOUS VISCOSITIES AND FILM THICKNESSES

Oil Viscosity lb. sec/in.^{2} (reyns)	100×10^{-6}	20×10^{-6}	10×10^{-6}	5×10^{-6}
Oil Film Thickness h = 0.010"	Q = 7.01 GPM/pad	35.06	70.12	140.25
	H = 6.38 HP/pad	31.90	63.81	127.6
	Heat = 16.23 $\frac{\text{BTU}}{\text{HR-Pad}}$	81.19	162.4	324.7
h = 0.015"	Q = 23.67	118.33	236.65	473.31
	H = 21.54	107.68	215.4	430.7
	Heat = 54.82	274.0	548.2	1096.1
h = 0.020"	Q = 56.10	280.48	560.96	1121.92
	H = 51.05	255.24	510.5	1020.9
	Heat = 130.0	649.6	1299.2	2598.2

For the purpose of visualizing what type of oil the various viscosities would represent we would say that 100×10^{-6} lb-sec/in.^{2} represents a heavy gear oil at 100°F, 20×10^{-6} is the viscosity of SAE 50 at 120°F and the viscosity of 5×10^{-6} is the viscosity of SAE 20 at 100°F. As it is well known the viscosity of oils varies greatly with temperature. For reference purposes one should be reminded that typical oils change their viscosities by a factor of 10 every 100°F. Figure 11 gives some data on the viscosity of common motor oils and an available heavy viscosity gear oil. Figure 12 shows the same oils in SSU units. In reading Table 2 for the heating effects in passing the oil from the recesses to the ambient it should be noticed that the oil temperature rise is always the same and given by

$$\text{Oil Temp. Rise} = \frac{H}{\rho Q c_p} = \frac{1}{\rho c_p a_f} \frac{W}{A_p} \qquad (28)$$

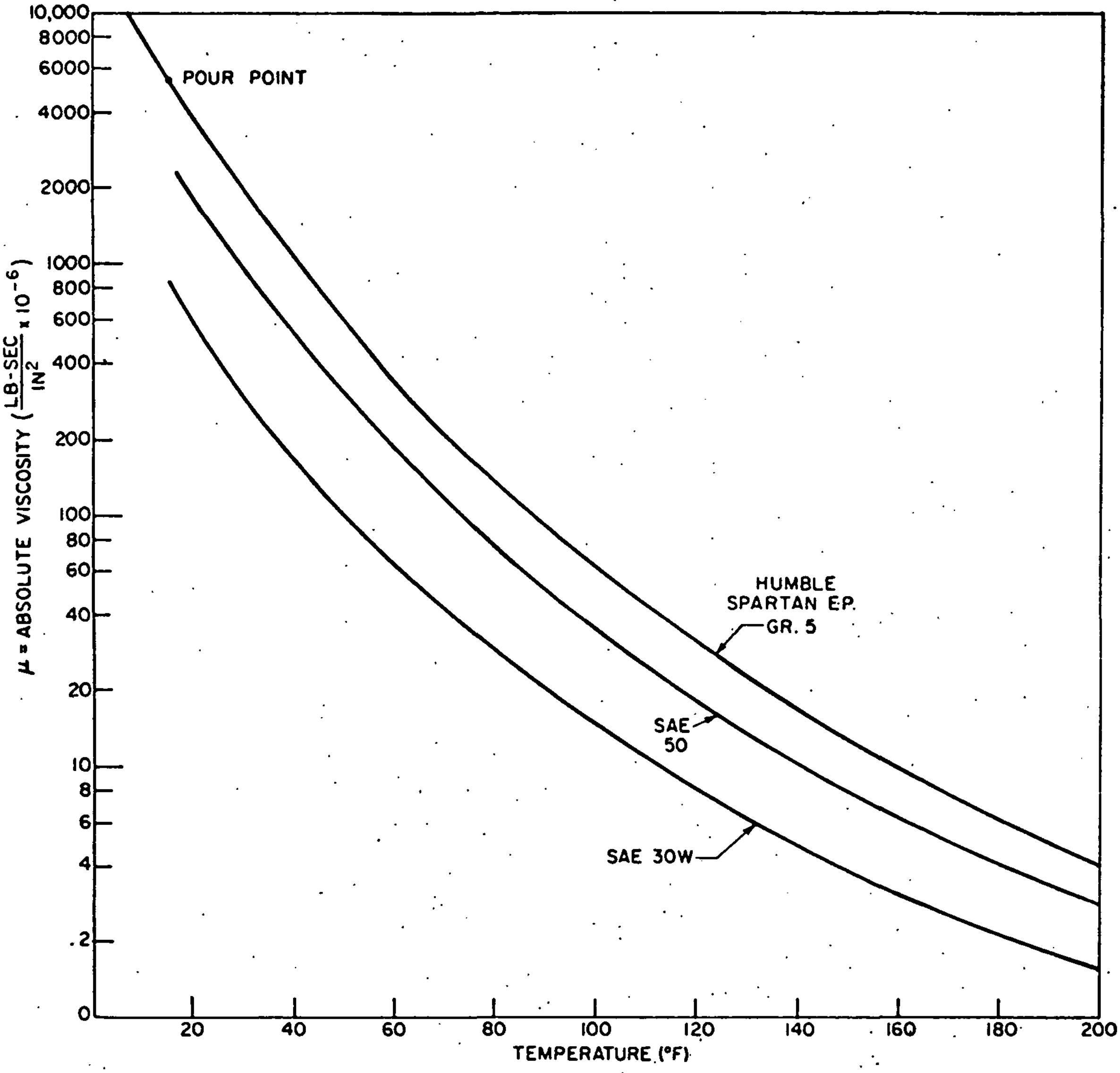

Figure 11. Absolute Viscosity of Various Oils vs. Temperature

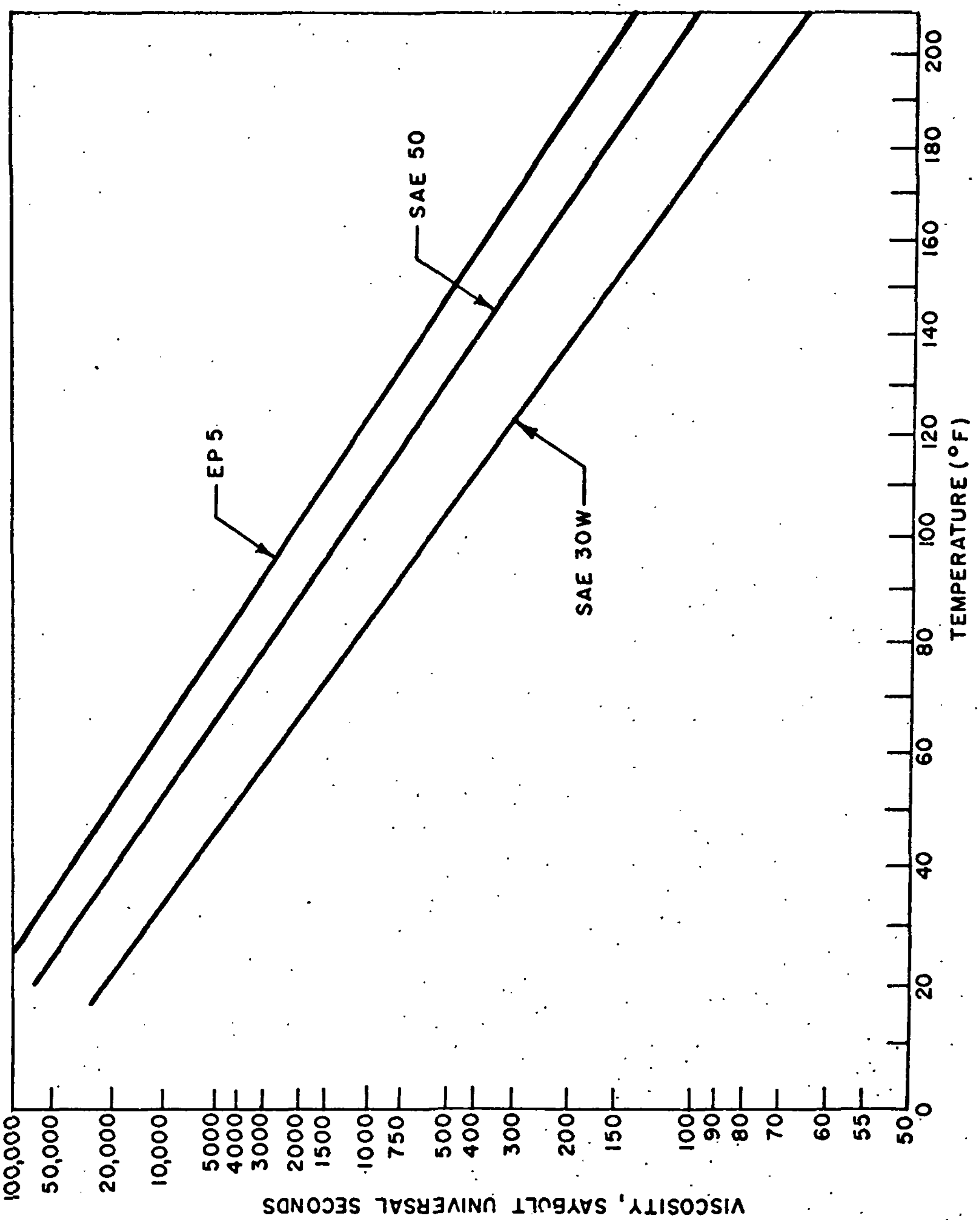

Figure 12. SSU Viscosity of Various Oils vs. Temperature

where

ρ = oil density

c_p = oil specific heat

It is interesting to note that the oil temperature rise is only dependent upon the unit load on the bearing. For our case under discussion this temperature rise is 12.3°F.

The heat values of Table 2 indicate that the total amount of heat dumped in the oil reservoir is indeed dependent on flow rate although the temperature rise in the oil through the pads is independent of it.

Due to the large size of the bearing pad and the experience with the 210 foot Goldstone Antenna it is deemed desirable to aim for a sufficiently large film thickness so as not to have to worry about contact occurring as a result of minor malfunctions or defects in machining. This is the reason why the row containing data for a clearance 20 mils has been introduced into Table 2.

If a clearance of 20 mils is selected, it becomes clear from Table 2 that enormous values of flows, pump horsepower and oil heating per pad are required unless a very high oil viscosity is used. Consulting Equation 25, 26 and 27, it is clear that higher oil viscosity is desirable from the point of view of reducing the flow and thus reducing the required horsepower once the unit load and the clearance have been selected. The column of Table 2 relating to the heavy gear oil, indicates that very reasonable flow rates and power requirements are needed even for a clearance of 20 mils. The big question with this high oil viscosity is the operation of the pumps themselves and of all ancillary equipment such as filters, heaters, coolers, pressure controls, pressure relief valves, flow controls and sump pumps. The other drawback with high viscosity oils is that the larger the viscosity the higher the temperature associated with the so-called pour point. An oil exhibiting a viscosity of 100 microreyns at 90 to 100°F would probably exhibit a pour point of 15°F (see Appendix A). This poses some problems in cases where the antenna is inactive on particularly cold winter days when the temperature goes below 15°F.

The problem of passing the oil through filters, control valves, and high pressure pumps can be alleviated by heating the oil as soon as it exits the main reservoir. From Figures 11 and 12 for a heavy gear oil it is evident that heating the oil 30°F from 80°F to 110°F would reduce the viscosity to one-third of its original value. After the oil exits the high pressure pumps it can be sent through a cooling system just preceeding and perhaps including the pads so that the low temperature reestablishes a high oil viscosity. This heating before the hydraulic equipment and cooling just before the oil entrance into the recesses and film could be accomplished by a heat pump arrangement. One pad using the heavy gear oil would require 11,719 BTU/hr (4.6Hp) for each °F for the 56.1 GPM flow which would result in a total 351,575 BTU/hr (138 Hp) for a 30°F change. The entire antenna system of nine pads would require a total of 3.16×10^6 BTU/hr (1242 Hp) for either heating or cooling the entire system 30°F. This power requirement is 2.7 times the total pump horsepower required to maintain flow at 100×10^{-6} reyns (per Table 2) and does not seem practical. The alternative of using the heavy gear oil at low temperature (85°F) and high viscosity (100×10^{-6} reyns) does not involve any major problems since equipment is presently available to filter and pump it.

