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A Conceptual Design Study for the Secondary Mirror Drive of the Shuttle Infrared Telescope Facility (SIRTF)
Final Report

R.E. Sager
D.W. Cox

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A Conceptual Design Study for the Secondary Mirror Drive of the Shuttle Infrared Telescope Facility (SIRTF) Final Report

R.E. Sager
D.W. Cox
Quantum Design, Inc.
San Diego, California

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Ames Research Center
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National Aeronautics and
Space Administration

Ames Research Center
Moffett Field, California 94035

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I. INTRODUCTION

This is the final report for our conceptual design study for the secondary mirror system to be used in the Shuttle Infrared Telescope Facility (SIRTF). The following eight tasks comprised our statement of work for the study.

1. Perform a literature review to identify designs which are currently in use or are now being developed.

2. In conjunction with NASA personnel, define the major issues which must be addressed by the SIRTF secondary mirror actuator design.

3. Develop alternative ideas for the secondary mirror actuator system, with particular attention to the requirement for operating the system in a cryogenic environment.

4. Perform a tradeoff analysis between different possible actuator systems in terms of the major SIRTF issues identified in Task 2 above, again with consideration for the cryogenic requirements.

5. Prepare and submit a midterm progress report.

6. Develop a specific conceptual design approach for the secondary mirror actuator system.

7. Outline a plan for developing the recommended actuators, and identify those issues which will be of critical importance in the developmental effort.

8. Prepare and submit a final project report.

The results from the first four tasks were discussed in detail in our midterm progress report, and while we will not repeat that entire discussion here, we have drawn extensively from some parts of the earlier report. In particular, Sections II, III, and V of the midterm report summarize essential parts of our work under this contract, and much of that information is central to our discussion of the conceptual design we present below. Section II of this report, for example, sets forth the issues defined under Task 2 which will be crucial to the success of SIRTF, and this discussion has been taken almost verbatim from our midterm report. While most of Sections III and IV in this report also appeared in the midterm report, the numerical estimates have been updated to reflect our most recent calculations.

Much of the information included from the midterm report forms the foundation for the conceptual design described in this report. As we identified the major issues for the SIRTF secondary mirror system, and later examined the fundamental problem of driving the mirror, a consistent theme became apparent leading to our conceptual design for the secondary mirror system. Hence, the design has evolved as a natural consequence of addressing the fundamental problems inherent in the SIRTF specifications, and seems to provide a sound approach to the most important problems which must be solved.

In presenting this type of comprehensive summary, we've also tried to construct a basis for evaluating other designs which might be suitable for the SIRTF secondary mirror system. It is our hope that such a summary can be used for preliminary evaluations of the various designs to identify those which merit further development. Our own conceptual design developed under Task 6 represents the culmination of our efforts under this study, and it must, of course, also meet the criteria we've set forth. In Section VI we try to evaluate the design against those criteria.

We have also tried to identify those issues in our design which present the greatest risk, and these are discussed in Section IX. These points played a central role during Task 7 in which we outlined a program for developing the proposed system. The program identifies milestones associated with each of the critical facets of our design, and presents an approximate timetable for resolving the most important questions and actually fabricating an operational prototype device.

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II. OPERATIONAL ISSUES FOR EVALUATING ACTUATOR DESIGNS

Continuing discussions with NASA personnel indicate that some details of the performance requirements for SIRTF have not yet been firmly established. To avoid confusion in our numerical estimates, and for completeness, we first summarize the numerical specifications which we have been using.[1]

1. Secondary mirror diameter: 10.7 cm.

2. Chopping motion specifications:

   a. Mirror drive must provide two-axis chopping.
   b. Chopping amplitude: Variable up to 28 arcmin.
   c. Chopping frequency: Variable up to 20 Hz.
   d. Chopping waveform: Square, triangular, sawtooth.
   e. Duty cycle: 90%.
   f. Chopping axis: Continuously variable.

3. Chopping drive should also provide fine guidance control

4. Absolute pointing stability: 0.1 arcsec.

5. Secondary mirror operating temperature: $T < 7K$.

6. Maximum power dissipation at secondary: 200 mwatts.
   Desirable power dissipation: 50 mwatts.

7. Thermal drift in the secondary mirror: 10 μK/sec.

8. Mirror drive must be reactionless.

The above constraints place stringent requirements on the mirror drive system. In particular, the simultaneous requirements for a 20 Hz chopping rate, a 90% duty cycle, and a 28 arcmin total throw with a 0.1 arcsec pointing stability over the duty cycle will be extremely difficult to achieve, even in the absence of other constraints.

To illustrate more clearly the difficulty of meeting these specifications, the servo-control system must swing the mirror through an angle of 28 arcmin, then position it to a precision of about .006 percent (referenced to the total throw) in a total time of 2.5 msecs. This is not a trivial problem. Our Tecnical Monitor at NASA has recently indicated that the duty cycle specification may be relaxed, which should substantially ease the requirements on the servo-control system, but meeting all of the specifications will still require careful design and engineering in the secondary mirror system.

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In addition to the above numerical specifications, however, there are several other issues we've identified which will severely affect its performance if not properly addressed. Since the expertise of Quantum Design personnel lies primarily in the areas of cryogenic instrumentation and low temperature physics, rather than optical systems or infrared astronomy, inputs from our Technical Monitors at NASA were particularly useful in helping define the issues discussed below.

We initially approached the problem by considering the fundamental physical problems inherent in meeting the secondary mirror specifications for mechanical chopping, pointing stability, thermal dissipation, and the cryogenic environment in which the system must function. Along with the additional requirements for simplicity and reliability in the system, these considerations are issues discussed below.

1. Reliability. One of the greatest concerns in any spaceborne system is reliability. Regardless of how well the system fulfills its other operational specifications, it will be a failure if the secondary mirror articulation system cannot function with nearly one hundred percent reliability. In cryogenic applications a primary key to reliability is simplicity of design minimizing the number of moving parts, the number of interfaces between different materials, and avoiding close mechanical tolerances in fabrication and assembly.

2. Thermal dissipation in the mirror. This is of vital importance for SIRTF since any thermal dissipation in the mirror itself will be directly visible to the IR detectors. Furthermore, the resulting thermal gradients in the mirror will combine with the chopping motion to produce a spurious modulation at the detectors. The problem will be exacerbated by the low heat capacity of materials at the required operating temperature of 4 to 7 Kelvin. (The heat capacity of beryllium at 7 Kelvin, for example, is about .0002 joules/gm-K, about a factor of 10^4 less than at room temperature.[2]) In spite of the high thermal conductivity of beryllium at these temperatures[3] (of order 13 watts/cm-K), even small amounts of eddy current dissipation in the mirror will produce unacceptable temperature fluctuations.

3. Distortions of the mirror. Another critical issue will be the magnitude of distortions in the mirror produced by the driving forces. Ideally, the driving forces will be applied to the mirror during the transition between its dwell points, rather than during the dwell period. If the inherent Q of oscillations in the material of the mirror is not too large, the resulting distortions should relax sufficiently quickly to avoid serious interference with the observations. This question will be closely linked with the problem of making the mirror as light weight as possible while still providing the required rigidity. Lightening the mirror also has the desirable feature of decreasing both its

moment of inertia and the forces which will be required to drive it.

4. Total thermal dissipation at the secondary mirror mount. While gaseous helium may be circulated to the secondary mirror site to provide cooling, care must still be taken to minimize the total dissipation in the secondary mirror drive system. The SIRTF design specification is 200 milliwatts maximum dissipation at the secondary mirror mount with a design goal of less than 50 mwatts. With the additional requirements for a ninety percent duty cycle at a 20 Hz chop, and a total throw of 28 arcmin, this low limit for power dissipation places stringent constraints on the mirror actuators and control system. An excellent approach to reducing the power dissipated at the secondary mirror mount will be to use superconducting components provided they can be implemented without compromising other aspects of the design.

5. Thermal management. Another aspect of the thermal dissipation problem is the way in which dissipated heat is conducted away from the mirror. For example, the thermal management of a design must address such points as the amount of heat dissipated in elements attached directly to the mirror, the thermal connections between the mirror and the heat sink, and thermal gradients in the mirror which might result from heat being extracted from the system through the mirror itself. Hence, the ideal design will minimize dissipation in any component connected directly to the mirror, and will intercept any dissipated heat before it reaches the reflecting surface.

6. Overall compatibility with operation in cryogenic environments. Since the temperature of the secondary mirror must be maintained below 7 Kelvin, all components of the system must be able to function at this very low temperature. Furthermore, the system must be able to survive repeated thermal cycling from room temperature, and it must be extremely reliable once it has been cooled to operating temperature. Cryogenic aspects of the design include such considerations as differential thermal contraction between different materials, heat sinking of dissipative components near the secondary mirror, and the expected lifetime of flexural joints when operating at cryogenic temperatures.

7. Inclusion of a reaction mass. While the requirement for a reactionless system is included in the above list of numerical specifications for SIRTF, we also include it here as a general consideration for evaluating different designs since the design concepts must be compatible with including a reaction mass. Consequently, to be acceptable the system must provide a reasonable implementation for the reaction mass.

These are the general issues which we believe will be important for the SIRTF secondary mirror design. In the next section (taken primarily from Section III of our midterm report) we discuss some of the more specific engineering aspects of the system which will be crucial in developing a device which can meet the stringent specifications for chopping capability and pointing stability, yet fulfill the requirements outlined above.

III. FUNDAMENTAL DESIGN ISSUES FOR THE MIRROR SYSTEM

The numerical specifications outlined in Section II place stringent requirements on the SIRTF secondary mirror system. When the additional, more general considerations of thermal dissipation, reliability, and cryogenic requirements are added, the problem becomes extremely difficult from an engineering standpoint. The arguments in this section resulted from trying to make the overall secondary mirror problem more tractable.

In general terms, the torque and energy required to achieve the desired motion of the secondary mirror will be determined by the angular acceleration, maximum angular velocity, and the total moment of inertia of the moving elements. Since the angular acceleration and the maximum angular velocity are essentially defined by the specifications for total throw and chopping rate, the only remaining property of the system we can easily manipulate is the moment of inertia of the system. To optimize the design then, we must address the following points.

1. Reduce the mass of the mirror to the maximum extent possible consistent with the requirements for rigidity and strength. In particular, eliminating excess material near the edges of the mirror will yield substantial reductions in the total moment of inertia of the system and the corresponding torques which will be required to drive it. This will have the added benefit of increasing the frequency of self resonances in the mirror structure which will be an important issue in designing the servo-control loop for the system.

