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ADVANCED PAYLOAD TRANSFER MEASUREMENT SYSTEM (APT-MS) MECHANICAL FEATURES

Dr. William V. Brewer
Professor
Technology Department
Jackson State University
Jackson, Mississippi

KSC Colleague - Eduardo Lopez del Castillo
Robotics

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It's been fun guys and I hope we can do it again sometime.

ABSTRACT

Purpose: APT is a relatively inexpensive electro-mechanical device that reduces both time and manpower required to make shuttle payload transfers.

Objective: Explore possible mechanical sources of measurement error.
Develop an improved prototype design that is compact, inexpensive, and addresses the measurement error issues.

Motivation: During the last two feet or so of transfer, for a large or heavy payload to its restraining fixture, the consequences of unplanned contact (or impact) between payload and support structure can range from merely annoying to something approaching disaster.. Current transfer methods employ technicians with meter sticks stationed at the various hold-down locations to estimate the distances to contact. This information is communicated orally to the crane operator. It is understandable that this work proceeds carefully and therefore slowly. APT measurement system would provide a GUI for the "move-conductor" (crane operator) so he could see the displacements of all hold-down interfaces as they move together and thus have a more accurate, comprehensive and "real-time" picture of the engagement activity.

Accomplishments:
An error model attempts to include all estimatable sources of mechanical error.
Design features were introduced to reduce or eliminate major sources of error.

SUMMARY

Purpose: APT is a relatively inexpensive electro-mechanical device that reduces both time and manpower required to make shuttle payload transfers.

Objective: Explore possible mechanical sources of measurement error.
Develop an improved prototype design that is compact, inexpensive, and addresses the measurement error issues.

Description: APT has a spring loaded retractable cable that is attached to a moving object whose location must be monitored. The drum (or reel) on which the cable is stored is gimbled and instrumented to supply the spherical coordinates of the moving object relative to the APT device. These may be converted to rectangular coordinates or whatever form best facilitates a useful GUI for the operator trying to guide the moving object.

Motivation: During the last two feet or so of transfer for a large or heavy payload to its restraining fixture, it is difficult to avoid undesirable contact (or impact) between payload and support structure. Current transfer methods employ technicians with meter sticks to estimate the distances to hold-down locations. This information is communicated orally to the crane operator. It is understandable that this work proceeds carefully and therefore slowly. APT measurement system would provide a GUI for the "move conductor" (crane operator) so he could see the displacements of all hold-down interfaces as they move together and thus have a more accurate, comprehensive and "real-time" picture of the engagement activity.

Accomplishments: An error model attempts to include all estimatable sources of mechanical error which fall into two categories: cable guide errors, bearing friction errors.

Estimates of positional error for the first prototype ranged from .5" to nearly 3" as the radial distance approached the limit of 72" for the device. Since the goal is an error on the order of .050" at a radial distance of 24", this is clearly unacceptable.

Cable guide errors could account for as much as 2/3 of the error, some effort was expended designing a better one. The so-called Zero Clearance Cable Guide was proposed as a possible solution to this part of the problem.

Bearing Friction is difficult to model. We have drawn from several sources to put together a model with as much of the geometric characteristics of the bearings and their operating environment as are available rather than relying on empirical aggregated numbers for specific bearings.

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I

INTRODUCTION

1.1 HISTORY

APT-MS is the brain child of Eduardo Lopez. Concept investigation of mechanical aspects was initiated 6/94 by N.S.Malladi [1], followed by work of E.Lopez and R.Bennett since then [2,3]. During that time an earlier hardware prototype was built and tested [2], primarily to develop the electronics associated with reading of angular displacement sensors. The electronic work was done by others in this multi-disciplinary effort. Electronic aspects are indispensable to the results of the project, but are beyond the scope of this discussion of mechanical functions.

1.2 MOTIVATION

Undesirable contact (or impact) between a large, cumbersome payload and its restraining fixture is difficult to avoid during the last 2ft. of transfer (Figure 1-1). Current transfer methods employ technicians with meter sticks to estimate the distances to hold-down locations. This information is communicated orally to the crane operator. It is understandable that this work proceeds carefully and therefore slowly. APT measurement system provides a Graphic User Interface (GUI), shown in Figure 1-1, lower left. A "move-conductor" can see displacements of all hold-down interfaces as they move together and thus have a more accurate, comprehensive and "real-time" picture of engagement activity. Optical devices are an alternative shown in Figure 1-1, lower middle, but these may be more expensive and require greater stand-off than the mechanical APT system discussed here.

1.3 DESCRIPTION

APT has a spring loaded retractable cable that is attached to a moving object whose location must be monitored (Figure 1-1, lower right). The drum (or reel) on which the cable is stored is gimbled and instrumented to supply spherical coordinates for the moving object relative to an APT device. These may be converted to rectangular coordinates or whatever form best facilitates a useful GUI for an operator trying to guide the moving object.

Figure 1-2 shows the cable drum or spool nested inside a cable guide cage or pointer. These elements interact but rotate on separate horizontal bearing shafts: cable shaft on the right; cage on the left. These in turn are carried in the uprights of a "Y" shaped yoke whose lower stem shaft pivots about a vertical axis on bearings set in the fixed base. At the outer end of each shaft is an optical encoder to read one of the spherical coordinates.. The base mounted encoder measures "yaw" angular displacement of the yoke stem code wheel. On the left, a yoke mounted encoder measures the "pitch" angle of the cage shaft and therefore of the cable-exit, guide-tube. On the right, a yoke mounted encoder measures rotation of the cable storage drum shaft. Radial coordinate measurements subtract cage rotation from drum rotation and convert the difference to length of cable played out. Cable retraction is accomplished with a "clock" spring mounted between cage and drum.