In this area some further studies are necessary in order to select the best solution. However, the approach of providing the system with 1500 psi normal pressure supply condition and 4500 psi emergency pressures supply condition can be considered to be within the realm of possibility.

4.3 POWER-FLOW-PRESSURE DISCUSSION

When the two side pads fail, the pressure in the center pad increases by a factor of 3, to 4500 psi. Assuming that the pumps feeding it would maintain their flow rate, equation 26 and Table 2 indicate that the clearance in the central pad would decrease by simply a factor of $3^{1/3}$ or 1.44. Thus, for a 20 mil initial clearance, the decrease would be to approximately 14 mils which is sufficiently safe. This lower clearance would provide the design criterion for the purpose of specifying the machining accuracy of the pad and runner surfaces. As far as the other parts are concerned, we see that the oil flow rate depends only on the unit load, the clearance, and the viscosity. Therefore, the selection of a rather low unit load is favorable from the point of view that it gives a relatively low flow rate. Likewise the selection of a high viscosity also indicates a low flow rate. Similarly, from equation 27 and Table 2, one can see that the low unit pressure and high viscosity contribute also to a low power dissipation.

From an examination of Table 2 we can also conclude that the two lighter oil viscosities combined with the 15 mil clearance are unfeasible, and the three lighter oil viscosities combined with the 20 mil clearance are also unfeasible. This is particularly clear when one considers that the horsepower requirements indicated in Table 2 are oil flow horsepower at the recess and thus do not include any line loss, pump loss or motor losses. Therefore, the horsepower figures in Table 2 should really be multiplied by a factor which nears 3 in order to visualize the actual horsepower per pad required at the motor.

4.4 COMPUTER PROGRAM

The analysis of behavior of the fluid film of the actual design will have to be carried out by means of an adequate computer program due to the fact the film thickness in such cases will be variable. However, the programs which had been developed for the design of the azimuth bearing of the 210-foot diameter AAS Goldstone Antenna are quite suitable for the handling of the present job. The only modification that might be necessary is that of the introduction of shear dependent viscosity in the case that the heavy oils may exhibit non-Newtonian behavior. This modification is not a major one and would not require any great expenditure of money.

Naturally, the use of this computer program will have to be coupled with programs which predict the deflection patterns of both pad and the runner as these actually determine what the clearance is. Those programs will be discussed later in this report.

5. ALIGNMENT AND LOAD SHARING MECHANISMS

In this section the mechanism which distributes the load among the pads and allow the alignment of each one of the pads with the runner will be discussed. This consists of two pistons and three lubricated ball joints.

5.1 PISTONS

From the description which was given in Section 3.4 of the piston support for satellite pads, it would seem that for proper operation, the ratio of pad area to piston area should be greater than the reciprocal of the factor a_f (which in the chosen example is equal to 1.6), and it is obvious that for proper compensation a substantial drop should exist between the piston cavity and the recess. For reference, an area ratio of 2.4 is selected here. This gives an area ratio larger than required and a piston pressure of 2350 psi for a diameter of 54 in. The proportioning and detail design of pistons would have to take advantage of the hydraulic jack technology. It can be seen that perhaps at the expense of a little flow reasonable clearance can be made to exist between the male and the female even if seals could not be designed for this purpose. It is clear that the leakage situation is helped by the high velocity in the oil and by the fact that in general very little motion is envisioned between the two parts.

Some further studies are necessary in this area from an optimization point of view; materials should be selected, a proper diameter to length ratio should be determined, seals should be investigated, and the possibility of transforming these pistons into bellows should also be looked into.

For reference purposes, if the cylinders were made with 50 in. of overlap and 40 mils of diametral clearance, the heavier of the selected oils (viscosity of 100 microreyns) would leak at a rate of 12 gallons per minute from the piston itself which is not excessive. If further studies indicate that contact wear occurs between the two moving parts of these pistons then some hydrostatic oil bearings could be interposed between the two parts thus eliminating all chances of contact.

5.2 GREASE BALL JOINTS

In order to compensate for the deflections of both the runner and the loading beam it is necessary to make the pads self-aligning so that the film may remain as uniform in thickness as possible. The ball and socket joints will then be retained. Since the balls can be scaled essentially the same way as the pads, the unit pressures will be the same order of magnitude as those that were encountered for the case of 210-foot diameter Goldstone Antenna. The operation of the Goldstone Antenna has been completely satisfactory from a point of view of the alignment mechanism. This suggests use of the same type of configuration of ball and socket with grease lubricant periodically resupplied. The only suggested change with respect to the Goldstone situation is that of locating the center of the ball and socket surfaces at approximately the level of the oil film. In this manner, an obstacle encountered by the pad in motion will cause the smallest possible moment tending to drive the pad further into the runner. In accordance with this, the anti-rotation mechanism, which must prevent the pad from rotating around the vertical axis in the grease ball and socket, should restrain the pad as close as possible to the level of the oil film. Figure 13 is a schematic of the ball and socket arrangement and antirotation mechanism. In the plan view of Figure 13, the example is shown of the case when the pad traveling from left to right, hits an obstacle on the runner and thus is confronted with the force labeled A against the pad itself. Assuming that there is no friction in the ball and socket, the only forces that can be generated by the alignment mechanism have to pass through the center of the ball and socket. The force that directly opposes A is labeled B. However, these two forces now create a couple about the vertical axis z, and would make the pad rotate in a counterclockwise direction if no restraint were imposed on it. The anti-rotation mechanism imposes the force labeled C. To balance this force, the ball and socket originate the force D through the center. All forces on the plane parallel to the runner have thus been eliminated and all moments along an axis perpendicular to the runner have also been eliminated. It is clear from the figure that if the center of rotation of the ball is above the level of the runner, that is, if force B is at a higher level than force A, the two will generate a moment about the minus R axis. This will tend to drive the front edge of the pad into the runner, thus increasing the contact force. Moreover, if the center of rotation of the ball is higher than the point of application of the anti-rotation force, i.e., if the level of force B is higher than the level of force C, they will generate a moment about the negative θ axis which will cause the top edge of the pad to be forced against the runner, further increasing the contact force.

The above discussion indicates that the center of rotation of the ball should be somewhat below the level of the runner surface. Furthermore, with this arrangement the location where the antirotation forces are transmitted should not be made unnecessarily high because in that case, the couple formed by the forces C and D would have a tendency to drive corner E into the runner as a result of impact at corner F.

The only area that might require further study in the alignment mechanism is that which concerns the automatic periodical application of grease to the ball recesses, thus eliminating the possibility of error in the reliance on a manual operation.

5.3 PAD DESIGN

As in the case of Goldstone Antenna, the main concern with the pad design is that of distributing the recess geometries and the pad

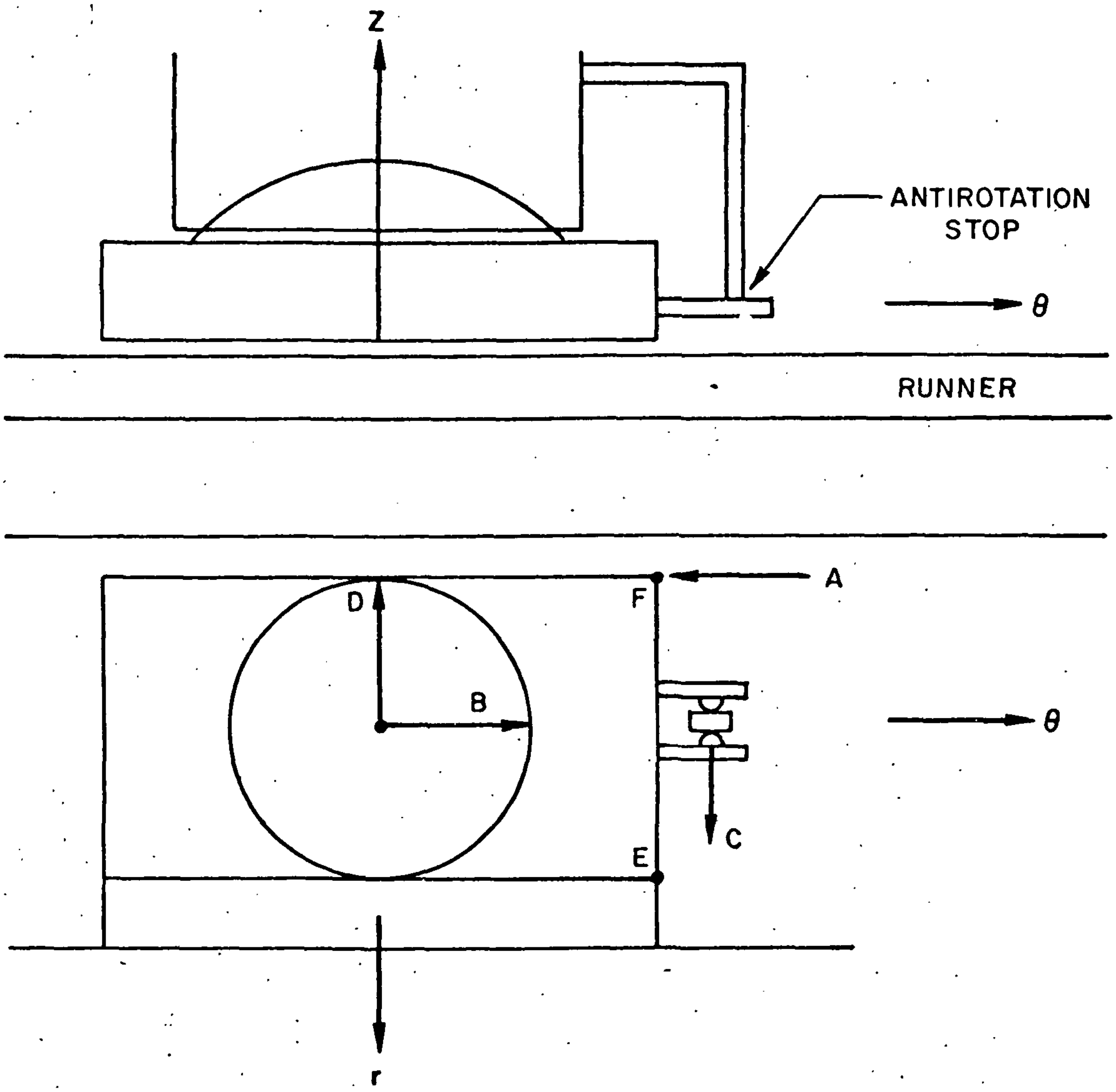

Figure 13. Grease Ball and Socket Orientation
Force Analysis

thickness so that the pad deflection distribution will match the deflection distribution at the other end of the foundation. Then the film thickness can remain at the design value at all points and avoid contact.

The first task consists of evaluating the forces generated by the hydrostatic oil film, and of using an iterative routine to match these forces with those necessary to generate the deflection distribution of the pad and runner foundation. This technique has already been well established from the study on the Goldstone Antenna.

The major difficulty to be encountered is in the area of the elasticity programs, which will have to be utilized in evaluating the deflection of the runner. The study of the pad deflections was made accurately enough for the 210-foot diameter case. In the three pad arrangement which is being envisioned for the present application, the major difficulty in deflection matching is going to occur with the satellite pad, due to the fact the central pad will essentially travel on a "flattened" runway. The two satellite pads will have the task of "steamrolling" the runner into its deflected shape. In order to provide a sufficient amount of local stiffness over the large area of these pads several recess configurations should be studied in the actual design calculation. It is quite possible that an 8 recess configuration could be more desirable than the 6 recess configuration used in the 210-foot case. Not much advantage is obtainable from circular pads since these do not minimize the dimensions of the runner.