2. Reduce the characteristic linear dimensions of the mirror, subject to the requirement that the reflecting surface have the required diameter. More specifically, we need to minimize the moment of inertia of any levers or other structures associated with the mirror actuators. This includes not only reducing their mass, but minimizing their distance from the axis of rotation.

3. Identify those actuators which can produce the necessary force with an absolute minimum of moving mass. This will be particularly advantageous since an actuator having little mass in its moving elements will have to generate less force to produce the required motion, and the total energy requirement will be reduced.

4. Optimize the radius on which the actuator operates. For any system in which the actuator forces are not applied directly to the mirror, there will be some optimum distance from the axis of rotation at which the actuator should be located. This optimum represents a tradeoff in which the increasing moment of inertia and decreasing rigidity of the actuator lever arm are balanced

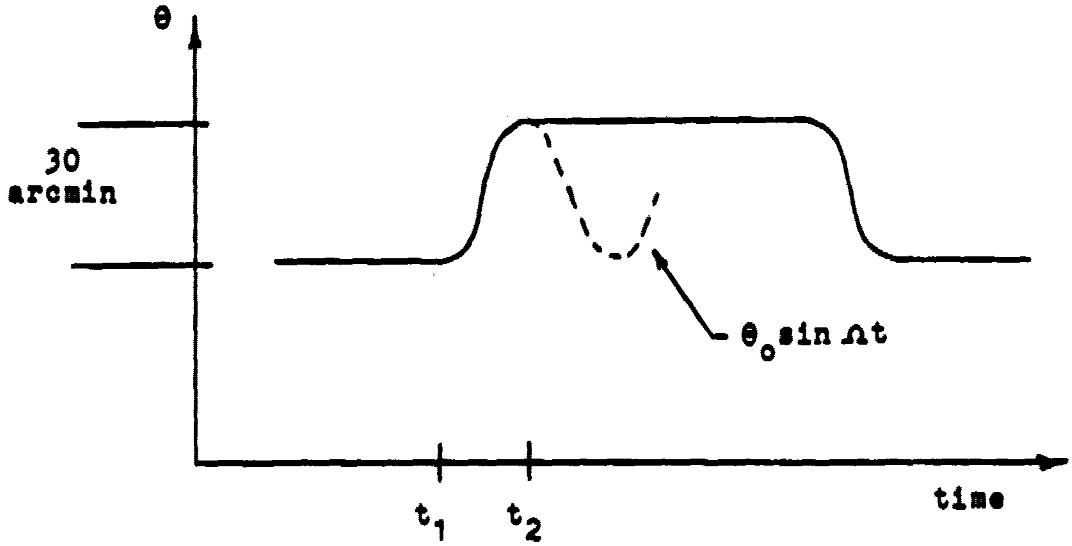
against the mechanical advantage obtained from a longer lever arm. Although one can sidestep this problem by allowing the actuator to act directly on the back side of the mirror, we believe that this technique will not be suitable for two reasons.

First, forces applied directly to the mirror, especially near its outer edges, will require increased rigidity and mass in the mirror structure to keep distortions within acceptable limits. Secondly, and more importantly, with actuators acting directly on the back of the mirror, it will be extremely difficult to completely eliminate all thermal dissipation in the mirror structure, especially if any type of electromagnetic actuators are used, as proposed in several designs [4-6].

To further illustrate the concerns outlined in the four points above, and make the discussion more quantitative, consider a mirror which moves between its two dwell points in a sinusoidal motion to produce a chopping waveform with a 90% duty cycle and 30 arcmin throw at 20 Hz as shown in Figure 1. From these values we find a maximum angular velocity of about 5.5 radians/sec. Our initial calculations for the moment of inertia of the mirror indicated that a value of about 1.0×10^{-4} kg-m^2 for the total moment of the moving elements might be possible with very careful design. Using this original estimate, we find a maximum kinetic energy for the mirror of about 1.5 millijoules. Since this amount of energy must be supplied twice during each cycle, the total energy delivered to the mirror at a 20 Hz chopping rate will be 60 milliwatts. Furthermore, if the energy is not dissipated in some damping mechanism at the mirror, the actuator drivers will have to reabsorb that 60 mWatts by generating reverse forces to stop the mirror.

More recent inertia calculations based on a well developed conceptual design, however, show that the value assumed above is probably too optimistic. As we discuss in some detail in Section VII, a value of 1.5×10^{-4} kg-m^2 should be readily achievable, with a value possibly as low as 1.3×10^{-4} kg-m^2. If we take the more pessimistic view, our calculation of the previous section becomes 90 mwatts of power being delivered to and reabsorbed from the mirror at the maximum chopping amplitude and frequency. If we assume lossless actuators, the energy can then be dissipated in the actuator electronics which can probably be located remotely from the secondary mirror mount. This will almost certainly be required to meet the SIRTF power dissipation limit at the secondary mount.

These arguments clearly demonstrate the importance of reducing the total moment which must be driven by the actuators. However, even these estimates are very optimistic because of our initial assumption of a quasi-sinusoidal transition between the dwell positions. To achieve the required amplitude and settling time, the servo-control system will have to deliver substantially more energy to the mirror than indicated in our simple calculation.



For sinusoidal motion:

$\theta = \theta_o \sin \Omega t$

θ_o = 15 arcmin (.00436 rad)

At a 20 Hz chopping rate and 90% duty cycle:

$t_1 - t_2 = 2.5$ msec

$\Omega = 1.26 \times 10^3$ rad/sec

Max angular velocity = 5.5 rad/sec

Max angular acceleration = 6.9×10^3 rad/sec^2

Figure 1. Rough estimate of required mirror motion for SIRTF secondary mirror.

Mike Dix[7] of NASA has made some preliminary calculations on the servo-control system requirements which we expect will be required to meet SIRTF pointing specifications, and which could generate the required chopping. For his model Dix assumed a total moment of inertia of 1.6×10^{-4} kg-m^2, and a push-pull actuator pair operating on 4 cm lever arms, and found a settling time of about 3 to 3.5 msecs (to a pointing accuracy of 0.1 arcsec) with a maximum force of about 45 newtons per actuator. To make a comparison to our conceptual design, we should be able to achieve a 10 to 20 percent reduction in moment of inertia from Dix's value, and we expect to be using a similar pair of actuators on a 20 to 50 percent longer lever arm. Consequently, these calculations indicate that we might be able to approach the settling time of 2.5 msec required to produce a 90% duty cycle at 20 Hz.

While the above estimates are reasonably consistent with SIRTF specifications, they effectively demonstrate the necessity for keeping the moment of inertia of the entire system at an absolute minimum. While optimizing the moment of inertia will require little sacrifice other than careful mechanical design, we believe that the probability of meeting the SIRTF specifications can be significantly improved at the beginning of the development effort by giving proper attention to this fundamental issue. In particular, any success in decreasing the total moment of the moving components will return the following benefits:

1. A decrease in the energy which must be supplied to the mirror during each transition between dwell points.

2. A decrease in the forces which must be applied to the mirror to produce the required motion.

3. A decrease in the voltages or currents which must be supplied to drive the actuator. This will also reduce stray magnetic fields which will decrease eddy current dissipation in the mirror or support structures which is particularly important for magnetic actuators.

4. Less dissipation in the actuators and electronics, which should decrease as the square of the applied current or voltage.

5. And finally, less inertial reaction which must be absorbed by a reaction mass or by the secondary mount itself.

A simple statement of our general approach, then, is to give particular attention to minimizing the total moment of the mirror, select an actuator which has the smallest possible mass in its moving elements, provide some mechanical advantage for the actuator with the goal of reducing the maximum forces which are required, and minimize or eliminate thermal dissipation in the mirror structure itself. In addition, we would hope to select an actuator which could be readily adapted to use superconducting elements, especially for those components which are to become part of the mirror assembly. The conceptual design we present below is consistent with these goals.

Finally, we would like to make the following comment with respect to the above discussion. The issue of minimizing the moment of inertia of the system may seem obvious or trivial, and perhaps undeserving of the attention we have given it. However, a review of some of the proposed actuator systems has led us to conclude that, in more than one instance, this simple and fundamental consideration has been sericusly compromised or even completely lost in the details of a particular mechanical design.

IV. RECOMMENDATIONS ON SPECIFIC ACTUATORS

At the beginning of this study our primary goal was to simply examine specific actuators for the secondary mirror without particular attention to the more general system design. However, it quickly became clear that the suitability of any given actuator would depend closely on the overall mirror system design. Consequently, our evaluation came to include not only different possible actuators, but also the particular mechanical configuration which was proposed for their implementation. This has been advantageous as we have been forced to think more in terms of the overall system and the fundamental issues of the mirror drive problem, rather than considering only one limited aspect of the problem. We have generally taken the broader view throughout this study, but in evaluating various designs and developing our own concepts we have also reached some conclusions about the actuators themselves.

After several weeks of computing moments of inertia, estimating the forces available from different actuators, and worrying about thermal dissipation and thermal anchoring of different components in the mirror system, we can make the following observations regarding the nature of an ideal actuator.

1. First, the actuator must contribute as little as possible to the moment of inertia of the system. Since it will be operating at some distance from the axis of rotation, any substantial moving mass will dramatically increase the moment of inertia of the system, require larger forces from the actuator, and ultimately lead to higher dissipation in the system.

2. Any actuators which produce stray magnetic fields must be removed from the immediate vicinity of the mirror to avoid eddy current dissipation in the mirror itself. One of the greatest concerns for SIRTF will be thermal dissipation and thermal gradients in or near the reflecting surface of the secondary mirror.

3. Our third concern is a further ramification of the thermal aspects of the design. Specifically, the actuator must produce the smallest possible dissipation in its moving elements. Since the moving components of the device will be, in some manner, directly mechanically connected to the mirror, heat dissipated in the moving parts of the driver will have a direct thermal path to the mirror. While the mirror system design must provide a suitable way of extracting dissipated heat before it reaches the reflecting surface, any reduction in the amount of heat which must be removed will ease the design problem.

4. Another concern is that the actuator must work well at cryogenic temperatures. Piezoelectric devices, for example,

produce only about one third as much displacement at 10 Kelvin as they do at room temperature. Also the actuator must produce very little dissipation at low temperatures. This is particularly important because of the very small heat capacities of materials at the operating temperature of the mirror.