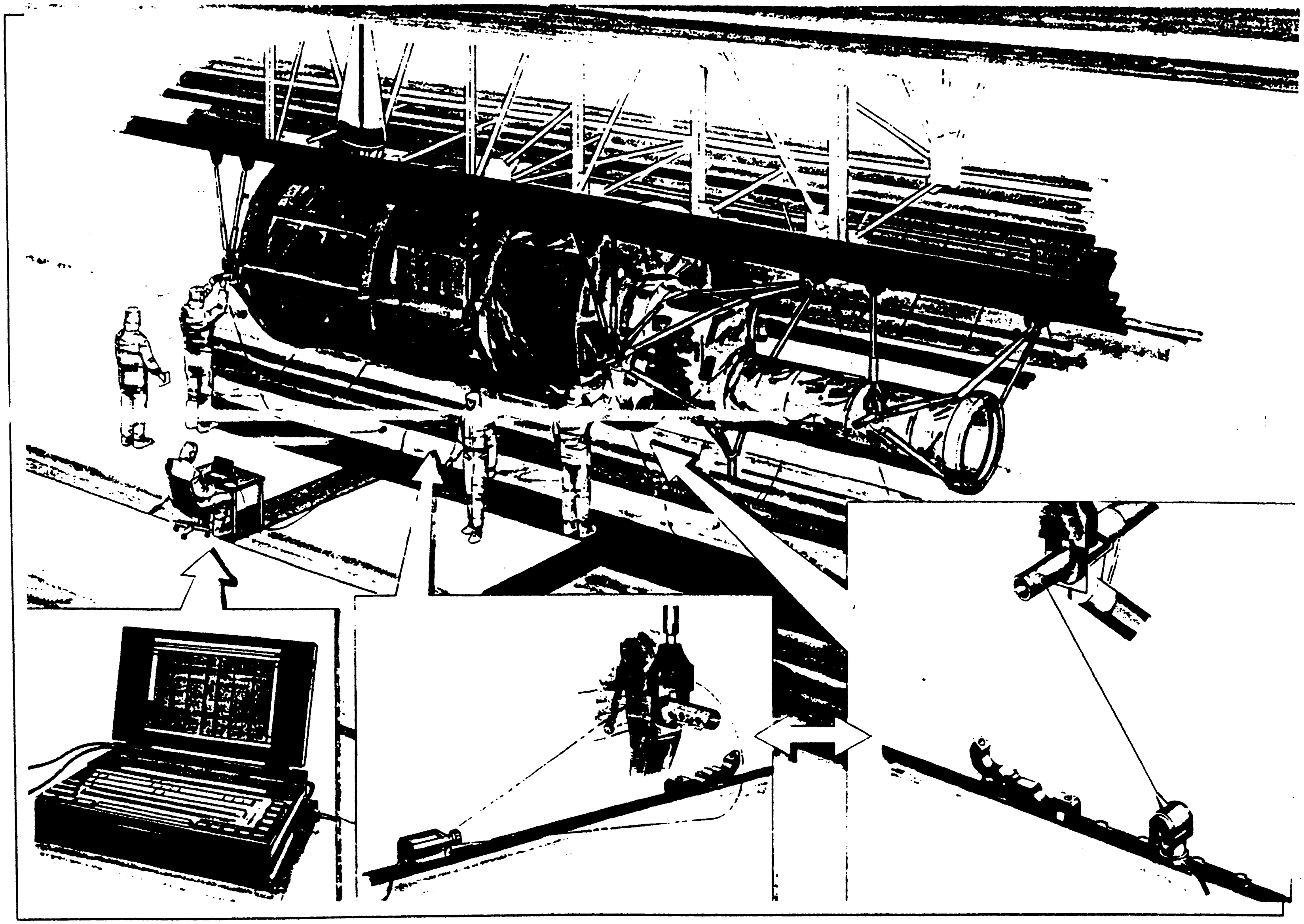

Figure1-1: APT–MS Work Environment Senario

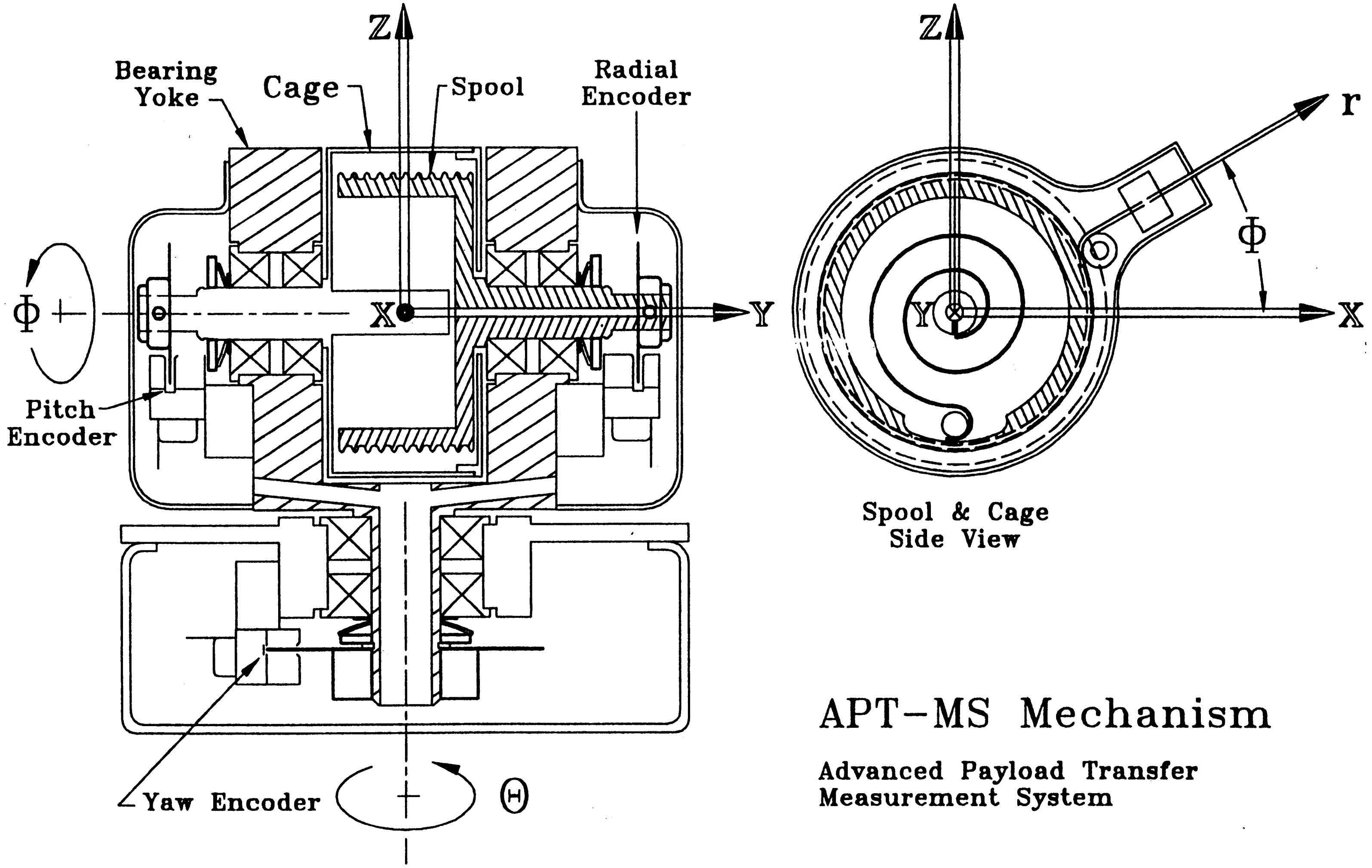

Figure1-2: Assembly Schematic

II

ERROR SOURCES

Estimatable sources of mechanical error considered, fall into two categories:
Cable clearance errors, Bearing friction errors.

2.1 CABLE GUIDE

In this case the term "cable guide" is an inverse of its usual meaning. Cable tension is used to guide cage rotation about both horizontal and vertical axes and thus obtain pitch and yaw angle readings. To understand this function, consider the diagram in Figure 2-1.a. Initially the cable is horizontal and the pitch angle reads zero. Then the object to which the right end of the cable is attached moves upward. This should be reflected with a change in pitch angle reading which is controlled by cage rotation. Only tangential components (i.e. perpendicular to the radial) of cable tension are able to produce moments about cage (or yoke) rotational axes and thus bring the cage cable guide in line with a new cable direction. The two error sources indicated above prevent a perfect alignment.

2.2 CABLE CLEARANCE

A cable guide contains the last 2 points where cable contacts the cage. Clearance should be minimized at these points. To whatever extent the cable may move without contacting the last bearing point, an error is introduced. Figure 2-1.b shows orifices of radius R being guided by a cable of a smaller radius r . Initially the cable is taken as horizontal where the object to which it is attached has been moving down from above the horizontal axis. The cable is therefore bearing on the lower surfaces of guide orifice peripheries. If the moving object then reverses its direction and returns upward, a distance $e = 2(R-r)$ is traversed by the cable between lower and upper exit contact points before a moment producing force can be applied to rotate the cage. This rotational lag is cable clearance error. It is approximated as Er for small angles ϵ as indicated in Figure 2-1.b .

2.3 BEARING FRICTION

As cable presses against the cable guide exit orifice, shown in Figure 2-1.a, resistance to cage rotation is expressed as a moment M, causing the cable to deflect until a tangential component of cable tension produces sufficient counter moment for equilibrium. Cable deflection results in a pointing error Er. Similar deflections are caused by rotation resistance about the vertical bearing axis.