Due to the particular task discharged by the satellite pads it might be quite convenient to consider elaborate schemes such as the tapering of their thickness at their outermost edge in the circumferential direction and possibly locate the ball and socket in an asymmetric position in order to facilitate the climb onto the undeflected part of the runner. Some of these solutions might entail the expenditure of large amounts of flow for some recesses but at lower pressure. Such a configuration is depicted in Figure 14. This variation of thickness and asymmetric ball position invalidates the type of analyses which were made

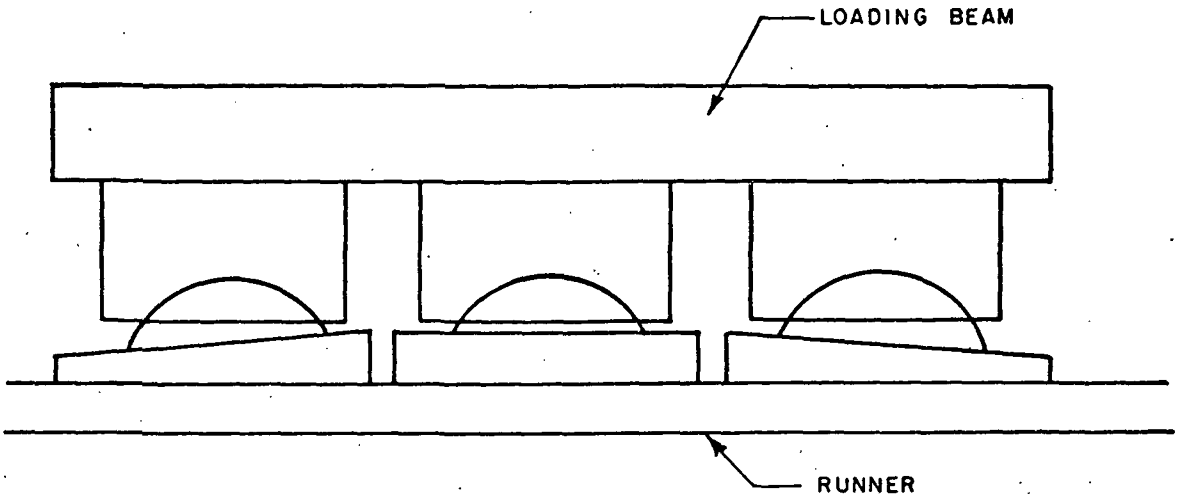

Figure 14. Asymmetric Satellite Pads

for the pad deflection of the 210-foot antenna. A different approach and program would be needed to evaluate the deflections in the present case. However, contrary to the situation which existed during the design stage to the 210-foot antenna, new finite element methods are presently available which can accurately predict the deflection of such an asymmetric pad.

As in the case of 210-foot antenna this area is not considered to be an unsurmountable problem. It just needs a lot of accurate and competent work. The deflections have to be evaluated in a painstaking manner and matched suitably well with those of the runner. A lot of this work is concerned with distributing the deflections in such a manner as to maintain a film in the situations which involve failure of the supply system to some of the recesses but which are desired to be "go" conditions for the antenna.

6. LOAD SUPPORTING STRUCTURE

The load supporting structure is perhaps the most troublesome area, and one that will require the most changes from the 210-foot design. Some of these changes are due to the increased load, and the increased values of mean pressure, and part of these changes will be necessary because of unsatisfactory experience with the 210-foot antenna. The major design areas in which attention should be focused are: (1) leveling mecnanism of the runner, both for the first installation and then afterwards, if anything causes it to lose level, (2) distribution of load at the metal-concrete interface in such a manner as to cause the load to fluctuate as little as possible and to exhibit a value of the maximum achieved stress which is as small as possible, and (3) design concepts should, if possible, provide a metal-concrete interface such that repairs may be accomplished in a simple manner without great expenditure of either down time or money.

As a reference dimension it will be hereafter assumed that the runner is 6 feet or less in width. (This will amply accommodate the 60 x 90 pad.) Other questions concerning the runner such as the size of the segment, the design of the joint sealing, and the rotation stops will be treated in Section 7.

6.1 LEVELING

Experience with the 210-foot antenna pointed out that easy access to the runner, in the sense of leveling ability, is quite desirable. When any malfunction of the under-structure which supports the load causes the runner to go out of level, it is quite important to be able to repair it with minimal expenditures of money and skilled manpower.

It was also learned that grout in close proximity to the runner and, therefore, to both high loads and to the presence of oil, will hardly survive a long time. It is therefore desirable to locate the steel-concrete interface, that is the grout, in a location which is as far as possible from the oil environment of the runner, and where the maximum load levels may be considerably lower than those experienced in the immediate vicinity of the runner itself. Figure 15 illustrates two concepts which can be utilized to accomplish both needs. The runner is supported by two or three vertical beams, the section of which is sufficient for supporting in compression the load on the structure. The top of the beams have rather narrow flanges. These flanges are separated from the runner by leveling shims. To limit the shim thickness to 1/4 in. requires a leveling arrangement of the vertical beams which brings their surfaces within perhaps 1/4 inch of an absolutely horizontal plane. Direct communication to the two external beams (and through holes in them to the internal beams in the case of the 3 beam geometry), is essential in order to easily introduce the shims. Due to the rather relaxed requirement on the flatness of the tracks (σ of 20 to 30 mils) it is quite probable that machining of the flanges on site is not going to be required.

If further study reveals that the on site machining of the flanges of foundation beams is necessary, this can be accomplished by means of a machine which uses very sensitive level as a reference and rides on temporary tracks. Such a machine, although expensive, is of rather straightforward conception and construction. It could be used for machining not only the surfaces of the supporting beam but also the runner surface itself.

As shown in Figure 15 the three vertical beams are spaced in such as to produce the smallest possible deflection of the runner. In the two beam case the deflection of the runner with the 1000 lb/per/square inch load could be limited to something in the order of 20 mils. In the three beam case the deflection could be cut down to something below 10 mils which is satisfactory. (All the above applies to a runner thickness in the order of 7 or 8 inches.)

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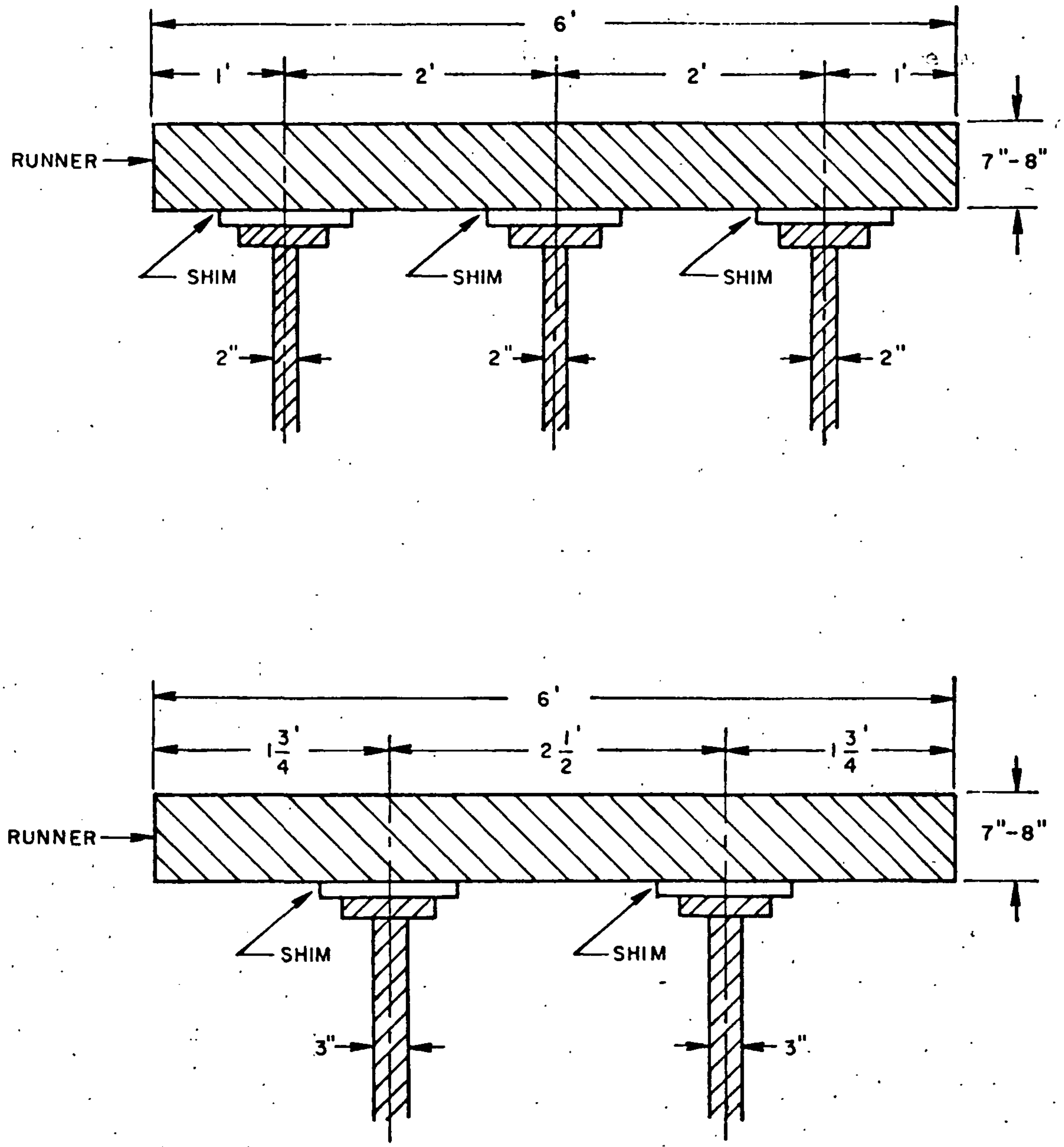

Figure 15. Runner Support and Levelling

Runner leveling by means of the shimming is to be performed using a sensitive mercury level as a reference. The consequences of aligning the runner by means of a level were studied:

a) the runner would not be flat but it would be higher by .53 x 10^{-3} inches at its east and west extremes

b) if the center of mass is 200 ft above the runner its load vector would not be normal to the runner by 4 x 10^{-8} rad.

Both of these are negligible.

It is also conceivable that after the initial leveling some rather inexpensive bubble capacitance levels could be installed at several places on the runner so that an automatic readout showing tilt from the horizontal could be had at all times. These levels could be easily attached to the underside of the runner. It is conceivable that a simple data acquisition system together with a computer program could be utilized to determine what adjustments would be necessary in the shimming in order to correct any lack of flatness of the runner. North American Rockwell presently manufactures levels which are approximately 1 inch in diameter and have a sensitivity of better than 1/10 of an arc second. Such a level could then be mounted one unit per 10 feet or less of runner. While the above scheme represents one means of leveling the runner, it is quite conceivable that other systems may be developed which are a lot simpler to implement and perhaps more trustworthy. Further attention should be given to this in the design stage.

6.2 LOAD SPREADING

To avoid having an alternating load of 1000 psi acting on the grout which forms the interface between the runner and the concrete base, it is desirable to spread the load in such a manner as to half the maximum stress. The first approach might be to have the load supported by beams (Figure 15) the height of which is sufficient to spread the load in the circumferential direction on the grout concrete interface. The spreading of the load is given by two factors: (1) the compression of the beams themselves, (2) the softness of the foundation which supports the beam. The deflection due to the softness of the beam in the longitudinal direction can be directly obtained from Filon's solution of the response to a point load, and an integration technique for spreading this solution. Figure 16 shows a plot and data for Filon's solution which spreads the load due to the beam thickness C when the load is concentrated. This solution can be utilized to obtain the spreading of the stress due to distributed load of intensity σ through a beam. In the present case the value of sigma is equal to 1000 psi multiplied by the width of the loaded region itself which is 60 inches.