5. The actuator of choice should be readily compatible with the use of superconducting components which can eliminate ohmic heating, especially if large currents are required. Also, if the actuators produce fluctuating magnetic fields or large field gradients, the nearby components of the actuators must be electrical insulators so that eddy current heating is also eliminated.

6. Sixth, to help reduce dissipation in the drive electronics, the actuator should produce an absolute minimum of dissipation during the dwell periods of the chopping cycle. Fulfillment of this condition will minimize the time averaged dissipation in the actuator and keep the total dissipation in the system low.

7. Due to the problems of differential thermal contraction, the actuator should not require that extremely close tolerances be maintained during fabrication and assembly in an attempt to achieve precision fits at cryogenic temperatures. Such designs can be frustrated by nonuniform thermal contraction of construction materials as the system is cooled to its operating temperature.

8. Finally, the actuator itself must be basically immune from mechanical resonances which may interfere with the servo-control system.

The various actuators we've considered include piezoelectric devices, linear motors, voice coil drives, the Lockheed style motor which is a somewhat different implementation of the linear motor concept[5], and mechanical springs operating against an electromagnet as in the GIRL design[6]. We make the following observations about these different types of actuators.

While linear piezoelectric actuators fulfill nearly all of the criteria, they are eliminated simply by their failure to produce sufficient linear displacement[8]. The upper limit on linear displacement using piezoelectric stacks is about .001 cm at room temperature which is at least a factor of 3 to 10 less than required, and the situation is substantially worse at cryogenic temperatures where the performance of the piezoelectric materials drops by a factor three or more. We also considered using a piezoelectric bender bimorph, but these devices do not provide sufficient force to actuate a single piece mirror. While the bimorphs might be suitable for a segmented mirror design, other considerations appear to overwhelm any possible advantages of this approach.

While the Lockheed design[5] should produce substantial forces, the motor itself fails the above tests on several counts. It will dramatically increase the moment of inertia of the system, and there may be substantial dissipation in the moving armature, both from electrical dissipation in the coil and eddy current dissipation in the armature core. In the Lockheed design, we also expect a substantial problem in achieving the required tolerances at cryogenic temperatures.

We have not proposed a specific implementation for a more conventional linear motor since it will suffer from many of the same maladies as the Lockheed design. Specifically, the magnetic armature material will dramatically increase the moment of inertia of the moving elements. While there would typically be no windings on the armature as in the Lockheed design, there will still be a problem of eddy current losses in the armature, and stray fields from the electromagnets if the actuator is near the mirror. Using a more conventional linear motor in a cryogenic environment may also present difficulties in maintaining the required clearances.

The GIRL design[6] offers an alternative approach using electromagnetic actuators in a rather different configuration. While the GIRL chopper will probably produce less eddy current heating than a linear motor, the period of maximum dissipation in the system is during the dwell periods. This is particularly unfavorable in terms of the duty cycle for the current which must be supplied to the actuators. Another potential difficulty with this design is that substantial forces are exerted on the mirror during its dwell period which will require a heavier mirror to prevent unacceptable distortions in the reflecting surface during the observation part of the chopping cycle.

However, as pointed out by Dr. Walter Brooks of NASA, one significant advantage of the general design is that in the event of a failure, the mechanical springs will insure that the mirror comes to rest at the central position rather than at one extremum of its throw. This would allow at least some types of observations even following a catastrophic failure of the secondary mirror drive system.

Finally we address the voice coil design which seems to fulfill our requirements better than any of the other actuators we've considered. Since the only moving component of the actuator is the coil itself, the additional moment of inertia can be small, especially if special attention is given to designing the support structure for the coil. Similarly, eddy current and ohmic losses in the moving elements can be nearly eliminated if the coil carrier is made of an electrical insulator and a superconducting coil is used.

Another natural feature of voice coil actuators is that their performance should significantly improve at cryogenic temperatures. Both the remnant field and coercivity of samarium cobalt increase at cryogenic temperatures producing improved performance in the permanent magnets[9], and the actuators are ideally suited for use with superconducting coils. Also, there is no particular requirement for holding extremely close tolerances in these devices.

Finally, the electronic dissipation for voice coils devices should also be acceptable since the forces required to hold the mirror at its dwell points will be reasonably small, assuming that there are no preloading forces (as in the GIRL design). And because of its mechanical simplicity, there are no obvious sources of mechanical resonances inherent in the actuator itself. To conclude, of all the actuators we've considered, the voice coil device seems to most naturally and completely fulfill the requirements outlined above, and the operating temperature specifications for the mirror are ideal for the use of superconducting coils. We discuss this conclusion in further detail in Section VIII.

V. THE CONCEPTUAL DESIGN

As outlined in our original proposal, we have tried to develop a design for the secondary mirror drive which solves many of the system's thermal and mechanical problems in a natural way, rather than trying to address each particular concern by simply introducing another modification to a preexisting device or design. Nonetheless, the general features of our conceptual design were largely determined by four particular considerations. Specifically, the most difficult problem was to provide the two-axis chopping capability while keeping the system as mechanically simple as possible, and yet minimize the total moment of inertia of the moving elements and provide good thermal isolation between the actuators and the mirror itself. Based on our findings in the first half of this study, we assumed that the design would use some type of voice coil actuator.

In this section we will discuss our approach to the secondary mirror design qualitatively, then present some preliminary numerical estimates for the system in Section VII. One note of importance here is that the SIRTF specifications given to us at the beginning of this work specified a ninety percent duty cycle for the mirror, and we have tried to address that requirement throughout this study effort. However, recent discussions with NASA personnel have indicated that the duty cycle specification may be relaxed to 70 percent. This will represent a significant relaxation of the requirements on the servo-control system, and we will comment further on this point later in this report.

1. The design geometry. The physical geometry of the mirror was driven primarily by the need to minimize its moment of inertia while providing enough mechanical advantage for the actuators so that reasonable forces could be used. The other major consideration was the requirement for a two-axis chopping capability. Several previous articulated mirror systems have used actuators in a push-pull configuration with the individual actuators located on opposite edges of the mirror[10-12]. This typically requires that the actuators operate directly on the back of the mirror, or that each actuator operate on its own lever arm. Lou Salerno and Mike Dix have done some preliminary calculations and experiments based on the latter configuration[1,7].

In our design, which is a modification of this scheme, the four actuators are clustered behind the secondary mirror and operate on a single drive stem which extends backward from the center of the mirror. The principal features and general geometry of the design are shown in Figure 2.

In this configuration the mirror is supported on a single flexural pivot which allows small rotations of the mirror about any axis perpendicular to the longitudinal axis of the telescope. (The flexural pivot design is discussed further below.). The two perpendicular chopping axes are each driven by a pair of voice coil actuators operating in a push-pull mode with the voice coils mounted in a cluster on the end of the mirror drive stem as shown in the cutaway view of Figure 3. In principle, the two orthogonal motions can then be combined to provide an arbitrary chopping axis. (This also will be discussed in more detail below.) Finally, position (and possibly velocity) transducers mounted at the positions shown in Figure 2 provide inputs to the servo-control positioning system for the mirror.

In the interest of clarity throughout the rest of this report, we define the phrase "mirror assembly" to include the mirror itself, the drive stem, and the actuator coil cluster, but not including the wedge-shaped permanent magnets. Also, we will use the term "mirror" to refer only to that portion of the secondary mirror structure which extends forward from the flex pivot outside of the mirror housing and faces the telescope's primary mirror.

To summarize the advantages of this design, there are no external forces applied to front part of the mirror which eliminates the problem of the actuators introducing distortions into the reflecting surface. The design also provides the maximum possible separation between the actuators and the mirror, can completely shield the mirror from any sources of thermal dissipation, and provides a natural heat sink at the flexural pivot which lies between the mirror and literally all of the dissipative elements in the system.

Another feature of the design is its ability to trade off the mechanical advantage of a longer drive stem against its increasing moment of inertia. Our geometry may be particularly helpful in this respect since there is no critical limit on the length of the secondary mirror housing, but its diameter must be kept less than about 10 to 15 cm. While there are very real limits on the length of the drive stem, dictated by its required rigidity and the need to optimize the moment of inertia of stem and coil assembly, the design provides a convenient geometry for optimizing the tradeoff without exceeding the allowable dimensions for the mirror housing. Furthermore, by requiring only a single actuator lever to provide both chopping axes, the system has an inherent simplicity which will help eliminate high frequency mechanical resonances in the system.

The reaction mass for the mirror is provided at the back end of the secondary mirror mount by a counterrotating cylindrical mass which might be driven with the mirror actuators. As shown in Figure 2, the reaction mass is supported by its own flexural pivot and coupled to the actuators through a third flexural joint. While the configuration in Figure 2 shows one possible implementation for the reaction mass, we have also considered two other designs, which we discuss in more detail later in this report. Since we