An error model attempts to capture as many of the contributing effects as possible. Information from several bearing vendors contributed to the model. Computations reflect those methods that consider the somewhat more complicated and variable static loadings and geometry of this bearing application. Computations using MathCad 5.0 are displayed on the following pages:

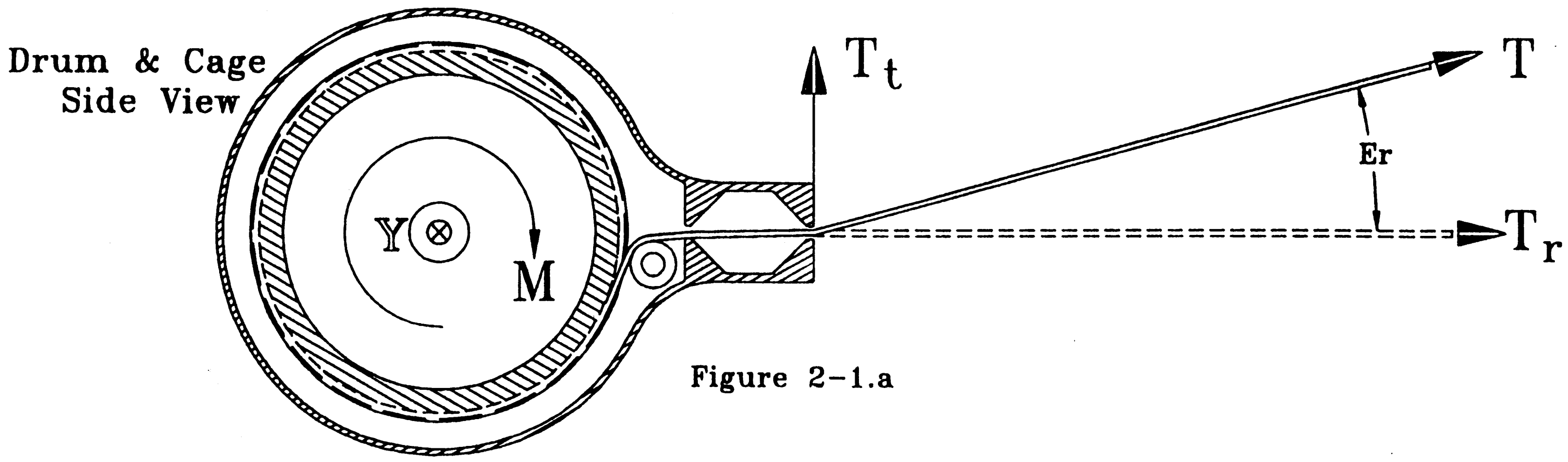

Figure 2-1.a

$$\tan(\varepsilon) = \frac{e}{s} = \frac{Er}{d + s}$$

$$Er = \frac{e}{s}(d + s)$$

$$Er = e \left(\frac{d}{s} + 1 \right)$$

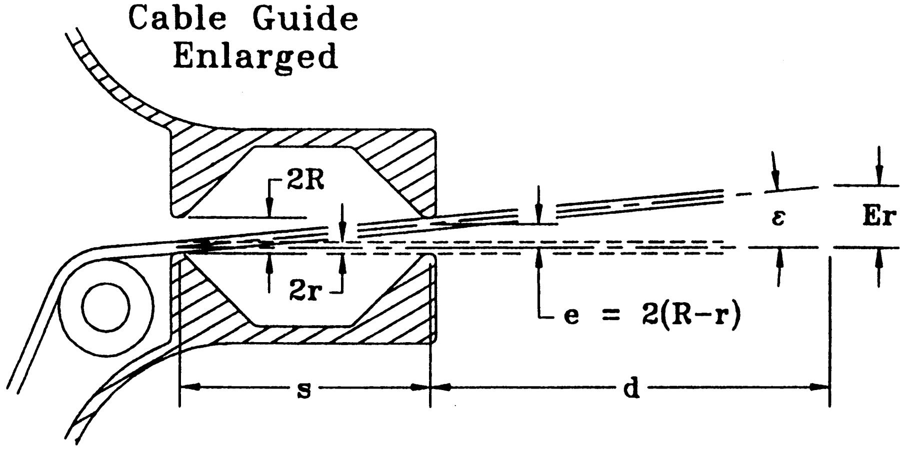

Figure 2-1.b

Figure 2-1: Error Sources

APT - MS

ERRORS from Rotational Resistance

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Tension in cable
$T := 5.000$

Spring force,
z component
$Sz := 0$ (neglect)

$d := 24.0$
from tube exit
to cable end

$rr := 1.0$ reel radius
$l := 2.0$ from y axis to tube exit
$a := 1.5$ from y axis to cable guide
$b := 0.0$ y distance free cable from reel to guide

$\alpha := \mathrm{asin}\left(\frac{rr}{l}\right) \quad \sin\delta := \frac{b}{\sqrt{a^2 + b^2 - rr^2}}$

$i := 1..2$ z, y axis bearings
$j := 1..4$ ball locations
$\gamma := 90 \cdot deg$ z axis from vertical
$\theta := 90 \cdot deg$ x axis from horizontal
$\phi_i :=$ pitch angle

$\begin{bmatrix} 45 \cdot deg \\ 0 \cdot deg \end{bmatrix}$ always zero

BEARING GEOMETRY

$\beta_i :=$ bearing contact angle
$\begin{bmatrix} 10 \cdot deg \\ 11 \cdot deg \end{bmatrix}$

$c_i :=$ center distance of bearing set measured
$\begin{bmatrix} .4155 \\ .3798 \end{bmatrix}$

$pd_i :=$ pitch diameter of bearing measured
$\begin{bmatrix} .800 \\ .632 \end{bmatrix}$

effective center distance
$C_i := c_i + pd_i \cdot \tan(\beta_i) \quad \rho_i := \frac{C_i}{2} \cdot \cos(\beta_i)$

effective pitch diameter
$PD_i := pd_i + c_i \cdot \cot(\beta_i) \quad \rho_i := \frac{PD_i}{2} \cdot \sin(\beta_i)$

ρ_i
$\begin{bmatrix} 0.274 \\ 0.247 \end{bmatrix}$ effective moment arm

ρ_i
$\begin{bmatrix} 0.274 \\ 0.247 \end{bmatrix}$ check

$\beta c_i := \mathrm{atan}\left(\frac{C_i}{PD_i}\right)$

$\frac{\beta c_i}{deg}$ check
$\begin{bmatrix} 10 \\ 11 \end{bmatrix}$

BEARING LOADS

assumes all loads
additive

Sources:

Weight, preLoad, Tension in cable

$W_i := \begin{bmatrix} 5.000 \\ 1.000 \end{bmatrix} \quad pL_i := \begin{bmatrix} 0.0 \\ 0.0 \end{bmatrix}$

$Tr_1 := T \cdot \cos(\phi_1)$
$Rx := T \cdot (2 - \cos(\alpha)) \quad Rx = 5.67$
$Rz := T \cdot \sin(\alpha) + Sz \quad Rz = 2.5$
$Wr_1 := W_1 \cdot \sin(\gamma) \quad Tr_2 := \sqrt{Rx^2 + Rz^2}$
$Wr_2 := W_2 \cdot (\sin(\theta) + \cos(\theta) \cdot \cos(\gamma))$

$Ta_1 := T \cdot \sin(\phi_1)$
$Ta_2 := T \cdot \sin\delta$

$Wa_1 := W_1 \cdot \cos(\gamma)$
$Wa_2 := W_2 \cdot \cos(\theta) \cdot \sin(\gamma)$

moment
arms:
$r_1 := 0$ always zero
$r_2 := 0$ Tsinδ not zero but neglected for simplicity
$a_i := \begin{bmatrix} 1.837 \\ .9283 \end{bmatrix}$