Figures 17 and 18 depict typical load spreading situations implemented by means of load-spreading beams in the circumferential direction. In both cases the loaded region is 24 feet long and it is spread by longitudinal beams which are assumed to rest on a foundation the top of which is of the same width as the runner. Two beam heights are considered: one of 15 feet and the other of 30 feet. The first one causes a good spread of the loaded region over the foundation, but it can also be seen that the maximum stress at the center of the loaded region is still approximately 94% of the stress on the runner itself. This can be explained by examining the figure. The load spreading action occurs over the portion of the load which is at the limit of loaded region itself and not over the portions of the load which are in the central region. This is dramatized by Figure 18, in which even a 30-foot beam only cuts the maximum stress to 2/3 of the value on the runner itself. Since the objective here is to cut the maximum stress on the steel foundation interface to a value below 500 psi, we see that a beam of approximately 50 or 60 feet in height would be necessary in order to achieve this goal using this type of solution. As a consequence alternate approaches have to be examined.

Figure 19 illustrates how spreading the load in the radial direction is much more effective. In this direction, we take advantage of the fact that the distance over which the load acts is much shorter than it is in

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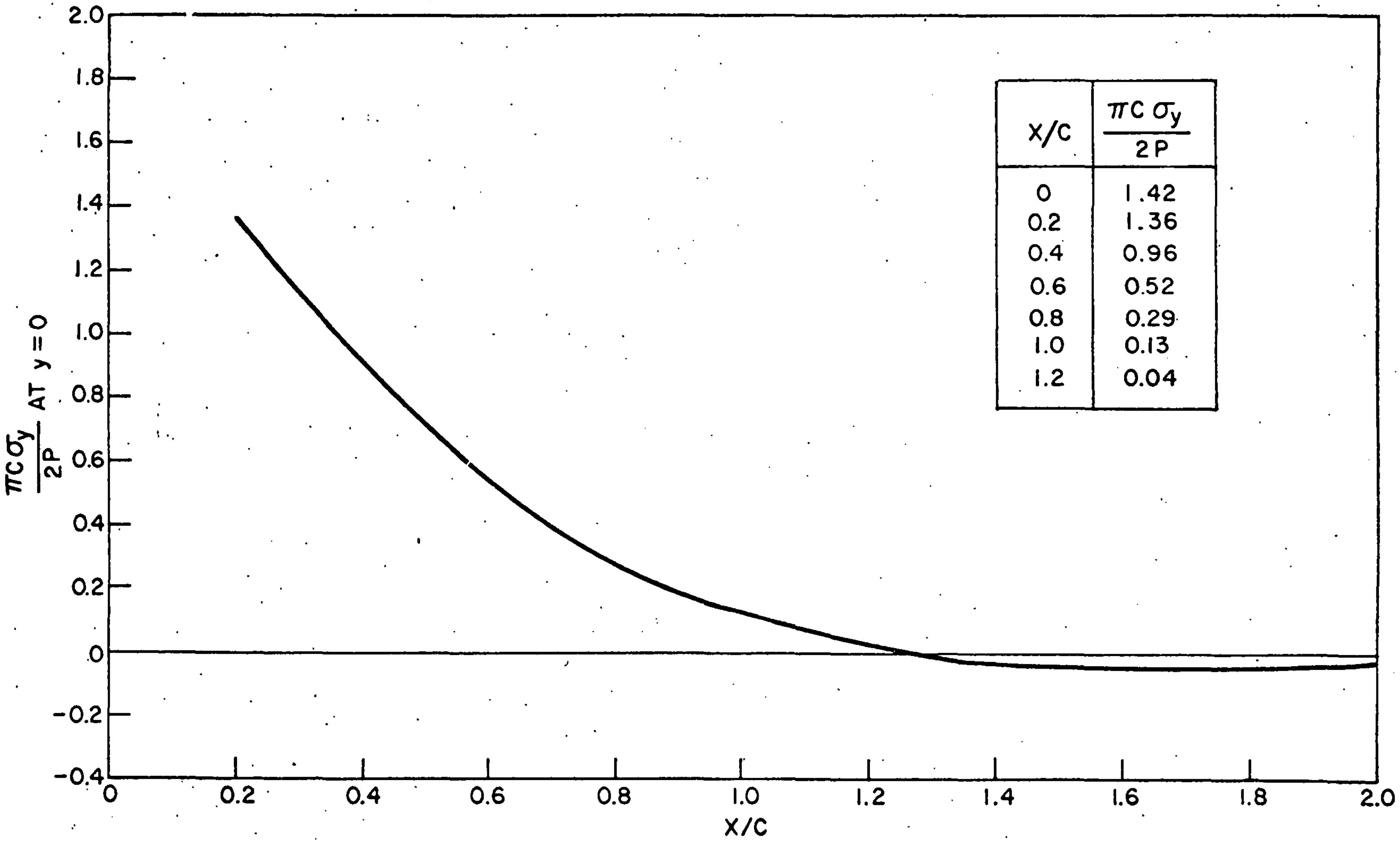

X/C	$\frac{\pi c \sigma_y}{2P}$
0	1.42
0.2	1.36
0.4	0.96
0.6	0.52
0.8	0.29
1.0	0.13
1.2	0.04

Figure 16. Filon's Solution

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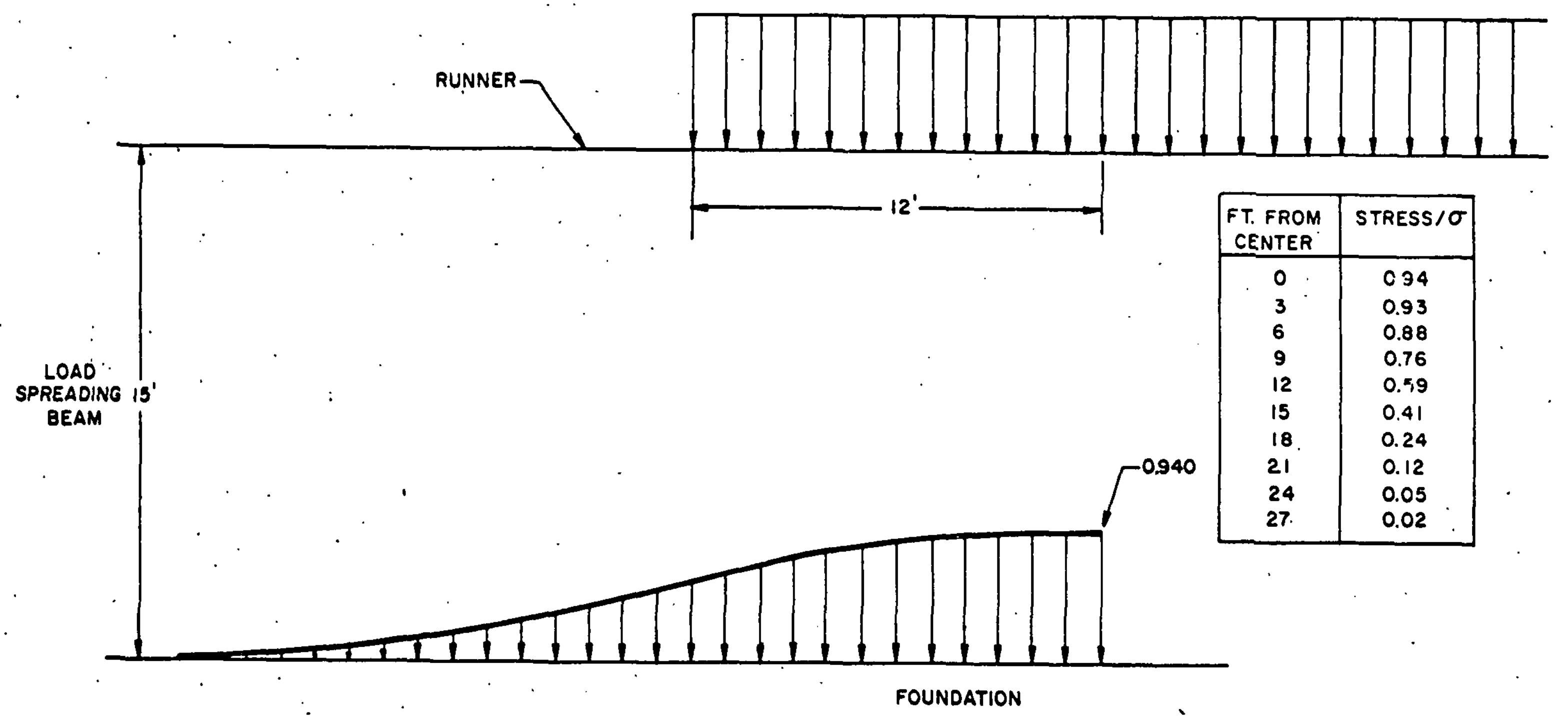
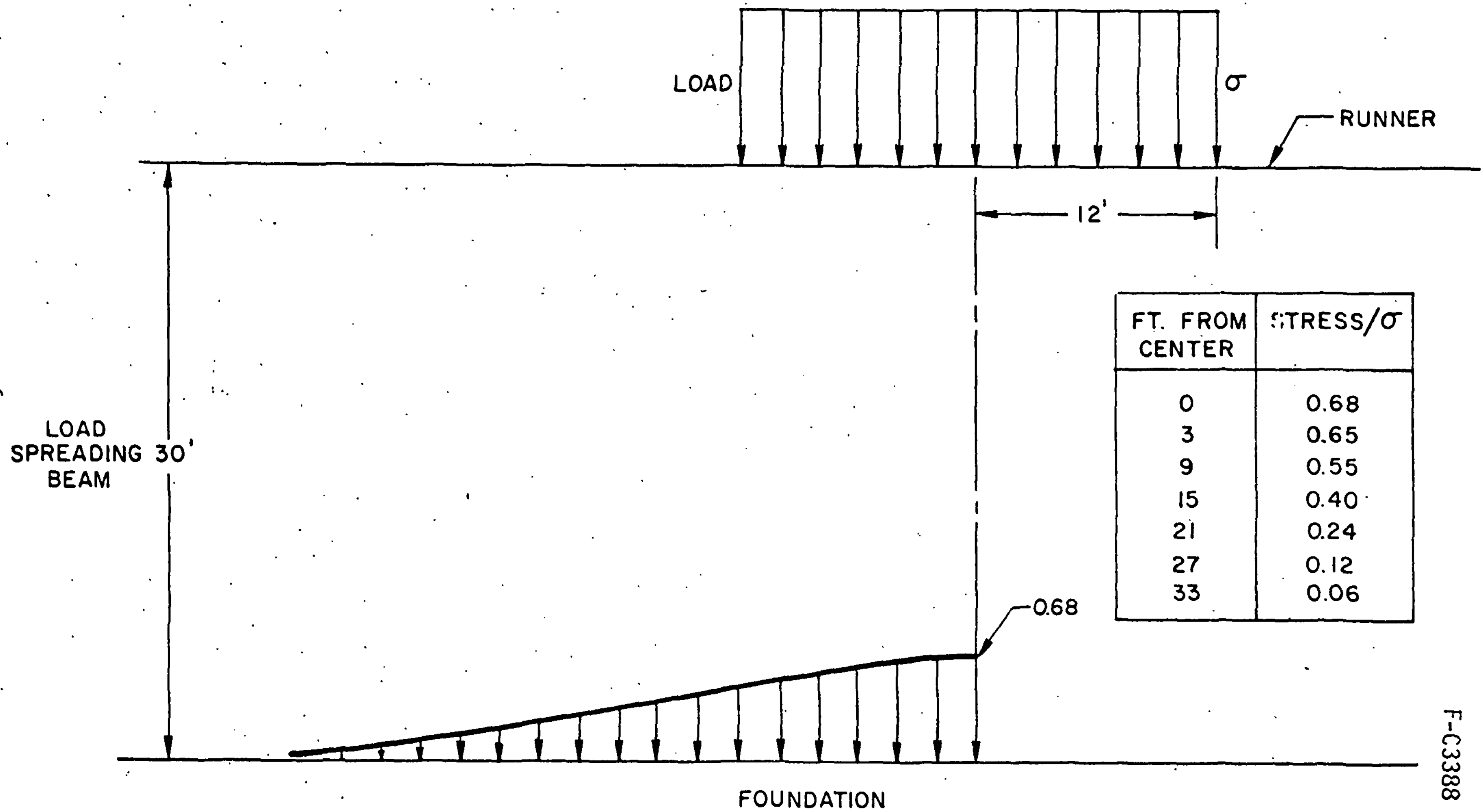