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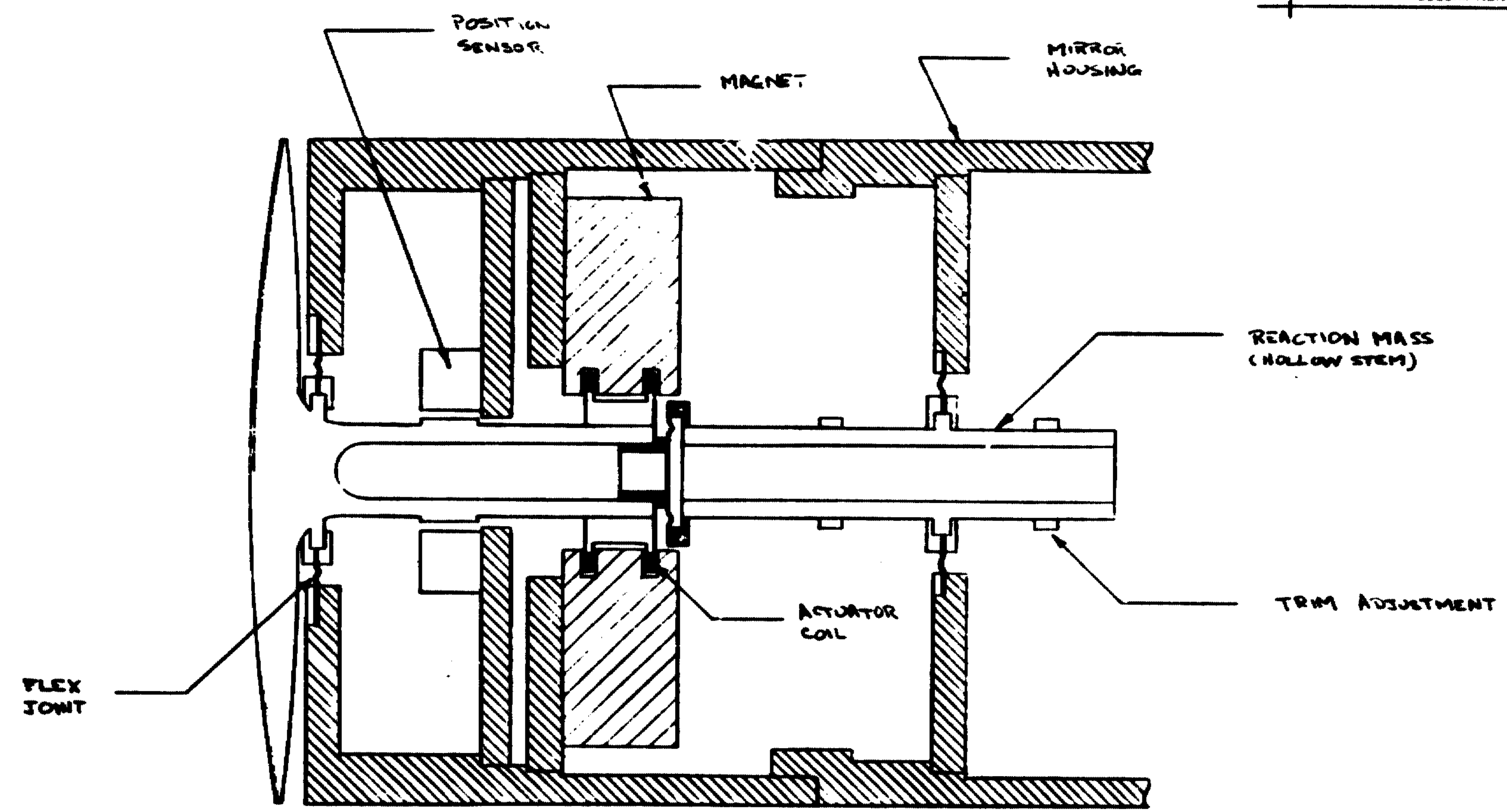

FIGURE 2

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SECONDARY MIRROR
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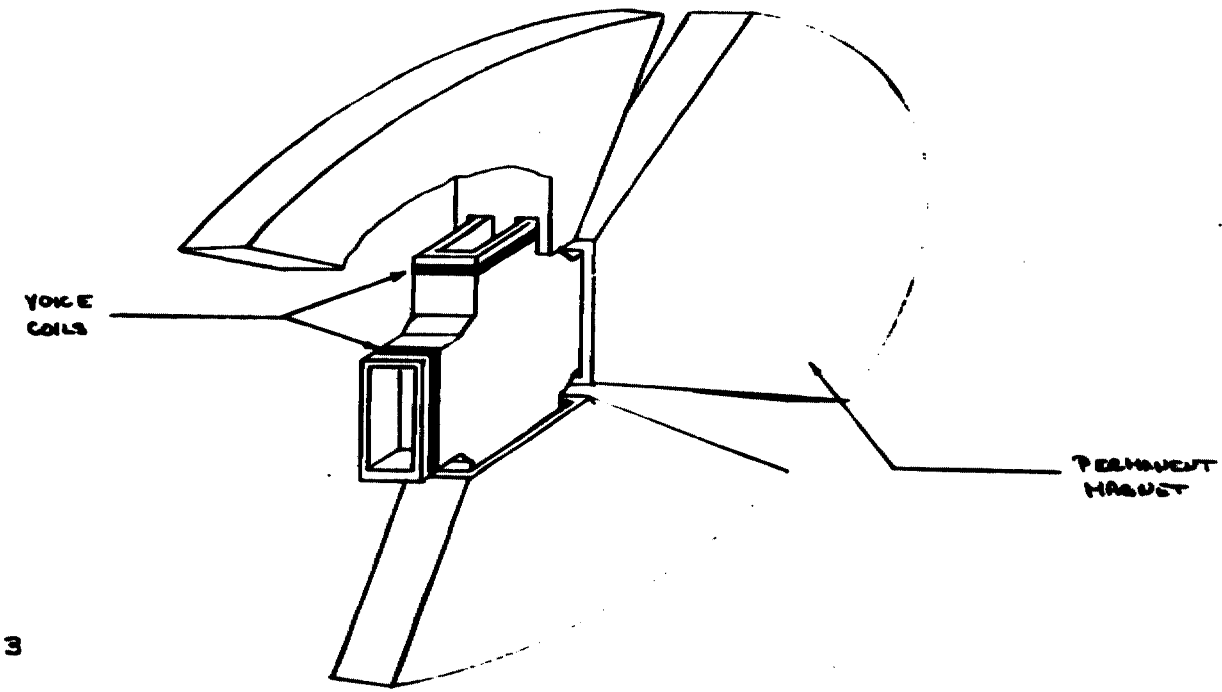

FIGURE 3

UNLESS OTHERWISE SPECIFIED
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have not yet reached a final conclusion on this point, the design depicted in Figure 2 must be considered as strictly a preliminary concept. We will consider the problem of the reaction mass at greater length in Section VII where we briefly discuss two other alternatives.

2. The thermal design. We believe that the thermal characteristics of this geometry represent one of the design's greatest assets, especially since the elimination of thermal dissipation and thermal fluctuations in the mirror will be crucial for SIRTF. The natural geometry of the design places all of the dissipative elements in the drive system at the end of the drive stem where they are removed from the immediate vicinity of the mirror. Furthermore, the mirror can be shielded from stray magnetic fields by placing a high permeability shield on the mirror housing directly behind the mirror (probably inside of the housing).

The thermal design is further strengthened by using the flexural pivot as the primary heat sink for the entire mirror assembly. Most importantly, the heat sink is naturally located between the mirror and the dissipative drive elements so that heat generated in the actuators and drive stem is removed before reaching the mirror without requiring any thermal conduction through the mirror. Furthermore, if the flexural joint is designed to have a low thermal impedance, the mirror will be in excellent thermal contact with the rather massive mirror housing which can serve as the local heat sink at the secondary mount. We present some brief numerical estimates for the thermal aspects of the design in Section VII.

The other significant thermal feature of the system concerns the thermal contact to the permanent magnets for the voice coil actuators. The voice coils themselves will be part of the mirror assembly, but since they will be constructed of superconducting wire, there should be no heat generated in the coils themselves. Hence, the primary source of thermal dissipation will be hysteresis and eddy current losses in the magnets. A major advantage of the moving coil/permanent magnet actuators is that the magnets, which produce most of the dissipation in the system, can be mounted in direct metallic contact with the mirror housing to provide an extremely low thermal impedance path between the magnets and the local heat sink. Also, since there will be no direct mechanical contact between the magnets and mirror stem or voice coils, there should be no significant transfer of heat from the magnets into the mirror assembly.

3. The Mirror. The optimized design for the mirror will be an important aspect of the prototype development program, since the entire control system and actuator design will revolve around the total moment of inertia of the system. If the SIRTF duty cycle specification is relaxed, there will be less need to achieve the ultimate reduction in the moment of inertia for the mirror.

Keeping its moment as small as possible, however, will still be a desirable goal since a small moment for the mirror will help ease the requirements on the actuators and servo-control system.

We have not yet developed the details of the mirror construction, but our preliminary calculations indicate that it may not be desirable to fabricate the mirror stem from beryllium. Because of its very low heat capacity and good thermal conductivity, any heat dissipated at the end of the stem near the actuators will probably produce unacceptable thermal fluctuations at the center of the mirror. Since the heat capacity of all solid materials decreases very rapidly at temperatures below about 15 Kelvin, the probable solution is to construct the drive stem from an electrically insulating material which has a low thermal conductivity. This accomplishes two things.

First, the insulating quality of the material will eliminate any eddy current dissipation in the drive stem, and since the voice coils will use superconducting wire, there should be almost no thermal dissipation in the drive stem or coil cluster assembly. Secondly, a low thermal conductivity in the drive stem material will result in a very diffusive heat flow so that the only effect at the flexural joint will be a slight warming due to the thermal impedance of the flexural joint. We discuss this point in more detail in Sections VII and VIII where we present some simple numerical calculations to further illustrate the problem.

The interface between the mirror and the drive stem is shown only schematically in Figure 2. The problem of differential thermal contraction will have to be addressed in the final design for mating the drive stem and mirror. However, we understand from NASA personnel that there will be a circular "dead zone" in the center of the secondary mirror which is not part of the optical system. Hence, it may be possible to fabricate the mirror with a central hole which would accommodate a suitably designed drive stem. This is similar to the arrangement used by Lou Salerno and Mike Dix of NASA in their design for a preliminary experimental model of the mirror[1,7].

4. The flex washer. An innovative feature of our present design is the use of a "flex washer" to provide the universal flexural joint upon which the mirror can rotate. The primary advantages of the flex washer are the simplicity with which it can be implemented and the natural way in which it provides the thermal heat sink for the mirror. Its primary disadvantage is that it represents an untested innovation which will require some developmental effort to insure that it will be suitable for use in SIRTF. We address this point more fully in Section X where we outline the tasks which will comprise a development program to produce a complete operating prototype of the articulated secondary mirror system.