Radial Loads: $Pr_i := Wr_i + Tr_i$

$Pr_i = \begin{bmatrix} 8.536 \\ 7.197 \end{bmatrix}$

Axial Loads: $Pa_i := Wa_i + Ta_i + pL_i$

$Pa_i = \begin{bmatrix} 3.536 \\ 0 \end{bmatrix}$

Moments: $M_i := r_i \cdot Pa_i + a_i \cdot Pr_i$

$M_i = \begin{bmatrix} 15.68 \\ 6.681 \end{bmatrix}$

APT - MS

ERRORS

from

Rotational Resistance

page 2 of 2

BEARING REACTIONS

$$R_{i,3} := \frac{Pr_i}{2\cdot\cos(\beta_i)} \qquad R_{i,3} = \begin{array}{|c|}\hline 4.334 \\ \hline 3.666 \\ \hline\end{array}$$

$$R_{i,2} := \frac{Pa_i}{2\cdot\sin(\beta_i)} \qquad R_{i,2} = \begin{array}{|c|}\hline 10.18 \\ \hline -1.003\cdot 10^{-15} \\ \hline\end{array}$$

$$R_{i,1} := \frac{M_i}{2\cdot\rho_i} \qquad R_{i,1} = \begin{array}{|c|}\hline 28.607 \\ \hline 13.54 \\ \hline\end{array}$$

$$R_{i,4} := R_{i,1} + R_{i,2} + R_{i,3} \qquad R_{i,4} = \begin{array}{|c|}\hline 43.121 \\ \hline 17.205 \\ \hline\end{array}$$

ROTATION RESISTANCE

Rolling Resistance of Bearing Ball

$f := .0018$ (this factor obtained by R.B. from F.A.G. bearing Co.)

$$Mrr_i := \frac{pd_i}{2}\cdot f\cdot\left(2\cdot\sum_{j=1}^{4} R_{i,j}\right) \qquad Mrr_i = \begin{array}{|c|}\hline 0.124 \\ \hline 0.039 \\ \hline\end{array}$$

Moment rotation resistance

ERROR from ROLLING RESISTANCE

$$Er_i := \left(\frac{Mrr_i}{T\cdot l\cdot\cos(\phi_i)}\right)\cdot d \qquad Er_i = \begin{array}{|c|}\hline 0.422 \\ \hline 0.094 \\ \hline\end{array}$$

Error resulting from Mrr

2.4 ERROR MODEL

An explanation of model parameters will serve to both indicate error sources and describe APT device functions (also see Figure 1-2). Symbols are listed in roughly the order of their introduction in the error model on the previous two pages, together with a discussion of their significance where appropriate:

T	cable Tension rotates the cage (beneficial) and increases bearing loads (detrimental).
Sz	radial Spring force is included for completeness but neglected in calculation.
d	distance from cable guide exit to the outer end of the cable. Two feet or less is believed to be the range of interest for most payload transfers.
rr	reel (or drum) radius used to compute the radial component of spherical coordinates.
l	length from y rotational axis to cable guide exit.
a	distance from y rotation axis to cable guide entrance.
b	y distance spanning free cable between leaving drum to contact with cable guide.
α	angle wrt x-axis of internal cable between drum and cage cable guide.
δ	angle between internal cable tension component and the y-axis.
i	$i = 1$ pertains to bearings on the y-axis, $i = 2$ those of the z-axis.
j	$j = 1$ to 4 pertains to bearing ball locations in each bearing set (cage or yoke).

The foregoing assumes for purposes of explanation that, the z-axis is vertical, x and y axes are horizontal, and the cable is in an x-z plane. The error model allows the z-axis to tilt away from the vertical causing the x-y plane to be tipped wrt the horizontal. Bearing loads attributed to component weights are affected.

γ	angle between z-axis and the vertical.
θ	angle of x-axis from intersection of x-y plane and horizontal plane.
ϕ	pitch angle of cable wrt x-y plane.

Pitch angle must be limited since the error in rotational position measurement about the z-axis becomes unbounded as the pitch approaches 90degrees.

2.4.1 BEARING GEOMETRY. Rolling resistance is a function of normal forces on the bearing balls which in turn depend on bearing geometry: bearing pitch diameter (pd); center distance between bearings (c); contact angle between ball and race (β). Effective moment arm for eccentric bearing loads is a computed quantity depending on the above physical parameters. Effective center distance (C) and pitch diameter (PD) are intermediate computations on the way to finding the effective moment arm (ρ).

2.4.2 BEARING LOADS. Contributions to aggregate bearing loads come from three major sources: weight of mechanical components (W); bearing preloads (pL); and tension in the cable (T). These have been broken into their contributions to orthogonal components, (Pr) radial or perpendicular to bearing axes, (Pa) axial or parallel to bearing axes, and (M) the aggregate moment that both of these forces cause. Of the load sources, T and W are treated in some detail. Preload has merely been added to the axial component of aggregated bearing loads. Physically, preload behavior is somewhat more complicated. In this application preloads are expected to be small therefore modeling of them has been postponed. They are included to recognize there presence but then neglected.

2.4.3 BEARING REACTIONS are the normal forces (R) acting on a diameter of the bearing ball where it is in contact with the races. Reactions are modelled as if they are all applied to just 4 bearing balls located in the plane of the aggregated loads (Pa and Pr). Two balls for each bearing are at the extremes of their respective pitch diameters.

2.4.4 ROLLING RESISTANCE is expressed as a moment (Mrr) resulting from tangential forces in the same fashion as friction on a sleeve bearing. The factor (f) is used like a friction coefficient but is not the result of sliding contact. We do not have a very certain number for this crucial quantity.

2.4.5 ERROR (Er) varies directly with rolling resistance (Mrr) and cable played out (d) and inversely with cable tension (T), distance from rotation axis to cable exit (l), and cos(ϕ) of pitch angle. Since Mrr is also a function of T, the expression can be put in a form similar to

$$Er(T) = K + C * (W/T) \qquad \text{where K and C are constants}$$

Increasing T reduces the error attributed to component weights (W) but does not affect those caused by T, expressed here as K.

Clearly, the cable direction cannot be allowed to approach the z-axis since a pitch angle of 90 degrees results in unbounded error.

Only compactness of the mechanism prevents the obvious alternative for error mitigation: increasing the distance from rotation axis to cable exit (l).

Rolling resistance can be reduced in various ways: reduce factor f ; increase bearing separation c; reduce component weights; increase cable tension; etc.

III

DESIGN DETAILS

3.1 DRUM & CAGE

Subassembly shown in Figure 3-1 calls out the various components that were not included in schematic Figures 1-2 and 2-1. Major components listed first are called out in the central column. Most components are indicated in both top (left) and side (right) views. Parts will be referenced by their number in the discussion that follows.

3.2 FUNCTION

Cable 19 is drawn from a helical storage groove in drum 3 . Tension is maintained in the cable with retraction "clock" spring 4 . The spring is anchored to the cage 2 by insertion of a tab at the inner end into a slot in post 5 , and fixed to drum 3 by a 180 degree loop at the outer end engaging the spring stud 10 . Cable is guided to the exit with rollers 6 and finally with the cable guide 1 . Set screw 11 fastens cable to drum. Counter weights 12 mounted on stud 13 serve to balance the heavier exit end of the cage subassembly on its bearing shaft.

3.3 CABLE GUIDE

Zero Clearance Cable Guide, part no.1, Figure 3-2, is a response to cable clearance error discussed in section 2.2 and illustrated in Figure 2-1 above. Thread on the HOUSING left end attaches to cage 2 with a nut. Thread on the right end is used by a CAP to compress the CORE. The CORE has two greatly reduced sections which act as thin-walled tubes. Axial compression causes the tubes to bulge radially inward reducing cable clearance which can be adjusted to the minimum tolerable. To insure that tubes bulge inward instead of outward as is their wont, voids surrounding the tubes are filled with silicon rubber. Rubber is considered incompressible under normal conditions. As the volume containing it is compressed, the metal housing prevents outward expansion therefore it will move against the tubes forcing them into the cable path. The core and its tubes are teflon to reduce cable friction.

3.4 CAGE

Guide cage 2 and its components are detailed in Figure 3-3. Rollers 6 are identical and made of teflon to reduce friction. Likewise the studs on which they are mounted are identical. Roller studs are mounted on the cage in " T " slots so their position can be adjusted to accommodate a range of cable and roller diameters. Thread on the snap ring groove end of the studs serves only to secure the outer (radial direction) roller cavity dust cover and has no function in the inner roller location.