FT. FROM CENTER	STRESS/σ
0	0.94
3	0.93
6	0.88
9	0.76
12	0.59
15	0.41
18	0.24
21	0.12
24	0.05
27	0.02

Figure 17. Typical Load Spreading Solution - Circumferential Beams, 15' Deep Beams



FT. FROM CENTER	STRESS/σ
0	0.68
3	0.65
9	0.55
15	0.40
21	0.24
27	0.12
33	0.06

Figure 18. Typical Load Spreading Situation for Circumferential Beams, 30' Deep Beams

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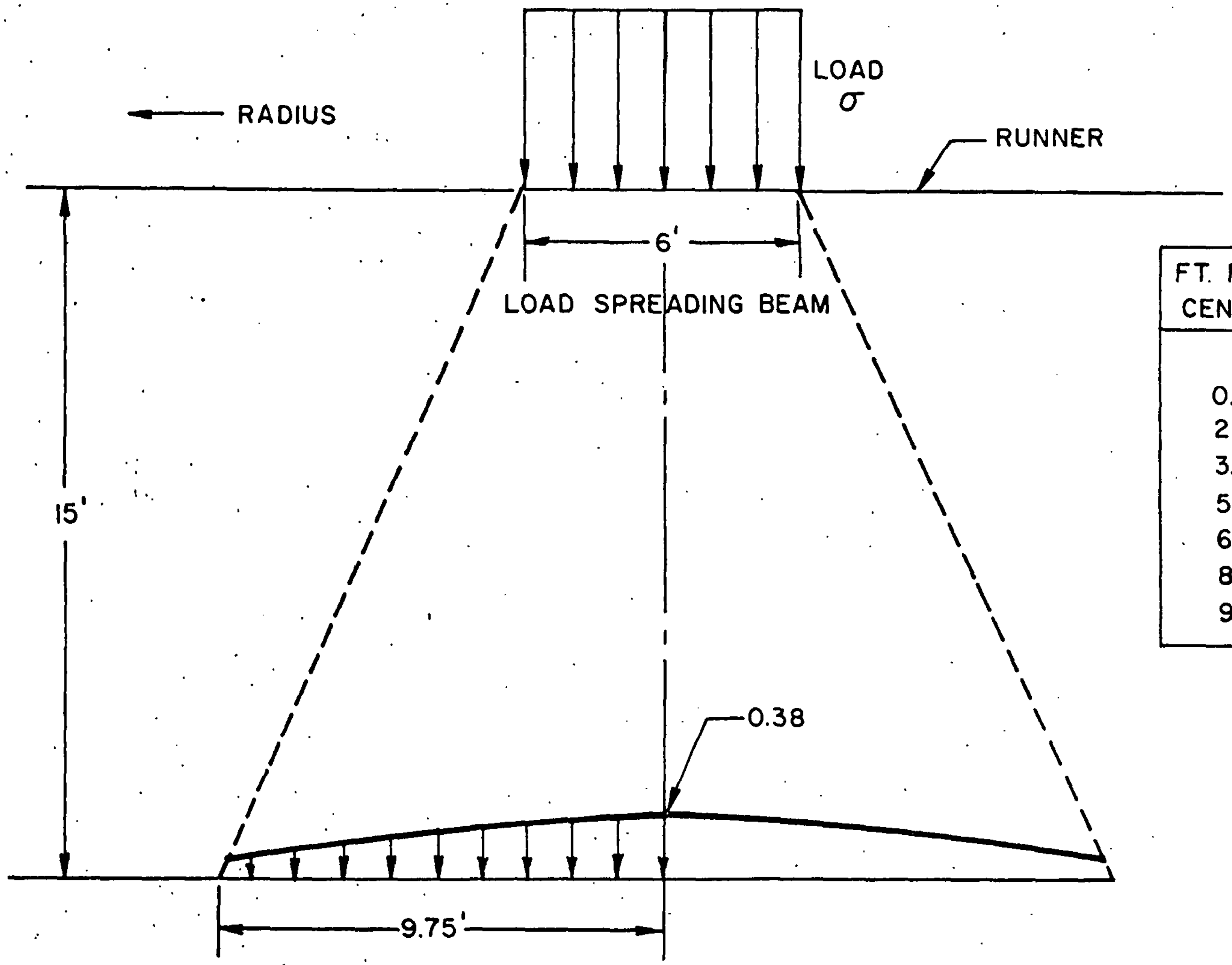

FT. FROM CENTER	STRESS/σ
0	0.38
0.75	0.37
2.25	0.34
3.75	0.31
5.25	0.27
6.75	0.22
8.25	0.16
9.75	0.12

Figure 19. Typical Load Spreading Situation with Radial Beams

the circumferential direction. Figure 19 considers the case of the load spread over a 6 foot width of runner. With a load spreading beam 15 ft in height, the load is spread to the point where the maximum at the center of the loaded region is 38% of the load on the runner. As far as the necessary radial width is concerned, Figure 19 also illustrates that at 10 feet from the midpoint the stress has already decreased to 10% of the maximum.

The above figures are based on computations which are valid for beams of infinite length, but the situation for finite beams is not going to be very different. Thus, the radial spreading of load accomplished by means of beams with a top section just as wide as the runner and a bottom section approximately 20 to 25 feet across, would satisfy completely the load spreading criteria. The dashed lines in Figure 19 illustrates the shape of such a beam.

The data given in Figures 17, 18 and 19 are all based solely on the inherent compression of the beams which support the runner. If the foundation on which the beams rest is in itself flexible, then the load will be spread a little more than shown in these figures. Sample calculations were carried out by means of a program which analyzes beams on elastic foundation. Some typical numbers that could represent compliance of the foundation were estimated.* The results obtained are shown in Figures 20 and 21. The data in Figure 20 correspond to supports spaced 10 feet on centers, with a stiffness of 2.7×10^8 lb/ft (the most realistic of the two sample values that were considered). Figure 21 shows the increased load spreading obtained in the case of the softer foundation. It should be pointed out that the latter value should be considered unrealistically soft. It can be seen from both figures that even with the help of a soft foundation and of beams of large size, it is almost

*"Runner Joint Stress Analysis - 210' Goldstone Antenna", performed for JPL by FIRL - Contract No. EX-515904, April 1970.