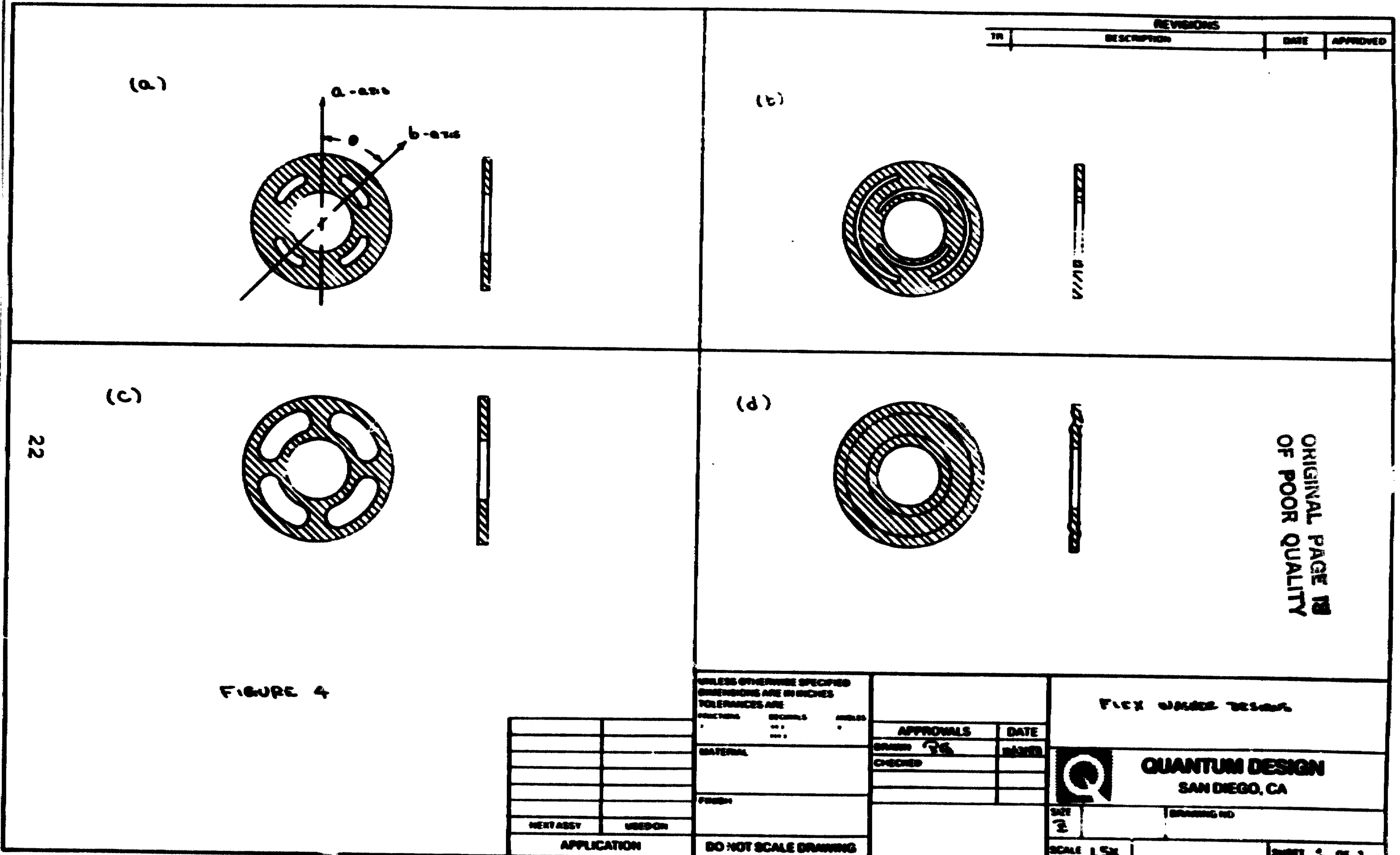
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FIGURE 4

One feature that we believe favors the flex washer design is the ability to design the dimensions of the flex joint to be particularly suited to the intended application. For example, Figure 4 shows four variations of the design, all of which fulfill the same function, but each having rather different mechanical properties. The geometry shown in Figure 4(a), for example, should provide a fairly rigid mounting, reasonable longitudinal support for the mirror, and fairly large restoring forces even for small rotations. The washer design in Figure 4(c), on the other hand, will provide a rather "soft" mounting for the mirror. However, both of these designs will probably introduce an asymmetry into the system since the restoring force experienced during rotations about the "a" axis will be somewhat different from that about the "b" axis (where θ = 45 degrees). This could possibly cause problems in the servo-control system since the two chopping axes could have different response functions depending on the chopping axis.

The design shown in Figure 4(b), suggested by Dr. Fred Witteborn of NASA, illustrates an alternative approach which will help decouple the two axes of rotation and eliminate the "stretching" effects which would arise from rotations about the "a" axis in the first geometry. However, this design, as well as the design shown in 4(c), may not provide sufficient longitudinal support for the mirror.

Yet another alternative is shown in figure 4(d) in which a solid washer is used, but with a slight corrugation between its outer diameter and inner diameter. This geometry will provide complete longitudinal symmetry with respect to the choice of chopping axis. In addition, by selecting the proper combination of thickness, inner and outer radii, and depth of corrugation, the washer can be designed to have the correct mechanical properties.

The one potential problem we can envision in the flex washer design is the possibility of longitudinal oscillations along the axis of the mirror, since that will clearly be one of the normal modes for the washer. However, the deformation produced in the washer during the normal chopping motion of the mirror should also be a normal mode of the washer. Since the two modes of oscillation appear to be orthogonal, we would expect energy to be coupled into the longitudinal mode only through nonlinear effects, especially since the magnitude of angular displacement will be very small. Furthermore, the chopping motion of the mirror will produce a node along the axis of rotation (along axis "a" in Figure 4a, for example), while a longitudinal oscillation requires a nonzero displacement everywhere except at the outer radius.

While these observations do not provide any quantitative insight into the longitudinal oscillation problem, the nature of this design makes it ideal for analysis using numerical techniques in a computerized mechanical design package. In view of the above discussion, we feel that it will be highly desirable to perform this type of calculation in the initial design stages to help characterise and evaluate a variety of different flex washer designs before proceeding with their fabrication and testing.

Our purpose in presenting the foregoing discussion is not to try to analyse all of the potential problems in the flex washer design, but rather to demonstrate an appreciation for some of the more straightforward considerations which must be addressed in developing a design which we believe has considerable merit. However, since it is an important component of our current conceptual design, it must be properly evaluated in the first stages of any development program to allow an early decision regarding its suitability for the SIRTF application. Our schedule for a developmental program, presented in Section X, addresses this point.

5. The actuators. As we discussed at length in Section IV, we believe that the most appropriate actuators for SIRTF will be voice coil actuators using superconducting wire for the coils themselves. We've made some very preliminary estimates pertaining to possible coil and magnet designs which would be suitable for the secondary mirror actuators. Numerical estimates are described more fully in Sections VII and VIII.

Our initial concept is to use rectangular voice coils as shown in the cutaway view in Figure 3 with the magnets configured approximately as shown. The rectangular coil design seem to be a natural geometry for the two-axis system since the long side of the coil can reside in a rather narrow magnet gap with a correspondingly higher field, while the magnet gap at the ends of the voice coils will be larger to accommodate the motion produced by the perpendicular drive axis. The design may also help to minimize interactions between the two chopping axes.

The magnets themselves provide an interesting design problem in magnetic reluctance paths and flux densities. Mr. Lou Salerno has been of great help in our preliminary investigations of this problem by providing notes from a short course he attended which deal with exactly this issue.[13] Moving coil actuators have attracted significant attention in recent years for use in computer disk storage units where access time and head stability are critical factors. These are, of course, exactly the same problems which must be addressed for SIRTF.

Since the magnet design will require some detailed analysis, we have not had the opportunity within the scope of the current effort to do a full preliminary magnet design for our proposed system. We have developed, however, some initial concepts as indicated by the wedge-shaped geometry shown in Figure 3. This design should increase the volume of the magnets to allow for a larger volume of permanent magnet material, but special care will be required to obtain the correct cross sectional area and field-shaping geometry in the region near the gap[14].

6. The position sensors. The position sensors represent one of the least well defined problems at this point. While we have begun to collect some preliminary documentation on various commercial proximity sensors, we have certainly not yet found a device which seems to be the natural choice. Mike Dix[7] of NASA has done some preliminary work regarding capacitive sensors, and this avenue needs to be more fully explored. Capacitive sensors are particularly attractive from the standpoint of simplicity and very low power dissipation. Dix has also suggested a very simple drive circuit which may be able to meet the SIRTF sensitivity and bandwidth requirements, provided electronic components with adequate performance can be obtained. This issue will be one of the first which must be addressed in the prototype development program.

7. The reaction mass. The reaction mass for the proposed drive is shown schematically in Figure 2. Like the mirror, it will be supported on a universal flexural joint at its center of mass and is coupled to the mirror drive stem and voice coil assembly via a second flexural joint. We've also schematically shown two adjustable "bands" which allow the reaction mass moment of inertia to be trimmed by moving both bands toward or away from the flexural joint. The use of two bands allows adjustments in its moment of inertia without moving its center of mass.

Our description here of the adjustable bands should be considered as conceptual rather than as a final description of the exact implementation of the trimming adjustments. A final design for this concept must await a more firm design for the overall configuration for the reaction mass.

To summarize the above discussion, we have attempted to develop a design for the SIRTF secondary mirror system which addresses the critical issues while retaining the simplicity required for a highly reliable spaceborne system. In addition, we have tried to integrate the design such that its thermal and cryogenic characteristics represent sound design and are a natural consequence of the design geometry. In the next section we try to evaluate the degree to which this design actually meets these standards by examining it with respect to each of the issues identified in Section II as being critical to the success of the SIRTF program.

VI. DISCUSSION OF THE DESIGN

In Section II we discussed several issues that we believe will be crucial to the success of SIRTF, and which could be used to evaluate the suitability of any given design for the SIRTF application. The design we've discussed must, of course, be subject to exactly the same evaluation, and we will now try to make that comparison.

We first want to consider the firm numerical specifications established by NASA which are listed at the beginning of Section II. The two-axis drive system and circular symmetry of our mounting scheme should be particularly suitable for meeting the requirement for a variable chopping axis, and the magnetic actuators should be able to produce the required chopping waveforms. The system should also be suitable for combining the fine guidance control and chopping functions as required by the current SIRTF specifications.

The greatest uncertainty in our design, which we expect will be common to virtually any proposed design, will be its ability to meet the full range of specifications for chopping amplitude, chopping frequency, duty cycle, and thermal dissipation. Our major concern is the requirement for a settling time of 2.5 msec to a precision of 0.1 arcsec, which explains our emphasis on minimizing the moment of inertia of the system. The final performance achieved by our system will depend on the final combination of position and velocity transducers, servo-control system, actuators, and mirror assembly. However, based on moment of inertia estimates for our system and Mike Dix's preliminary work on the servo-control system, we appear to be in a favorable position in this respect.

In Section III we gave special emphasis to the problem of minimizing the total moment of inertia of the moving elements of the system. As we discussed, this is a vital consideration in meeting the ninety percent duty cycle specification for the SIRTF secondary mirror. However, recent discussions with NASA personnel have indicated that the requirement may be relaxed to read 70 to 90 percent, with a value of 80 percent being highly desirable. If we assume an 80 percent duty cycle, the allowed settling time for the mirror is doubled which results in a substantial relaxation of the requirements on the mirror and actuator system. Since our conceptual design was developed with the goal of achieving a ninety percent duty cycle, we have high confidence in being able to meet a relaxed specification of 80 percent.