APT-MS
Drum & Cage Sub-Assembly

no.	description	req'd.	mat'l.
1	cable guide	1	sub-assmb
2	guide cage	1	aluminum
3	cable drum	1	aluminum
4	clock spring	1	s.s.
5	spring post	1	aluminum
6	guide roller	3	teflon
7	roller stud	3	c.r.s
8	T-nut, #5-40	3	c.r.s.
9	snap ring	3	steel
10	spring stud	1	c.r.s.
11	set screw	1	#5-40
12	contr. wt.	N	Pb
13	c. w. stud	1	c.r.s.
14	std. nut	3	#5-40
15	std. nut	1	.25-20
16	drum cover	1	aluminum
17	roller cover	1	aluminum
18	flt. hd. scrw.	4	#2-56
19	cable, .030"d.	1	coated
20	cable terminal	1	sub-assmb
21	spring cover	1	aluminum

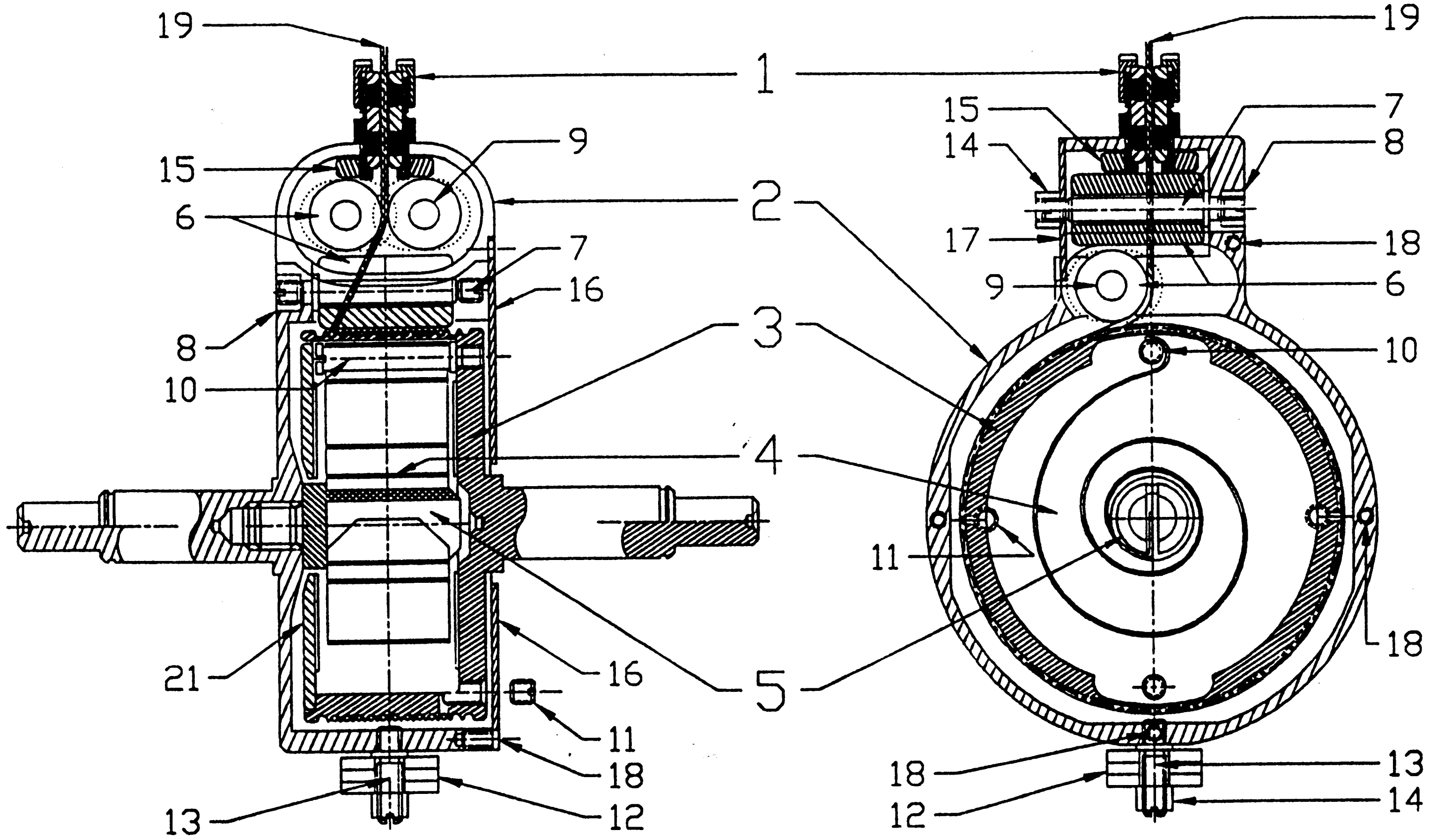

Figure 3-1: Drum & Cage Sub-Assembly

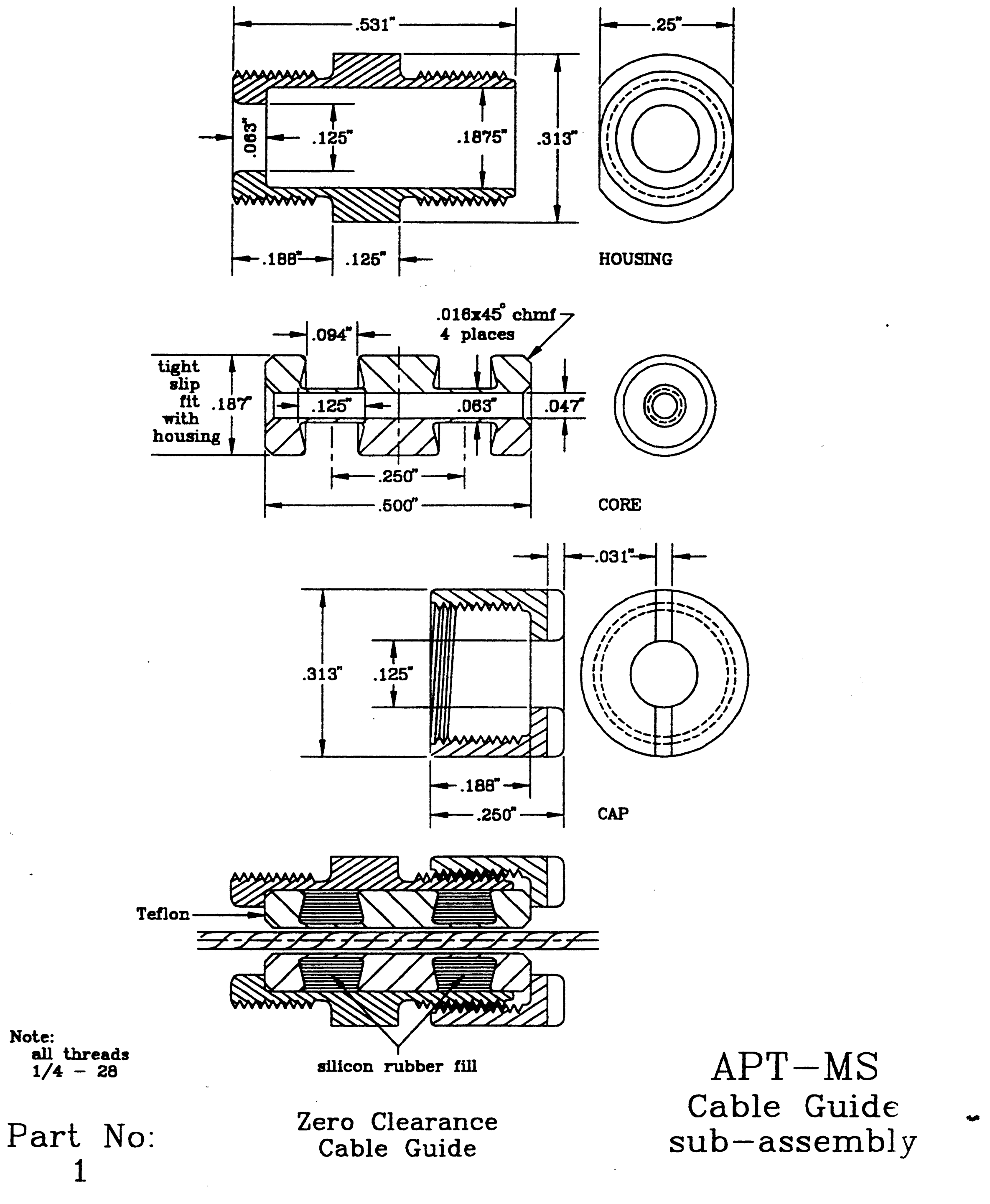

Figure 3-2: Zero Clearance Cable Guide

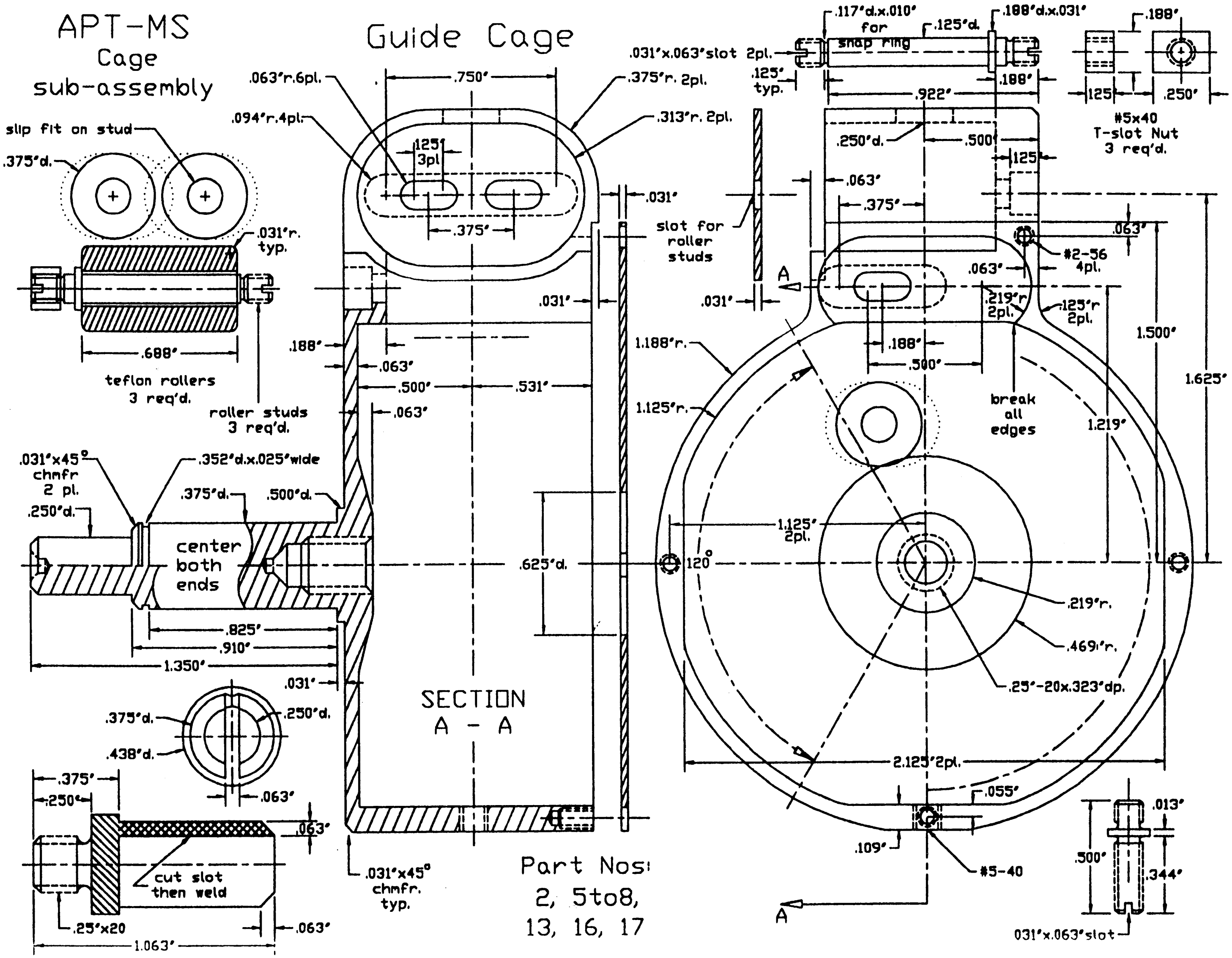

Figure 3-3: Cage Details

3.5 DRUM

Cable drum 3 and components are detailed in Figure 3-4. Explanation of the crater like feature opposite the left end of the shaft may be appropriate. When Drum 3 and Cage 2 are being assembled, the spring 4 is inaccessible and invisible. The crater fits over the right end of spring post 5 forcing the spring against the base of the post at its left end thus centering it in the spring cavity.

3.6 TERMINALS

Cable terminals, Figures 3-5 and 3-6, part no. 20, are not shown in the DRUM & CAGE subassembly. Friction error caused by these features has not been evaluated.