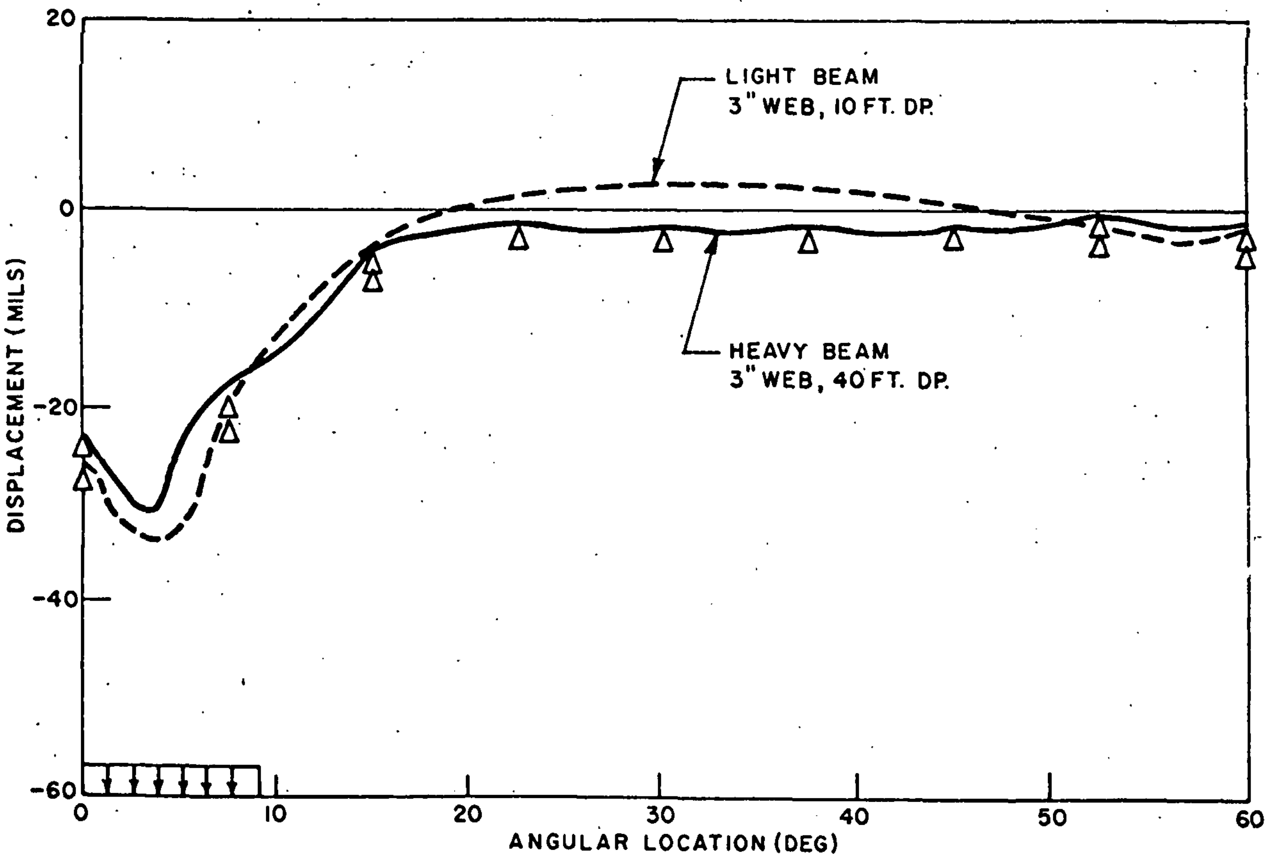

Figure 20. Load Spreading of Circumferential Beam on a Compliant Foundation

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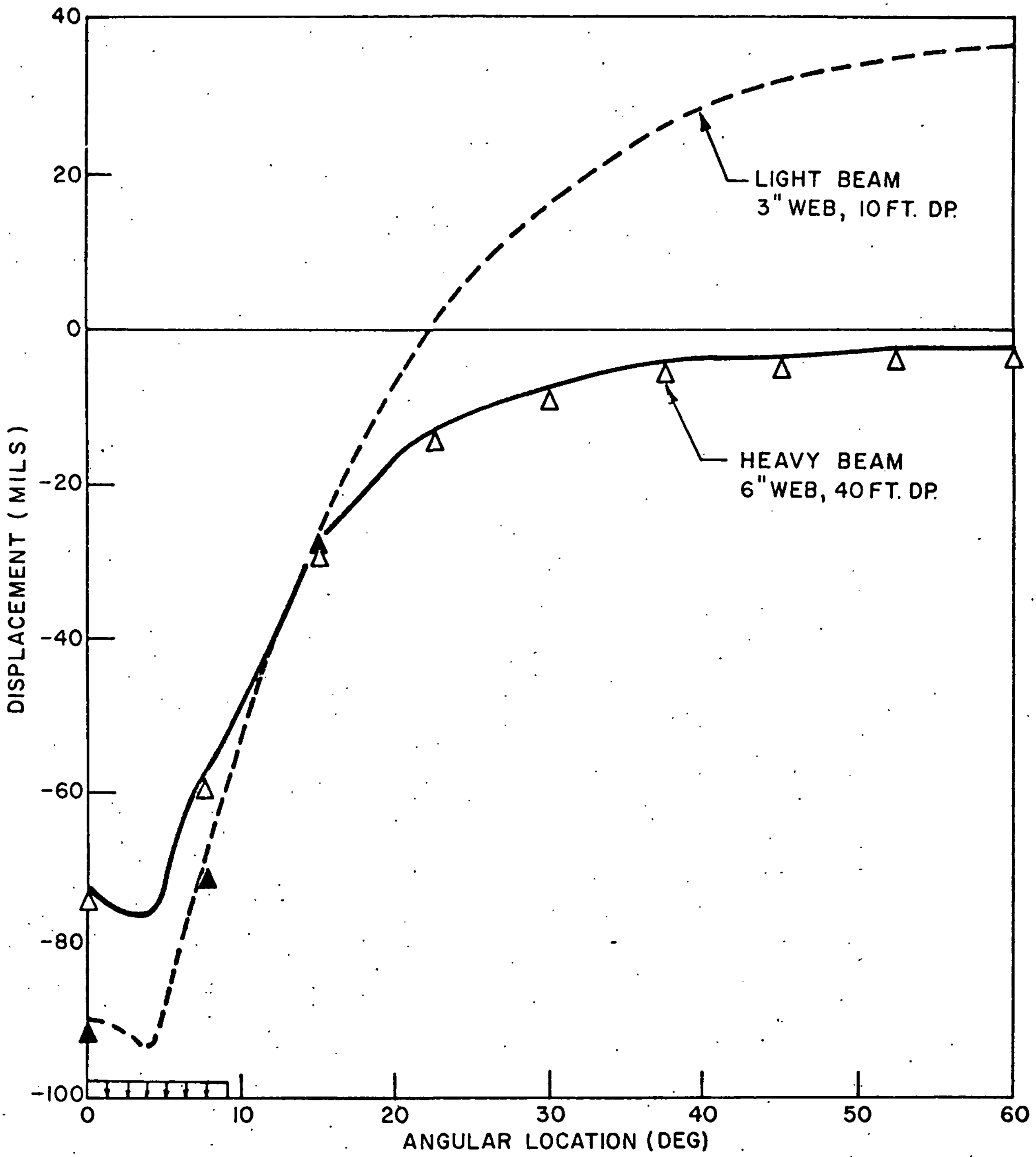

Figure 21. Load Spreading of Circumferential Beam on a Compliant Foundation

impossible to spread the load in this manner. Also a light beam of 15' depth and a heavy beam of 30' depth are shown to make very little difference.

Spreading the load in the radial direction can be implemented by the geometry which is depicted in Figure 22. This concept consists of many radial steel plates, which spread the load in the radial direction. Above these are three separate horizontal beams supporting the runner whose task is to eliminate the necessity of having too many vertical plates.

The runner provides the bearing surface for the pad, and must have sufficient cross-section to keep the deflection in the radial direction down to a minimum. For this purpose it is sufficient to have three supporting (deflection) beams under the runner to limit the runner deflection to approximately 5 mils.

The deflection beams resist deflection in the direction of motion, i.e., the circumferential direction. Five foot deep beams, resting on supports which are 6 feet apart, give deflection in the runner of less than 5 mils. For the purpose of these calculations the three deflection beams were taken to have 2 inch webs.

The load spreading plates transmit the load from the deflection beams and spread it in the radial direction, thus minimizing the peak loads experienced by the grout. From Figure 22 it is also evident that each one of the foundation plates resting on a grout pad is independent from any other grout pad. When any repair is needed to the grouted portion of the foundation, the load spreading plate can be removed in its entirety thus uncovering the grout, allowing for easy repairs. The runner support structure could be designed so that operation of the antenna with one load spreading plate missing would still be possible even if marginal.

The solution depicted in Figure 22 also results in keeping the oil 20 feet away from the grout, thus avoiding the problem that has occurred at Goldstone.

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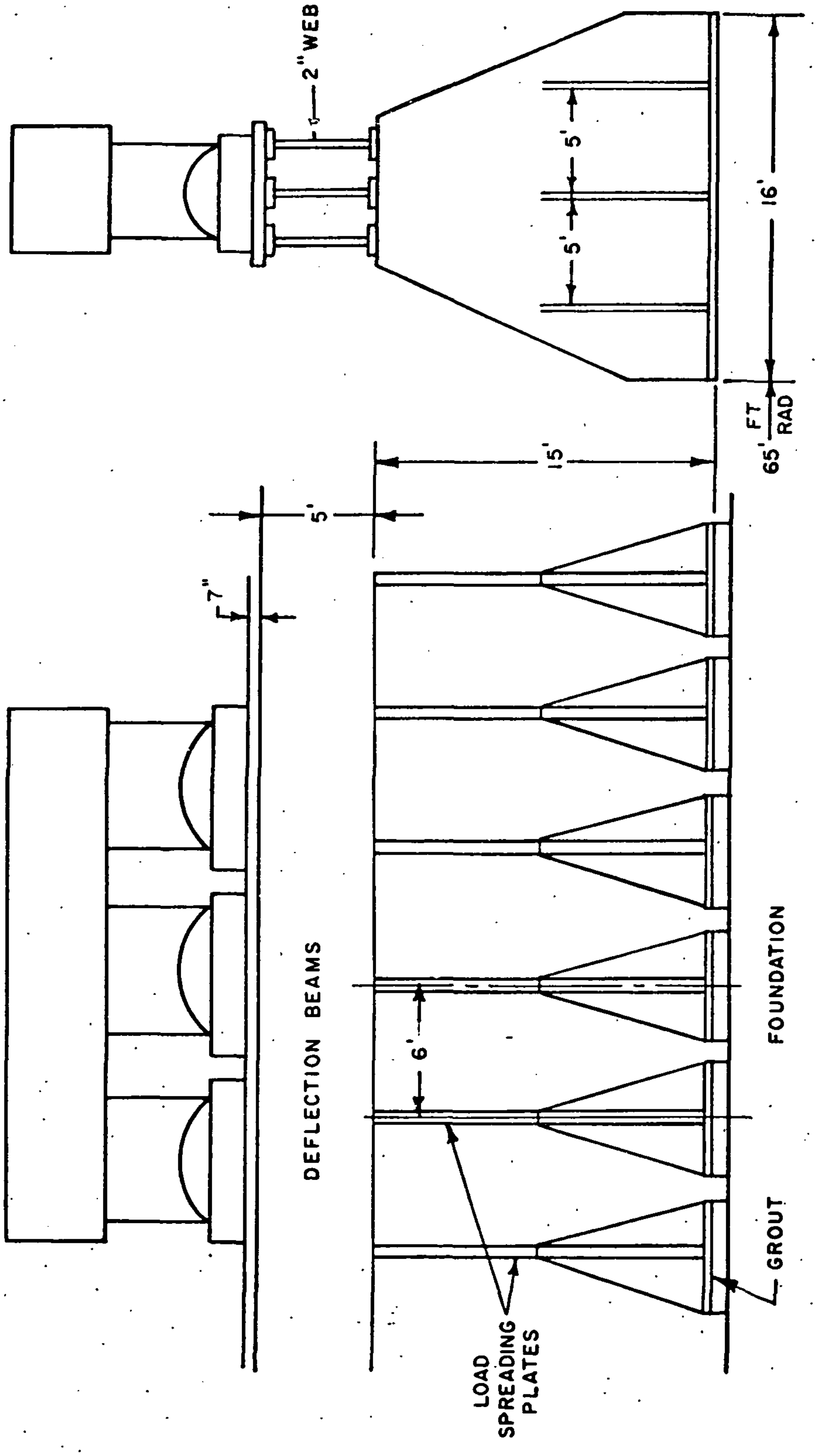

Figure 22. Runner Foundation Interface

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As far as the weight of this foundation is concerned, with load spreading plates of two inch thickness, the foundation runner interface would involve over 2 million lbs of steel, which is not excessive.

At the design stage of this foundation interface, it will be necessary to set up a deflection and stress program which will solve the problem implied by this steel construction. This is not a very expensive program to build because it can rely on pre-existing finite element programs in which the two dimensional plane stress type of problem is solved. It should be noted that the results of such a program would be much more reliable than those obtained from three-dimensional elasticity programs, such as the ones that would have to be used in the case where the runner pedestal is all in concrete. The operation of this type of program will also be much cheaper than that of a complete three-dimensional program.

6.3 GROUT INTERFACE

The grout interface introduced by Figure 22 is modular and would involve modules 6 feet by 16 feet. They would be further subdivided into 2 or 4 subsections, especially for the purpose of erection. The erection sequence would probably involve the installation of deflection beams on top of some of the loading spreading plates, and the leveling of the deflection beams by adjustment of the height below the load spreading plates in every fourth location, the grouting of these one out of four positions all the way around. Next the permanent attachment of the load spreading plates to the deflection beams and leveling of these intermediate points at the top of the deflection beam surface and grouting of all the remaining pedestals could be accomplished. The removal of a foundation load spreading plate would be performed by slight jacking up of the deflection beam above it and sliding it sideways.

The grout to be used for this purpose should be investigated well in advance of the start of the erection program, since even though provisions have been made for easy access to the grout surfaces, such repair

7. RUNNER

Some aspects of design of the runner will have to be treated with attention due to the fact that they may be rather delicate. They are the runner joints, the shimming mechanism, the stops which will prevent the runner from wandering off the foundation, the sealing, etc. Attention will also have to be given to the proper materials and sizes so that the manufacturing of such a huge piece of metal does not give excessive difficulty and so that transportation to the site will be possible.

The mounting of the runner on the foundation needs to be studied. It is probable that direct fastening of the runner on the deflection beams will not be possible due to the possible introduction of thermal expansion problems. Indeed, the runner will be essentially maintained at the temperature of the oil bath whereas the rest of foundation will follow much more closely the ambient temperature. This can give vast differences in the temperature, with the consequence of mismatching of the position of the runner and the foundation.

Before the special aspects of runner design are discussed it will be necessary to describe its support system in some detail. Spacing between the supporting (deflection) beams is 22 inches. This was selected in order to make the maximum deflection of the runner between supporting beams equal to that on the outboard side. This deflection amounts to just a few mils. During the actual design phase a choice will have to be made between using only two beams under and perhaps having to make the runner thicker in order to minimize the deflections, and using three beams as shown in Figure 23 with a thinner runner, such as 7 inches. The main advantage of the two supporting beam configurations is the fact that the leveling of the runner would be much simpler, and perhaps the total cost of the structure would be less.