The system should also have an excellent chance of meeting the requirements for thermal drift and total dissipation at the mirror mount. In our design there are no thermal sources or sinks located such that they will produce heat flows through the mirror proper. Furthermore, the location of the flex washer, which also serves as the thermal sink for the mirror assembly, enhances the thermal design of the system by providing an effective heat sink between the dissipative elements in the system and the mirror.

Finally, the system should also meet the specification for total dissipation, even possibly the very stringent specification for 50 mwatts of dissipation at the secondary mount. The use of superconducting wire will eliminate ohmic heating in the coils leaving only mechanical and eddy current dissipation. The estimates we give in Section VIII indicate a total dissipation from these sources of order 50 milliwatts. Measurements on the experimental actuators to be constructed in the first part of the development program should yield more definitive data on this issue.

To summarize, we feel that our design represents a well integrated approach to the SIRTF secondary mirror system, that it has a reasonable chance of achieving the current SIRTF numerical specifications, and that it is based on sound thermal and cryogenic design. We will now consider, item by item, the more general requirements discussed in Section II.

1. Reliability. Mechanically the system is extremely simple. Once the mirror and voice coils are assembled into a single unit there are only two moving parts -- the mirror assembly itself and its reaction mass. The single flexural joint on the mirror also represents an extremely simple implementation for the flexural pivot.

Perhaps the biggest question of reliability is the problem of contraction and the possibility of wire breakage in the voice coils during cooldown, but this will be a problem in any system using electromagnetic actuators. Another potential difficulty is the leads which will connect the moving coils to the secondary mirror structural supports. These leads may have to survive billions of cycles at cryogenic temperatures without breaking. However, the amplitude of oscillation is minute, and if the transition from the mirror to the stationary housing is designed to insure that the oscillations do not exceed the elastic limits of the connecting wires the system should be capable of reliable, long term operation.

2. Thermal dissipation in the mirror. A serious concern for SIRTF is the problem of thermal dissipation in the secondary mirror which can produce several severe problems for infrared observations. As we discussed in Section II, thermal dissipation in the mirror will produce temperature gradients and time-dependent temperature fluctuations in the mirror. If the magnitude of such thermal effects are large enough to affect the infrared detectors in the telescope, the thermal gradients and fluctuations will combine with the chopping action of the

secondary mirror to produce a spurious modulation in the detectors. Any increase in the temperature of the secondary mirror also contributes to the general infrared background level of the telescope.

We feel that our design is particularly strong in this respect. The actuators are located as far from the mirror as possible and there should be virtually no heating in any of the elements connected directly to the mirror. Furthermore, the structural support provides a natural radiation shield directly behind the mirror which can include a layer of high-mu material to shield the mirror from any stray magnetic fields. The geometry is also ideal for including a layer of high heat capacity material directly behind the mirror to help stabilize the temperature of the support structure, as suggested during a project review with NASA personnel near the beginning of our current study.

The real problem here is that of thermal stability in the mirror. The difficulty is rooted in the extremely small heat capacities of materials at low temperatures. We will quantify this in more detail in Section VIII where we consider the more general problem of thermal dissipation in the mirror system. We simply make the observation here that the 10 μKelvin specification for thermal drift in the mirror will tolerate virtually no dissipation in the mirror, and that any dissipation in the mirror assembly must be both physically and thermally isolated from the mirror. The geometry of our design naturally guarantees that the dissipative elements in the system are physically removed from the mirror. The required thermal isolation can be addressed in the design of the drive stem, and we take this issue up in more detail in Section VIII.

The only source of thermal dissipation in the immediate vicinity of the mirror will be from mechanical losses in the flex washer. While we have not yet tried to compute the magnitude of such dissipation, we expect that since the flex washer will have to operate within its elastic limits to achieve the required lifetime, the mechanical dissipation will be very small. Furthermore, at its outer diameter the washer itself is connected directly to the support structure which serves as the local heat sink for the secondary mirror. This combination should insure that any heat dissipated in the flexural joint should have an insignificant effect on the temperature of the mirror.

3. Distortions of the mirror. The next issue is distortions in the reflecting surface which will be produced by thermal cycling, forces which might be applied to the mirror during the chopping cycle (if voice coils were mounted directly on the back of the mirror, for example), and from any angular accelerations produced by the chopping motion. Our approach completely avoids the problem of forces applied to the outside edges of the mirror, so our main concerns here are flexing of the mirror as a result of the chopping action, and mirror distortions resulting from thermal

stresses developed during cooldown. The question of flexing under the chopping accelerations will have to addressed as part of the mirror design process, in which the mirror will be lightened as much as possible while maintaining sufficient rigidity to keep such flexing within allowable limits.

The mirror drive stem must also be considered, of course, but the length and other dimensions of the stem will be primarily determined by the actuator forces and the rigidity requirements between the back end of the drive stem and the mirror. Our design also offers an advantage here. Since the length of the entire assembly is not critical, the length of the drive stem can be optimized by balancing the increased mechanical advantage from a longer stem against its increasing moment of inertia and decreasing rigidity. Our preliminary estimates in Section VIII indicate that the increasing moment of the actuator coil cluster will limit the drive stem length to about 5 to 6 cm.

We have not yet investigated the question of distortions in the mirror as a result of thermal stresses developed during cooldown from room temperature. This question is one of the important aspects of the mirror design effort which will have to be performed at the beginning of the prototype development program. In terms of the impact of this issue on our design, we simply comment that this particular problem will be have to be solved regardless of the final actuator design.

4. Total thermal dissipation at the secondary mirror mount. In terms of a general design for the secondary mirror system, the total dissipation at the secondary mirror will be determined by such factors as ohmic heating in the actuator coils and electronics, eddy current dissipation and hysteresis losses in the voice coil magnets (especially if the magnets include a "shorted turn"), eddy current dissipation in the electrically conducting parts of the mirror assembly (as a result of the chopping motion), and mechanical dissipation in the flexural joints.

We have already discussed briefly the problem of mechanical dissipation in the flexural joints, and do not believe that this will represent a significant contribution to the dissipation at the secondary mirror mount. Furthermore, the rigidity requirements on the mirror assembly are so stringent that the tiny amplitude of permissible mechanical distortions in the mirror structure should insure that these contributions are also negligible. This conclusion should, however, be confirmed in the design stages of the prototype development.

Ohmic heating in the actuator coils of our design is eliminated by using superconducting voice coils for the actuators, and the actuator electronics can be located at a remote position. Consequently, the only electronics which may be required at the secondary mount would be preamplifiers for the position and velocity transducers which provide the inputs for the servo-control system. However, this requirement is not yet established, and if required, any such electronics will be mounted on the mirror housing rather than on the mirror assembly. Hence, its impact will be only in terms of the total dissipation requirement, and the required cryogenic electronics would have to be designed to meet that specification. This should not be a major problem, but it will not be resolved until the final choice of position and/or velocity sensors is made.

A substantial concern in the secondary mirror system is the possibility of eddy current losses in the magnets and in electrically conducting materials exposed to stray fields from the magnets. As we've discussed, the problem should be virtually eliminated in the drive stem itself by using only electrically insulating materials in regions where there are stray magnetic fields. In Section VIII we also address briefly the problem of losses in the permanent magnets. Since we have not actually designed the magnets, the calculations have to be taken only as extremely rough estimates to indicate an order of magnitude value. In designing the magnets we expect to use a "flux focusing" or field shaping technique similar to that described by Kimble[14]. Based on this concept and the hysteresis characteristics for commercial soft iron (which Kimble uses in his design), and using approximate values for the fields we expect from the voice coils, we expect the hysteresis losses to be of order 15 mwatts at 20 Hz. Using similarly crude estimates for the expected eddy current paths in the magnets, we estimate the eddy current dissipation to be possibly of the order of 50 mwatts for the four actuators.

While the calculations we've performed to date are only rough estimates, they are all generally consistent with SIRTF requirements. At the present it is not entirely clear that the system can meet the design goal of 50 mwatts dissipation at the secondary mount, but keeping it below the 200 mwatt maximum should not be a major problem. In a more comparative approach, we note that the design we have proposed should have a substantially better chance of meeting the dissipation requirements than any of the alternate designs we've reviewed. The arguments we present to support that statement include our use of nondissipative superconducting actuator coils, the low moment of inertia we can achieve with the drive stem design, and the elimination of electrically conducting materials from regions of high fields except for the permanent magnets.

5. Thermal management. We use the term "thermal management" in a qualitative sense to describe the overall effectiveness of a given design in isolating the secondary mirror from dissipative parts of the system, and in its ability to remove heat from the system without producing thermal gradients and fluctuations in mirror. Since eliminating both of these effects will be absolutely critical to obtaining the ultimate sensitivity from SIRTF, we feel that the question of thermal management warrants

consideration as a separate and specific issue.

Once the mirror has reached thermal equilibrium at its operating temperature, any remaining gradients must be the result of some continuous (but perhaps fluctuating) heat flow across the mirror, which, in turn, implies a thermal source and sink to sustain the heat flow. The solution, then, is to eliminate thermal sources which produce heat flow through the mirror, and to locate thermal connections to the mirror such that the effective heat sink is between the mirror and any dissipative elements in the system. We've already commented in Section V on some of the thermal aspects of our design; the following list provides a complete summary of the qualitative thermal characteristics of the system.

a. All of the dissipative components in the system are contained inside the secondary mirror housing which functions as the local heat sink for the system. This serves to completely isolate these components from the rest of the telescope.

b. Eddy current dissipation in the mirror can be virtually eliminated by using a layer of high-mu material directly behind the mirror to shield it from stray magnetic fields.

c. There are no sources of thermal dissipation in the mirror itself, and the only dissipative element near the mirror is the beryllium-copper flex washer which is directly attached to the mirror housing structure. As we commented above, mechanical dissipation in the flex joint should be negligible anyway.

d. All of the dissipation in the mirror assembly from the actuators occurs at the end of the mirror drive stem, as far as possible from the reflecting surface. Furthermore, dissipation at this position is minimized by using superconducting voice coils, and electrically insulating material for the coil carriers.

e. The thermal heat sink for the mirror lies between the mirror and the dissipative elements in the system so that the dissipated heat can be extracted from the system without producing heat flow and associated thermal gradients across the mirror.

f. A good thermal connection to the mirror assembly is provided by the flex washer as a natural consequence of the geometry of the design. Furthermore, the thermal sink for the mirror system occurs immediately behind the mirror insuring a low-impedance thermal connection between the mirror and the mirror housing. (We discuss this in detail in Section VIII.)

g. The permanent magnets, in which most of the dissipation will occur, will be thermally and mechanically attached directly to the mirror housing. This will prevent any significant increase in the temperature of the magnets during the operation of the mirror.

h. While it is not yet certain that the washer design for the flexural pivot will be acceptable, even if the design should revert to a gimballed mounting using standard flex pivots, the geometry is easily adaptable to the more brute force technique of using a high conductivity flexible braid. This would probably be implemented by using a braid with a geometry very similar to that of the flex washer with its inside diameter attached to the stem with a mechanical clamp, and its outer diameter anchored to the secondary mirror housing. Thermally, it would function exactly as the proposed flex washer.