3.6.1 RING - ENDS, Figure 3-5, allow 360 degrees of pitch rotation but only 90 degrees of yaw motion. Stand-off may be required to prevent interference with the yaw motion Roll rotation is not required and may be detrimental to performance. A "fish line swivel" can be introduced between terminal and cable to nullify some of the latent cable roll which may be present due to varying tension.

Reference points on an object being tracked are not completely fixed relative to that object when ring type terminals are used. If the center of the least diameter on the mounting finial is taken as reference then it is fixed for pitch rotation of the terminal. Yaw motions cause this reference to move thru an arc:

$$s = r\theta = +/-- \ (.086'')(45\pi/180) = +/-- \ .068''$$

where $r = .086''$ is the least finial radius and yaw is +/-- 45deg.. If the center of yaw rotation is the reference then it is fixed relative to yaw rotations but moves with pitch rotation. The same formula above describes the arc length of travel.

3.6.2 BALL - ENDS, Figure 3-6, achieve a stable reference point. Disadvantages to this approach include: higher cost, less compact, and rotations limited to a cone angle of less than 180 degrees. Practical cone angles are much less. Angles of less than 90 degrees are used in the design shown in Figure 3-6. Friction increases with cone angle of the finial due to higher normal forces caused by "wedging" action of the ball in its socket.