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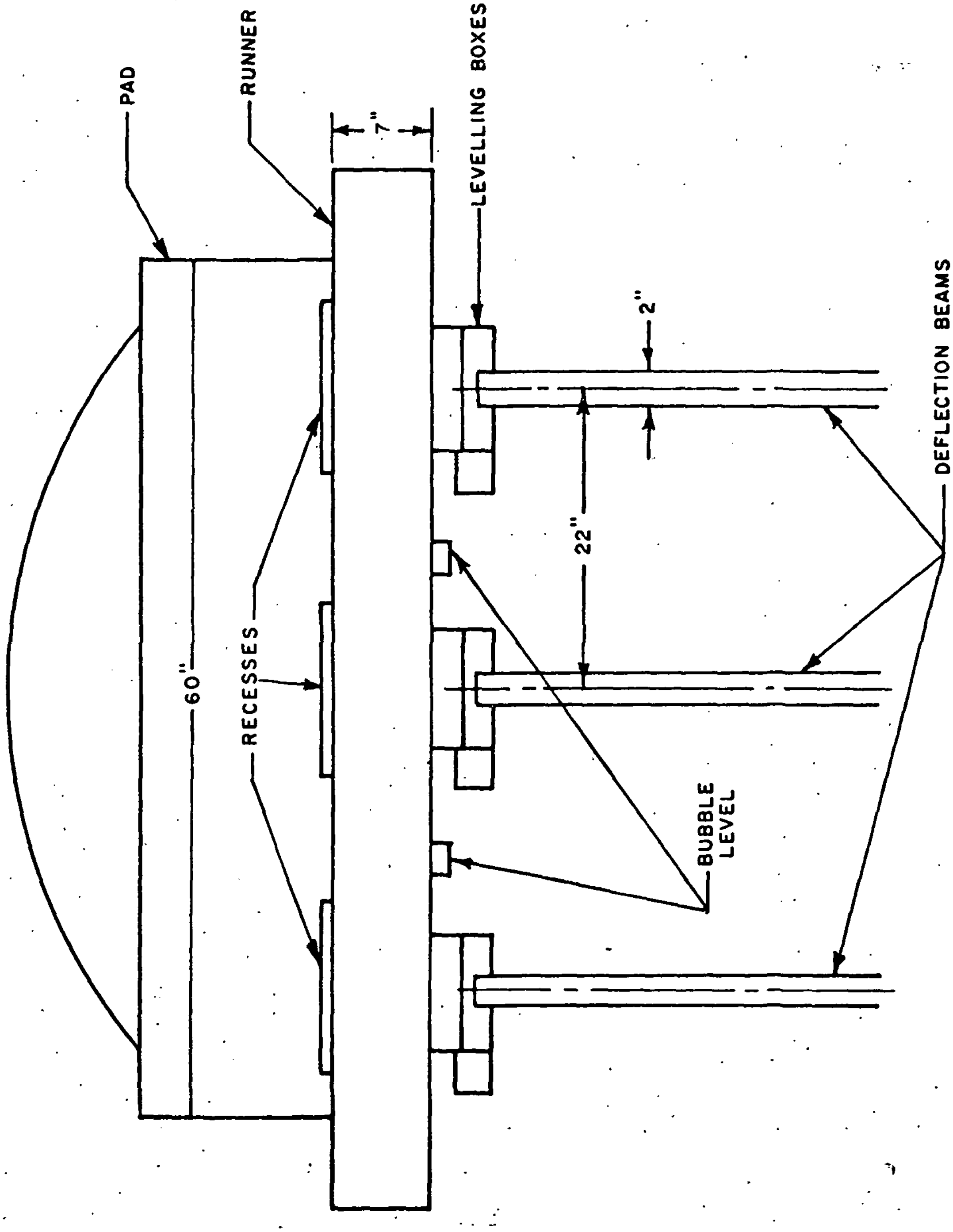

Figure 23. Runner Support and Alignment

Figure 23 also suggests that the leveling of the runner be made by means of leveling boxes which are nothing more than one wedge plate on top of another, driven by a screw and possibly locked by another screw. If screws with 10 threads per inch are selected, a taper slant of 3° would yield a height advance rate of 5 mils per screw revolution which is fine enough for all adjustments. In this situation a horizontal excursion of 5 inches in the leveling box wedges would result in a total vertical excursion of 1/4 inch. These alignment boxes would essentially serve the purpose of being adjustable shims, with proper proportioning to allow having a good distribution of sensitivity and range.

If the boxes are designed so that they fit properly on top of the deflection beams, it might be possible to avoid the use of a separate sole plate to provide to the runner the freedom to slide in order to compensate for thermal effects. The system of alignment boxes should be studied further to see if other concepts would be more practical, but they would definitely be preferable to a system of direct shimming both from the point of view of rapidity of execution and that of stability in time. These leveling boxes would have to be distributed evenly at close intervals in the circumferential direction, but actually could be spaced from one another as much as 10 or 20 inches on centers.

The monitoring of the levelness of the runner could either be done by means of hand levels or automatically by means of electronic bubble levels approximately one inch in diameter, which give an electrical output signal on two perpendicular axes, obtained by means of a capacitance sensing setup. The sensitivity of these instruments is commonly well beyond 1/10 of an arc second, which represents 30 microinches per inch or 1 mil per 3 feet. This sensitivity is more than sufficient for the purpose of leveling of the runner. A number of these levels could be permanently attached to the underside of the runner and biased electrically for lack of parallelism with respect to the top surface of the runner. Direct electrical monitoring of these levels could be done automatically by a computer system yielding periodic up-to-date information on the state of levelness of the runner. Similarly, leveling procedures could be developed which would be specified by the computer. All these schemes assume that absolute continuity can be achieved in all segments of the runner.

7.1 SEGMENT SIZE

Runner segments will have a size dictated by convenience of fabrication and transportation. Twenty-four segments of 15° inclined angle each are the maximum practical size which can be considered. The base rectangular slab would be 7.5' wide x 32.5 ft long x 8" thick and would weigh 40 tons. This size and tonnage is beyond the capability of Bethlehem Steel Co. and USS Steel slub mills and ingot sizes. Lukens Steel does have the ingot size and slab mill capacity to roll such sections. Machining the slab to a circular segment 6' wide with a 70' inner radius will reduce the weight to 30 tons per segment of 8" thickness. Ingersol Milling Machine Co., Rockford, Ill., for example, has ample planer miller capacity is machine segments of this size. Flatness within 0.002" over the entire segment on one side can be obtained. Flatness and parallelism of both sides would require wider tolerance limits, however, this is not of importance since the use of leveling devices is contemplated for the under side of the runner.

It is also possible that for assembly purposes a very short section of the runner could be used, if it were necessary for convenience of handling and so forth. However, in the latter case, a price might have to be paid in terms of thickness of the runner, complication of the sealing problem, and of the alignment problem.

The quality of machining of the runner top surface is not of too great an importance if the oil film thickness is maintained at 20 mils or so. However, the choice still exists of whether the runner should be final machined in a shop and transported to the site, or have the runner installed on the supporting structure approximately leveled, and then machined on the spot by means of a specially designed and fabricated grinding machine. Such a machine would be supported on reference rails mounted on the supporting structure. Two approaches could then be followed for the design of the machine. The first is to very accurately prelevel the rails and to construct them in a manner such that deflections and thermal effects are small enough not to bother the adjustment. Then the work of machining the runner can then be done in a relatively passive manner. The only measurement that would then be necessary is the measurement of level of the machined runner surface with respect to the rails themselves. The second approach that could be used is to put less accurate adjustment in the rail and make the machine more "active." This could be done by means of very accurate levels mounted on board such that the machine would establish a plane which passes through a predetermined reference point, and would maintain itself on that plane by active, continuous adjustment regardless of the lack of planarity of the rails themselves. In both of these cases the just machined portion of the runner would also be monitored with respect to the main portion of the machine and the position of the cutting wheel would be automatically adjusted so as to achieve the desired position. This would then compensate for tool wear and could make the procedure relatively automatic.

The decision of whether or not to build the second machine would depend on the results of conversations held with runner manufacturing people. The design and construction of the moving grinding machine would probably run in the $100,000 range and this is not counting the fabrication and installation of the rails.

In what has been said above, it should be noted that the need for surfacing and resurfacing of the runner is greatly decreased if a leveling mechanism is designed which can be readily utilized during the life of the antenna.

7.2 JOINTS

The problem of how to join segments of the runner requires thorough study by means of the deflection program, showing how the deflection

1°

pattern of the runner joint behaves when the antenna pads go over it. This can be accomplished by state-of-the-art programs now available, at least in the form of rather reliable solution techniques.

Due to the presence of the deflection beams, the runner joint will have to transmit only shear in order not to create a discontinuity when the antenna pads approach the joint. In such a case the same type of joint as is being used on the Australian and Spanish 210 foot antenna can be used. The possibility of using welds or other type of joints should also be investigated because they could at the same time solve the sealing problem. Due to the high value of oil film thickness which has been proposed, small discontinuities of the joints will not be very important except as they affect alignment. This would be further aided by a design where the satellite pads operate with film thickness larger than nominal at the two circumferential leading edges.

7.3 SEALING

The oil reservoir concept which was utilized in the 210-foot antenna is thought to be quite sound and much better than alternate approaches. It should therefore be maintained in the 420 foot telescopes. This then introduces the problem of sealing the reservoir walls where they meet the runner, sealing the runner joints when the high pressure pads are located on the top of a joint. The sealing problem is less important than what it was in the 210 foot antenna for two main reasons: (1) the very high oil viscosity employed will reduce the leakage substantially, (2) the grout is removed from the immediate vicinity of the oil, so the leakage of oil should not cause direct deterioration of the antenna support.

The best design approach is probably to attempt a reasonable sealing job by rather conventional methods and accept the fact that it would probably fail in spots. The failures could be compensated for by having the leakage collected and periodically sent back to the reservoir itself. Due to the fact that this is not a critical problem to the operation of the antenna, no further study will be made in this report.

7.4 STOPS

The runner is maintained at essentially a constant temperature by the presence of the oil reservoir surrounding it. On the other hand, the structure which supports the runner is constantly changing in temperature. Its dimensions will also change, thus causing relative motion between it and the runner. In order to avoid undue stresses in the connection between the two above parts, it is advisable to let the runner move freely with respect to the foundation on which it rests. This is accomplished above by the utilization of alignment or leveling boxes and flat top surfaces on which the runner can slide. Naturally the cyclic motion of the foundation under the runner will cause displacements which are not symmetrical thus inviting the possibility that the runner could walk off the foundation after a while. Further problems can be also connected with the runner "walking" in a circumferential direction. These last problems are not going to be given too much attention here, since a releveling procedure is contemplated and the greatest consequence of circumferential walk is change of runner level. As far as the runner walking off the foundation is concerned, it is necessary to impose some restraint to the motion of the runner. In the design of such constraints it is important to keep in mind that very large stress levels can occur as a result of temperature change.

A study of constraint geometry which would minimize the effects of off center walk of the runner was made when designing the 210 foot AAS antenna. That study concluded that it was sufficient to use springs located at either 3 or 6 circumferential positions and which impose a constraint simply to the radial motion of the runner. The main reason for using springs is that fixed stops have a tendency to generate very large stresses when changes in dimension due to changes in temperature occur, while the position of the antenna feet prevents relative motion of the runner with respect to the foundation. The springs will instead constitute a "memory" to the fact that the runner is out of position, eventually causing the runner to move back where it belongs, when the feet of the antenna move to a more favorable position. Figure 24 shows the position of the antiwalk springs, and the detailed concept on which their design was based at the time of the study for the 210 foot antenna.

The design procedure for these stops is to consider a deformation of the runner into a configuration which is no longer circular, but which essentially maintains the foundation diameter only at the position of the antenna feet. This configuration then stresses the spring the design of which is such as to allow almost the entire motion of the runner that would have occurred if the springs had not been there. This gives rise to low stresses because the runner is bending in its own plane. The springs then provide a bias toward a concentric position so that the runner can move back when the temperature cycle ends. Likewise, if the distorted position of the runner is "frozen" by the fact the antenna clamps the runner to the foundation, the spring provides a memory for the runner to go back when the antenna feet are moved elsewhere.

No problem is associated with this area since in the above mentioned design effort, it was proven feasible and easy to implement. Figure 24 of this report shows that a convenient geometry for the radial spring is that of a beam deflecting in the plane of the runner.

7.5 DEFLECTION PROGRAM

The key item in the runner design will be the assembly of an effective deflection program coupled with the appropriate bearing computer program representing the hydrostatic bearings and their resulting pressure distribution. This would allow the obtaining of as uniform a film thickness distribution as possible. The runner deflection program would have to be assembled from presently available techniques and portions of programs which are of the finite element type. A plate approach is thought to be suitable for the runner itself, but it must include the two dimensional study of the underlying deflection beams which support the runner. These beams would furthermore be resting on the load-spreading plates which can actually be represented as springs.