To summarize, we have tried to take advantage of the natural geometry of the stem-drive design in providing adequate thermal connections to the mirror, and keeping the dissipative elements removed from the immediate vicinity of the mirror. We believe that the washer design will work well in the system, but this remains to be verified during the first part of the prototype development. If the technique is indeed acceptable, it will further strengthen the integral nature of the mechanical and thermal design of the system.

6. Compatibility with cryogenic environments. We believe that the design should be completely compatible with long-term, highly reliable operation in a cryogenic environment. Our choice of voice coil drivers eliminates any requirement to maintain extremely close tolerances as the system is cooled to operating temperature. Also the flex washer design allows a very simple connection between the mirror and washer with a simple ring clamp which can be easily designed to accommodate differential thermal contraction.

The one critical interface interface between different materials is the joint between the low thermal conductivity drive stem and the mirror. The design of this joint must be included as part of the mirror design, and we anticipate that some simple experiments will be required to demonstrate its reliability under thermal cycling. The other interface which must be designed is that between the drive stem and the carriage for the coil cluster. However, this interface is much less critical since small distortions during cooldown will be much less important and we will have much greater latitude in our material selection for the coil carriage. Consequently, we believe that the only real design problem will be the mirror/drive stem interface.

There is one final point here regarding the compatibility of our system with a cryogenic environment. It is interesting to note that that the operation of the voice coil drivers is substantially enhanced at low temperatures. Not only can we use superconducting voice coils, but the actuator performance is even further improved at low temperatures by the permanent magnets since both the remnant field and coercivity of samarium cobalt increase at low temperatures. This makes our selection of voice coil actuators seem particularly appropriate.

7. Provision for reaction mass. This is probably the weakest aspect of our proposed design. As in several other designs we've examined, our solution to this problem is to simply provide an appropriate reaction mass mounted directly behind the real secondary mirror. While the implementation illustrated in Figure 2 may be satisfactory, we have not yet reached a final decision on what we believe to be the best approach. In Section VIII we discuss two other alternatives to the reaction mass problem.

To conclude, we believe that the above discussion demonstrates that our design has a reasonable chance of meeting the full range of SIRTF specifications as they now exist. As we've commented previously, the evolution of the design was completely driven by those issues which we identified as being the most important to SIRTF. Consequently, it is hardly surprising that it should compare favorably when measured in terms of the very parameters which drove its development. Nonetheless, we certainly do not want to imply that all the problems are solved, and in the above discussions we have tried to identify the issues which must be addressed in substantially greater detail. In the next section we more completely summarize those problems which we believe represent the greatest obstacles to a successful system.

VII. THE MECHANICAL AND THERMAL DESIGN

Task 6 of our Statement of Work is to provide a recommendation for a specific cc ceptual design for SIRTF and provide justification for the recommendation, with particular attention to be given to the critical issues outlined in Section II above. In the preceding sections of this report we have tried to define the most important issues for SIRTF, describe the impact of those issues on the secondary mirror and its actuators, and demonstrate the way in which those factors drove the evolution of our design. In Section VI we presented arguments which we believe strongly support our recommendations.

Before discussing our outline for the prototype development and the major issues it must address, we will present some of the preliminary numerical estimates we've made in the process of examining various design alternatives. The estimates deal primarily with the central issues we've identified, such as the moment of inertia of the system, and not all of our results are entirely comforting. In those cases where the estimate may raise more questions than it answers, we have made some observations regarding design alternatives which may help alleviate part of the potential problem.

1. The moment of inertia of the system. A major emphasis throughout this study has been to reduce the moment of inertia of the system. To obtain a feeling for the actual moment to be expected from our design, we've have made some preliminary estimates for the mirror shape shown qualitatively in Figure 2.

The analytical expression for the shape of the mirror was derived from the equation for the deflection of the end of a horizontal rectangular beam which is supported at one end[15]. (This approximation was suggested by our initial attempts to calculate the deflection of the outer edge of the mirror under a specified angular acceleration.) While the functional form may not be exact, it should be a reasonable qualitative estimate for the required cross sectional shape of the mirror.

Assuming a beryllium mirror which is a solid of revolution having the cross section shown, we compute its moment of inertia to be approximately 4.5×10^{-5} kg-m^2 where we have used 1.8 gm/cm^3 for the density of beryllium. This value is encouraging, but we expect that the moment of inertia of an optimally designed mirror could possibly be more than a factor of two less than this. For example, we have assumed a solid mirror structure with the cross section shown, but this would probably not be the design for a mirror which was fully optimized to provide the greatest possible rigidity to weight ratio (or more correctly, rigidity to moment of inertia ratio). Two approaches we have briefly considered include a honeycombed structure for the mirror behind a thin layer which provides the reflecting surface, or alternatively, a ribbed structure in which radial ribs support the reflecting surface.

We expect that the correct approach to this problem will be to use a computer-based mechanical design package to analyze the amplitude and frequency of the normal modes for a variety of proposed designs. Since the mirror assembly will be a relatively complicated structure, a fully optimized design will almost certainly require the use of sophisticated numerical techniques.

Nonetheless, we can still make some initial estimates for the components of the mirror system based on more elementary considerations. For purposes of this rough calculation we will assume some type of ceramic material for the drive stem with a density of order 4 gm/cm^3. Using preliminary values for the expected moments and forces to be encountered, we estimate that a 6 cm hollow drive stem would need to be about 1.5 cm in diameter with a wall thickness of 0.25 cm to have the required rigidity. The moment of inertia for this structure rotating about one end will be about 2.9×10^{-5} $kg\text{-}m^2$. Since we are already assuming a hollow stem, this value can probably not be significantly reduced.

The last item to be added is the carriage for the voice coils which is attached to the extreme end of the drive stem. Because it moves at the largest radius, it is particularly important to minimize its mass. Without trying to do a detailed design for the voice coils, we've assumed a rectangular coil 1 cm by 2 cm with 300 turns of #38 niobium-titanium wire. This will give a mass of about 0.8 grams for each of the coils, assuming the coils are imbedded in epoxy after they are wound. We must also allow for the mass of the coil carriage. Using a very rough estimate for its shape and assuming that it is fabricated from some type of plastic material (having a density of order 2.0 gm/cm^3), we estimate a total mass for the coils and carriage of about 12 grams. The resulting moment of inertia for this mass moving on a lever arm of 6 cm is about 4.3×10^{-5} $kg\text{-}m^2$.

Adding these contributions gives an approximate moment of inertia for the mirror and voice coils of about 1.2×10^{-4} $kg\text{-}m^2$. We must also, however, assume an identical moment for the reaction mass, which gives a total moment for the system of 2.4×10^{-4} $kg\text{-}m^2$. Our first sobering observation is that this is more than a factor of two greater than our design goal. However, the following comments are relevant to this calculation.

a. First we have taken the moment of inertia for the mirror we calculated above which is appropriate for a solid mirror. We want to emphasize that we believe this to be a worst case estimate, and that it should be possible to achieve a value of order 2.5×10^{-5} $kg\text{-}m^2$ for the mirror itself. Note also that any reduction in the moment of the mirror reduces the total moment by twice that amount since the reaction mass moment is reduced the same amount. This observation also emphasizes the importance of establishing an optimized mirror design as quickly as possible.

b. A second improvement in the moment of the system can be realized by shortening the drive stem to 5 cm which gives values of about 1.7×10^{-5} kg-m^2 for the moment of the drive stem and 3.0×10^{-5} kg-m^2 for the coil cluster. If we use a value of 3×10^{-5} kg-m^2 for the mirror (which we feel is rather conservative) we estimate a total moment for the system of about 1.5×10^{-4} kg-m^2 (including the reaction mass) which is substantially closer to our goal of 1.0×10^{-4} kg-m^2.

c. We have also considered two more radical approaches to obtaining further reductions in the moment of inertia of the system one of which is shown in Figure 5. This configuration, in which the actuator coils are mounted on the reaction mass and coupled to the mirror through a flexural joint as shown, has the interesting numerical effect of placing the moment of inertia of the coils on the reaction mass rather than on the mirror giving a substantial reduction in the total moment of inertia of the system.

For example, if we take the moment of inertia for the mirror to be 3.0×10^{-5} kg-m^2 with a 5 cm drive stem, the total moment of the system could be of the order of 1.0×10^{-4} kg-m^2 since the coils would act to cancel the torque from the mirror rather than add to it. Our greatest objection to this technique is that it represents a substantial complication in the design of the combined mirror/actuator system, and interposes an additional flexural joint between the actuators and the mirror.