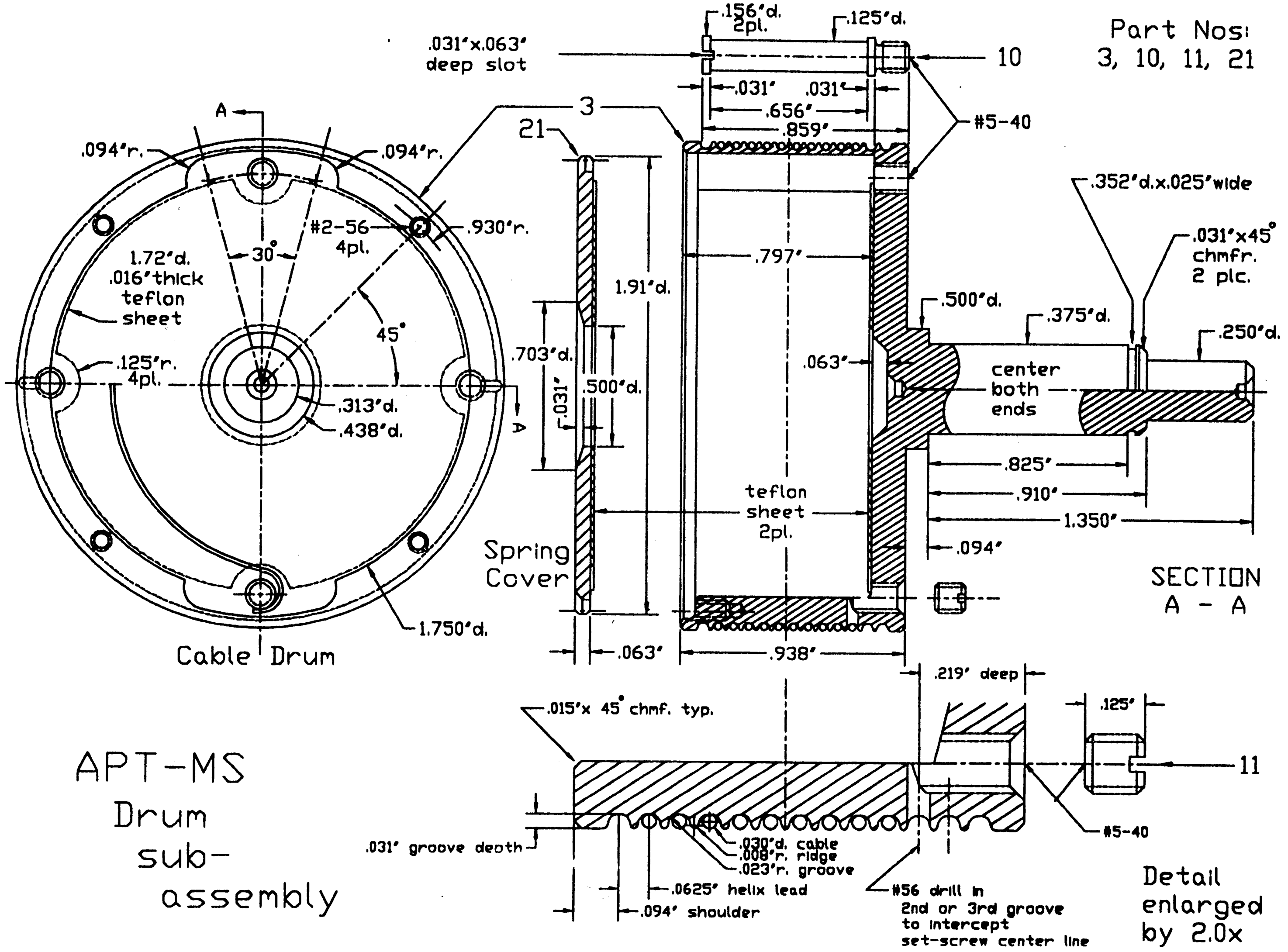

Figure 3-4: Drum Details

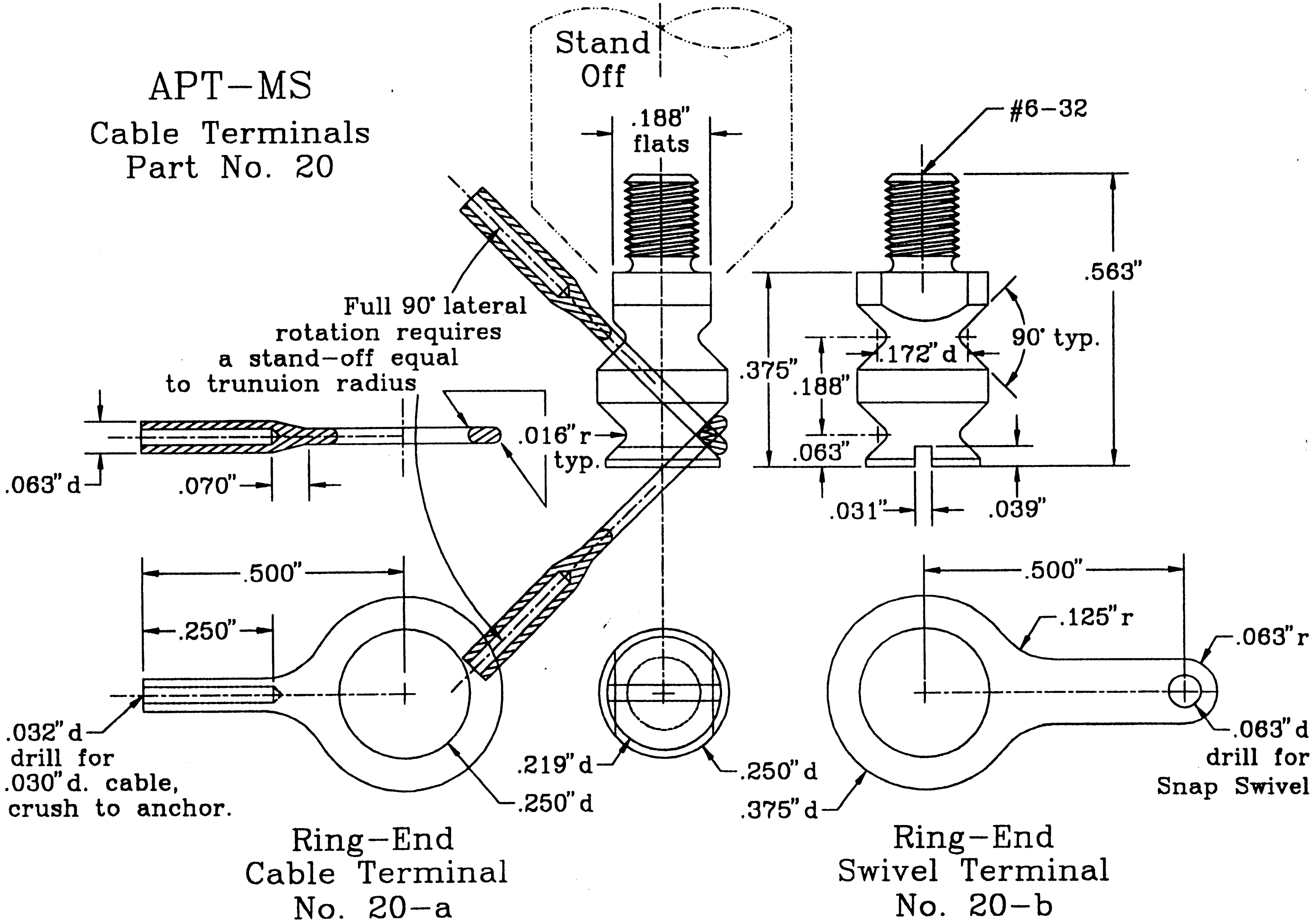

Figure 3-5: Ring-End Cable Terminals

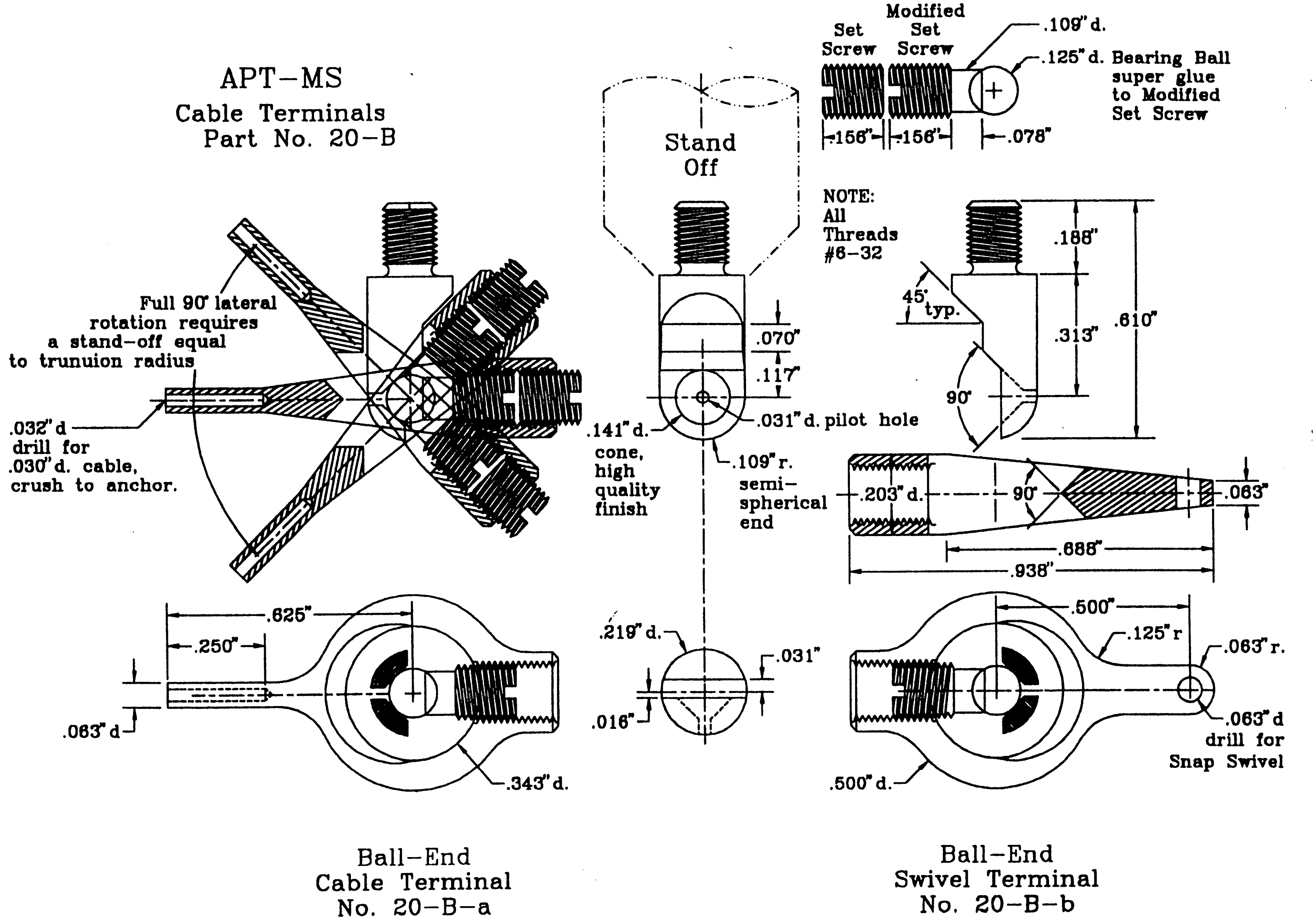

Figure 3-6: Ball-End Cable Terminals

IV

DISCUSSION

4.1 MEASUREMENT ERROR

4.1.1 GENERALITIES. Repeatability is more important than absolute positional accuracy for most of the tasks envisioned for the APT mechanism. This observation does not affect the course of development for the device since most identifiable sources of non-repeatability are the same as those causing measurement error.

Goals for positional accuracy are on the order of .050" at a distance of 24" from the device. "Worst case scenario" vector error was estimated .800" at the same distance for the design in-progress when the error model was introduced. An order of magnitude difference is not encouraging.

Error models that accurately reflect loading and geometry of the operational environment are most apt to be useful. Effort in this direction produced the model described in section 2.3.

Uncertainties regarding model input parameters, rolling resistance factor and weights in particular, leave the above error estimates in some doubt.

Usefulness of the model will come after the hardware is realized. It can then be calibrated and validated or revised as a predictive tool toward further development. Until then it serves to provide an understanding of the influence of relevant parameters.

4.1.2 CABLE CLEARANCE. Zero clearance would eliminate this source of guide error. In the real world zero clearance is unlikely to be achieved without encountering another Achilles heel in the mechanism: friction. After reducing sliding friction as much as possible and clearance as much as is tolerable (with the device proposed in section 3.3 above or some other scheme with the same goals), error can be further diminished by increasing distance between the last contact points of cable to guide (Figure 2-1.b).

4.1.3 BEARING FRICTION. Using the model described in section 2.4 as a guide, it appears that the effect of moment on bearing loads and therefore rolling resistance is likely to be greater than that of the forces causing them, for this mechanism design. Reducing the ratio of a / ρ would be beneficial (where "a" is the moment arm of eccentric radial loads and ρ is the effect of space between bearings in a pair). Smaller bearing pitch diameters and rolling resistance coefficient are bearing parameters that would help. Reducing component weights helps. Increasing tension reduces the effect of component weights. Tension appears to be neutral in effect, with rolling resistance being offset by greater turning force at the cable guide. Increasing the length " l " from bearing axis to cable guide exit is an easy way to improve performance.

4.1.4 EXCLUSION CONES. Unbounded errors result from operating the mechanism at pitch angles ϕ approaching 90 degrees. An operational range must be established within acceptable limits of error. Unacceptable regions can be visualized as cones of revolution about the z-axes. APT device cannot approach $\phi = -- 90$ degrees because of interference with the mechanism base. The current prototype permits operation at values of $\phi = + 90$ degrees so that exclusion cone errors can be explored with the hardware. Operational units would have to be restricted either physically or electronically with appropriate warnings given to the user. In the scenario of Figure 1-1, where motions are always near a vertical plane, more accuracy may be achieved by mounting the device with the x-y plane parallel to the plane of motion rather than horizontally as shown.