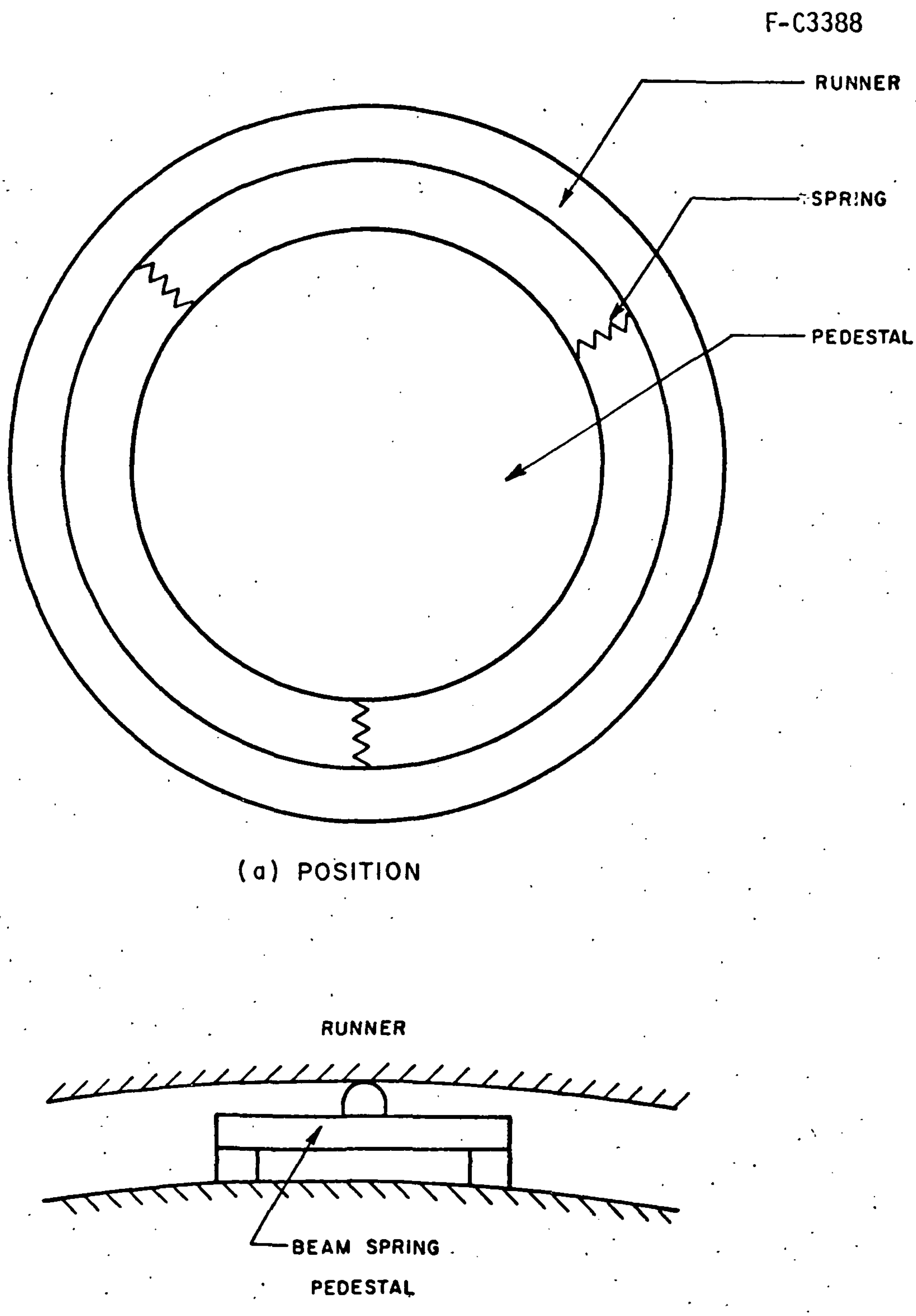

Figure 24. Anti-Walk Springs

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These programs must be usable both for the study of the continuous sketches of runner and for the study of the case where the pad is close to a runner joint. Runner joints could cause complications in the plate approach to the runner study and if it is proven necessary to use a joint which does transmit shear or moment or both it will also be necessary to represent the joint region by means of a three-dimensional elasticity program which must be tied in with the two-dimensional representation of the continuous portions of the runner. Therefore, this becomes an important point to ascertain and the study of such an eventuality would have to be made as soon as possible. This is due to the fact that three-dimensional programs are always difficult to assemble and check out and expensive to operate. Therefore, as much time as possible should be taken in trying to avoid their use.

8. CONCLUSIONS

8.1 GENERAL

Hydrostatic oil pad type of support for the azimuth bearing of a 420 foot antenna weighing approximately 50 million pounds is feasible within the present state-of-the-art.

A few areas will have to be given particular attention and must be considered to have a long lead time. These areas are listed below in two groupings depending upon their need for additional antenna studies by JPL or others prior to initiation.

8.2 NO ADDITIONAL ANTENNA STUDIES REQUIRED PRIOR TO INITIATION

A preliminary study is required of the effect of runner joints from the point of view of transmission of shear and moment. A preliminary study of this area of the antenna design must be performed to establish whether or not a three dimensional computer program representing the runner joint or a testing program to study the same effects will be necessary.

Preliminary attention should be devoted to a study of grout, grouting methods, and establishment of life and environmental criteria dependent on the operation of grout under cyclic loads. The purpose of this study would be to establish what kinds of environment are not harmful and what magnitude of maximum loads are considered safe for long life of the grout interface.

Preliminary studies should be conducted of the recirculating pumping and filtering of very viscous oil at high pressures. A high level of viscosity (approximately 100×10^{-6} reyns or 3000 Saybolt Universal Seconds) is desirable in the hydrostatic oil pads. Operating pressures should be approximately 1500 psi, with the capability to raise to 4500 psi

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for short periods of time under emergency operating conditions. The possibility of oil-cooling after the pumping and filtering should be investigated. Some effective means should be devised for removing impurities such as sand, chips, and condensed water from the hydraulic system. This might be accomplished by means of a system which scavenges the parts of the reservoir which are most likely to collect such impurities.

8.3 ADDITIONAL ANTENNA STUDIES REQUIRED PRIOR TO INITIATION

The runner support structure should be optimized into a geometry which could then be programmed for determination of deflection patterns, and optimized in terms of weight, ease of construction, and maintenance.

A consistent and easy leveling scheme should be developed for the runner. Both initial execution and measurement and re-leveling or correcting capabilities must be included.

Loading levels on the hydrostatic oil film pads will need to be established more accurately. It should be remembered that a decrease in the weight of the moving part by a factor of two or possibly three would enable consideration of compliant surface bearings in the hydrostatic pads. This would introduce a number of advantages in the construction and operation of the azimuth bearing system.

Advanced studies are recommended to develop a computer program for calculating pad deflections including effects of the runner support structure.

Development of an automatic grease feeding scheme would be desirable for the self-aligning ball socket joints.

Respectfully submitted

John H. Rumbarger
Program Manager

W. Shapiro, Manager
Friction & Lubrication Lab.

Z. Zudans, Director
Mechanical and Nuclear Engineering Department

F-C3388

Appendix

A

HUMBLE OIL DATA SHEET

DG-2C

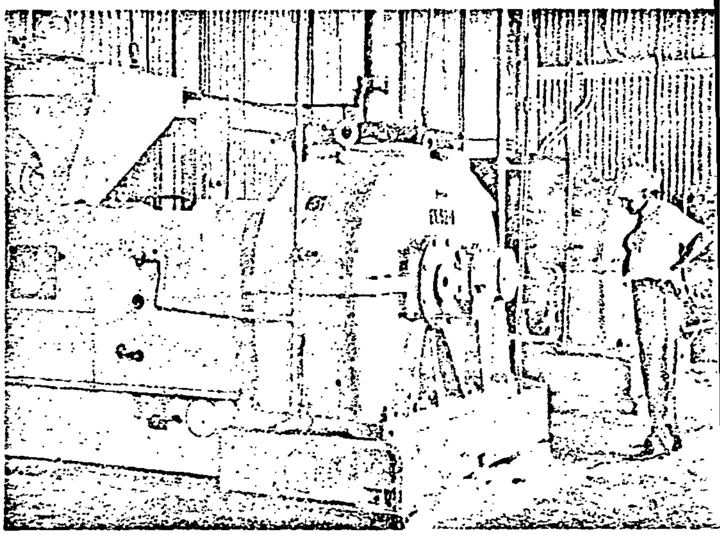
THE FRANKLIN INSTITUTE RESEARCH LABORATORIES
BENJAMIN FRANKLIN PARKWAY • PHILADELPHIA, PENNA. 19103

HUMBLE
OIL & REFINING COMPANY

DG-2C
Data Sheet

SPARTAN® EP

High-temperature industrial gear oils



The 7 SPARTAN EP gear oils are designed for use at operating temperatures up to 225°F. Unlike the more conventional mild-EP gear oils, the SPARTAN EP products are not formulated with lead and sulfur extreme pressure agents. Consequently, they are not subject to the formation of insoluble sulfur-lead compounds at elevated temperatures.

Increasingly, industrial gear drives are being employed in operations that involve continuous high temperatures. Such applications occur in the processing of rubbers, plastics, chemicals, and even paper. The high temperatures result from heat soak-back from the process itself as well as from the heavy power loads involved. It is for applications like this that SPARTAN EP is expressly designed.

Properties

SPARTAN EP lubricants are formulated with high-quality paraffinic base oils and a special chemical EP additive. The presence of a stabilizing agent fortifies the resistance of this additive against the deteriorating influences of heat and oxidation and assures dependable performance over the service life of the lubricant. The base oils, themselves, have high resistance to oxidation.

Typical Timken OK loads, which range from 40 to 50 or more pounds, are well above the 30-pound minimum required by the AGMA standards for mild-EP gear oils.

Typical Inspections: SPARTAN EP

Grade	0	1	2	3	4	5	6
Gravity, °API	28.0	27.0	26.0	25.0	23.0	22.0	19.5
Viscosity at 100°F	315	465	700	1000	1500	2150	3150
at 210°F	54.3	62.0	74.3	88.2	110	133	165
Viscosity Index	105	100	98	96	94	91	90
Flash Point, °F	415	435	435	445	450	460	470
Pour Point, °F	−5	−5	0	5	10	15	20
Timken OK Load	40	45	50	50	50	50	50
Copper Corrosion, 3 hr at 212°F	pass	pass	pass	pass	pass	pass	pass
AGMA No.	2EP	3EP	3, 4EP	4, 5EP	6EP	7EP	8EP
ASTM Viscosity No.	315	465	700	1000	1500	2150	3150

Printed in U.S.A. 7-30-65

These values are sufficiently high to provide protection against intermittent shock loads—which cause welding and gross damage. Under continuous overload conditions—which cause excessive wear—SPARTAN EP will provide superior protection against such wear. This is indicated by the results of the 4-Ball Wear Test, in which SPARTAN EP 3 shows scar diameters of 0.38mm for steel-on-steel (50-kg load) and 0.35mm for steel-on-bronze (10-kg load), both tests run at 600 rpm, 75°C, for 1 hour.

SPARTAN EP has particularly good water-separation properties. For example, SPARTAN EP 3 gives the following results in the Dynamic Demulsibility Test (at 180°F): top sample, 8% water and emulsion; bottom sample 100% water and emulsion. These results are considerably better than commonly accepted standards.

All products are non-corrosive to iron, copper, and bronze alloys, they are resistant to foaming, and they have high viscosity indices.

Grades

Each of the 7 grades conforms to an ASTM Viscosity Classification grade; and, as shown in the table of typical inspections, each grade meets one of the AGMA standards.

Oil Mist Lubrication

For mist-lubricated gears that require extreme pressure oils, lead-compounded lubricants are not recommended. The potential danger of the lead compounds in stray mist in the atmosphere precludes their use. In contrast, SPARTAN EP contains no lead and can be used in oil mist applications without any unusual precautions.

Applications

SPARTAN EP is used wherever an industrial EP gear oil is required to serve under unusually high temperatures. It is suitable for heavily loaded gear sets—including worm drives—and for gears that are subject to shock loading. It is also recommended for heavily loaded plain and antifriction bearings, particularly in circumstances where low speeds or intermittent motion prevent the formation of a full fluid lubricating film.

All grades are suitable for circulating lubrication systems, although the lower viscosity grades are more commonly preferred for such applications.

In general, grade selection can be based on the appropriate AGMA No. recommended for the particular application.

Compatability

In order to achieve the full performance advantages of SPARTAN EP lubricants, it is recommended that they not be mixed with lead-sulfur-compounded gear oils. However, under emergency conditions, where a complete oil change is precluded, there is no hazard in mixing these two types of gear oils.