4.2 DESIGN ALTERNATIVES

4.2.1 OPTICAL DEVICES usually have better performance and cost more than mechanical counterparts. An inexpensive optical alternative is being pursued in parallel with the mechanical prototype. Optical solutions are beyond the scope of this discussion of a mechanical option.

4.2.2 METAL TAPE. Replacing the guide cable in the mechanical prototype with a metal tape like that found in tape measure rulers has the advantage of stiffness [1]. Since metal tape with a curved cross-section resists bending about both cross-section axes it can deliver a tangential force to guide its cage with negligible tape defection. Better accuracy can be expected on this account. Placing the major bending axis in the x-z plane reduces exclusion cones for any give level of accuracy. Weight / length of tape is greater than that of the cable it replaces therefore errors due to sagging may increase also. These would doubtless be negligible at operational distances of 24 inches.

4.2.3 AIR BEARINGS. Greatly reduced friction obtained would come at a cost. This possibility was considered but rejected without research because of a belief that cost and time to develop it would exceed constraints of an inexpensive, easily portable measurement device.

4.2.4 SHAKE DOWN. Introduction of a vibrator to brake loose stick-slip friction may enhance performance. Frequencies and amplitudes must be compatible with the optical encoders.

4.3 EVALUATION

4.3.1 ACCURACY. Comparison with point to point readings of a digitally controlled milling machine, lathe, or robot to which the cable is attached could evaluate accuracy. Increments small enough that no response is detected in any direction can then be increased until a reading is evident. This may be done in the direction of each spherical coordinate of the mechanism to identify likely sources of error. Permutations and combinations of coordinates can also be tested. Orientation of all device axes should be varied to evaluate the effect of component weights.

4.3.2 REPEATABILITY. Tests repeated over large displacements of each spherical coordinate should be used following the pattern described for accuracy testing.

V

CONCLUSIONS

5.1 SUMMARY

- Error Model is based on mechanism attributes as far as that information is available, including: geometry, component weights, cable tension, device orientation, bearing characteristics, etc.
- Error indicated from worst case scenario is an order of magnitude greater than desired.
- Detailed Design of second prototype, produced in 2 versions depending on spring width, has some parametric flexibility where contact with the cable is concerned.
- Hardware is being fabricated for the narrower of two spring widths considered.

5.2 CONCLUSIONS

- Error model results are inconclusive because of input data uncertainties.
- Calibration of the error model requires hardware test data.
- Evaluation rather than prediction is the best use of the error model.

5.3 RECOMMENDATIONS

- Build and test the second prototype.
- Calibrate the error model.
- Use the error model to identify the greatest error sources.
- Consider design alternatives: replacement of the cable with metal tape in particular.

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