Another alternative approach would be simply to construct the reaction mass as a completely separate unit having its own voice coils and magnets, but driven synchronously with the real mirror. This has the distinct advantage of mechanically decoupling the reaction mass from the actual mirror while still providing a reactionless system. The primary disadvantages are the added weight of a second complete set of permanent magnets at the secondary mount, and an increase in the thermal dissipation at the secondary mirror mount. However, since each of the eight actuators would have to deliver only about half of the force required if only four actuators were present, we estimate that the total dissipation from the actuators (both electronic and dissipative) would increase by only about 25 percent.

d. Our final observation here concerns recent indications that the SIRTF duty cycle specification may be relaxed to require only an 80 percent duty cycle. As we have commented previously, this will result in a significant relaxation of the performance requirements for the actuators and servo-control system, since an 80 percent duty cycle means that the settling time will be doubled. If, indeed, the SIRTF duty cycle specification is

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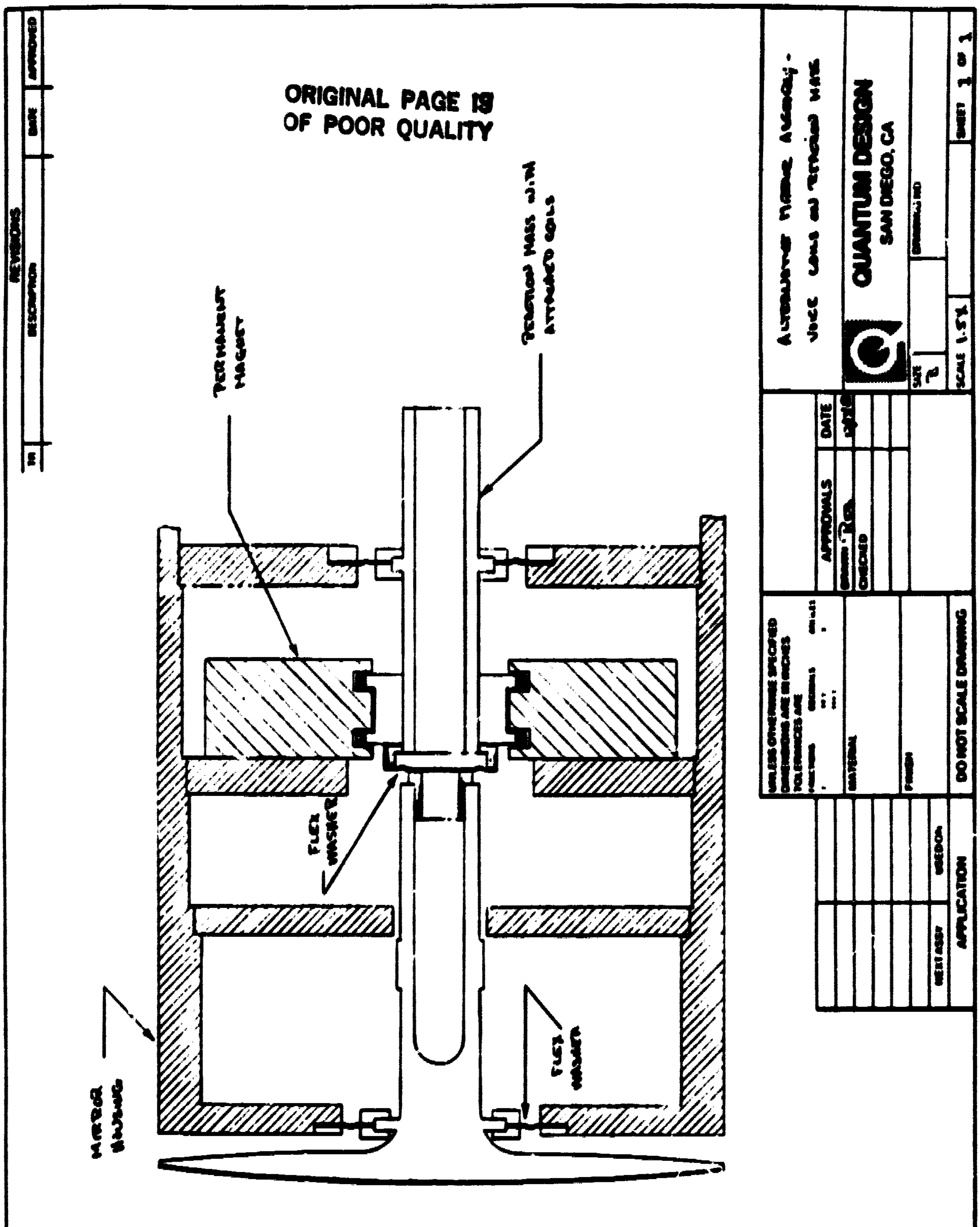

FIGURE 5

relaxed, the importance of achieving the absolute minimum moment of inertia will be correspondingly reduced.

To summarize the above observations, it is apparent that there is no clearcut choice for implementing the reaction mass, and this is consistent with our earlier comment that this aspect of our design is probably the weakest. A more firm decision will have to await a significantly more detailed system design effort than we have undertaken here.

2. A second critical issue for the SIRTF secondary mirror is the question of thermal dissipation at the secondary mirror mount. This is a another particularly difficult problem to solve analytically for a complicated shape, but one can make some simplifying assumptions in our geometry. Specifically, since there are no sources of thermal dissipation in our secondary mirror design which can introduce heat directly into the reflecting portion of the mirror, our primary concerns are that the flexural joint provide adequate thermal contact to the mirror housing, and that dissipation at the end of the mirror drive stem be sufficiently small to prevent significant thermal fluctuations at the mirror.

There is one very important point we wish to make here regarding thermal stability in the mirror structure. Because of the extremely low heat capacity of beryllium, and indeed nearly all solid materials at temperatures below 10 Kelvin, only very small amounts of heat will cause significant temperature fluctuations. For example, let us compute the approximate the total mass of the mirror by a disk 10.7 cm in diameter with an average thickness of 0.5 cm, which gives a total volume of about 45 cm^3. Now the heat capacity of beryllium at 7 Kelvin is of order 3.6×10^{-4} joules/K-cm^3, so that the total heat capacity of the entire mirror is about .016 joules/Kelvin. This means that a temperature change in the mirror of 10 microKelvin (the SIRTF specification) corresponds to a total energy fluctuation of 0.2 microjoules in the entire mirror. This is not a comforting observation.

This observation clearly demonstrates that any system which produces any thermal dissipation in or near the mirror will have no chance of even approaching the thermal stability requirement for SIRTF. The calculation also effectively demonstrates the necessity for an extremely low thermal impedance connection between the mirror assembly and the mirror housing, even in our design where all of the dissipative elements are located relatively far away from the reflecting surface.

We do not pretend to have adequately addressed this problem in this preliminary effort. However, because of our physical geometry and the effort we have made to minimize the total amount of energy which must be dealt with in our system, we believe that our design can deal with the problem at least as well as any other design we've considered. This discussion also explains our insistence on using an electrical insulator for the drive stem, which must extend into a region where there will be stray fields from the magnets, even at the expense of creating a difficult and critical joint between the drive stem and the mirror. It is clear from the above discussion that every effort must be made to virtually eliminate thermal dissipation in the mirror assembly, and insure that even tiny amounts of dissipation occur as far from the mirror as possible. The discussion in Section VIII provides further insight into this problem.

3. A problem which has been observed in other articulated secondary mirror systems[10,11] is that of mechanical resonances. Mike Dix and Lou Salerno of NASA have also recently encountered some problems with resonances in their first test device[1,7]. Specifically, they report two resonances - one at about 30 Hz having a Q of about 6, and a second at about 1.5 KHz, also with a Q of about 5 to 6. We have also given some consideration to the problem of mechanical resonances in our system.

In Section IV we commented on the problem of mechanical resonances in the context of the actuators for the system. Specifically, we noted that the actuator devices need to be free from any resonances which can interfere with the operation of the servo-control system (or more specifically, lie within the bandwidth of the feedback system). This is, of course, also true of the entire mechanical structure of the mirror system, and this has been a major consideration in our efforts to keep the system mechanically simple. Nonetheless, there are three components of our system which must can generate mechanical resonances which we've considered.

a. One possible resonance could be generated by flexing of the drive stem with respect to the front part of the mirror. To estimate the force constant associated with this type of oscillation, we used the expression for the deflection of a beam supported at one end with a force applied to the other end[15]. While we have not made a final decision on the material to be used for the drive stem, to illustrate the problem we will consider a drive stem fabricated from an aluminum oxide ceramic material which has a rigidity similar to that of beryllium[16].

Assuming a stem geometry of 1.5 cm outside diameter, 1.0 cm inside diameter and 5 cm long, we found a force constant of about 4.9 x 10^5 newtons/radian. For the moment of inertia of the drive stem we use the value computed above (including the coil assembly) which gives a total moment of about 4.6 x 10^{-5} kg-m^2. The resonant frequency of the system can now be estimated as for a simple oscillatory system from:

$$w_o = [k/I]^{1/2}$$

where k is the effective torsional spring constant and I is the

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moment of inertia of the system. This gives a resonant frequency of about 16 kHz for the tail stem.

Mike Dix of NASA has been performing some preliminary calculations on the servo-control system for the secondary mirror which indicate that the required bandwidth must extend up to about 3 to 4 kHz to meet the secondary mirror drive specifications[1,7]. Hence, the tail stem resonance in our preliminary design should not present an insurmountable obstacle. However, this issue should receive careful attention in the initial mirror design to insure that the resonance is kept at as high a frequency as possible.

There are at least two straightfoward ways to push this resonance to a higher frequency. First we can increase the diameter of the tail stem which will substantially increase its rigidity with only a nominal increase in the moment of inertia. (The rigidity of the stem is roughly proportional to the fourth power of its diameter while the moment of inertia we are using is dominated by the contribution from the voice coils.) Alternatively, we can design the stem to have a thicker wall thickness near the flexural joint where it experiences the greatest bending moment. A more detailed calculation for the stem resonance should be done in the initial stages of the mirror system design.

b. A second source of mechanical resonance is in the mirror structure itself. This problem must be solved as an integral part of the mirror design, however, and will need to be addressed along with such other considerations as mirror distortions under the chopping accelerations, damping rates of structural resonances excited by the chopping, and the exact mechanical design of the mirror for optimized performance. This comment is not to be construed as simply a handwaving dismissal of the problem, but rather to emphasize the importance of the secondary mirror design and of performing some of the initial computations for its design at an early date. Finally, we do not believe that this is a problem which can be approached in any simplified manner, and the analysis which would be required to obtain even preliminary results is beyond the scope of our current preliminary work.

c. The other resonance which will certainly appear in the response function for the system is that determined by the restoring force of the flex washer acting on the mirror and drive stem assembly. This will not be a high frequency resonance, and will certainly fall within the control system bandwidth. However, provided the Q of the resonance is not too large and does not occur at too high a frequency, it should be possible to keep the response of the closed loop control system reasonably uniform.

As above, we estimate the resonant frequency for the system from an effective spring constant for the flex washer and the moment of inertia upon which the restoring torque acts. In this case we use the moment of inertia for the mirror and drive stem assembly (the reaction mass has its own flex washer) as computed above, and we make a crude estimate for the effective restoring torque to be of order 20 newtons/radian for small deflections. This gives a resonant frequency of about 75 Hz. Recall that Lou Salerno and Mike Dix of NASA found a resonance at about 30 Hz with a Q of 6 in their system which they interpret as the natural frequency of the flex pivots and lever arm in the device.

Because of the very small moment of inertia of the mirror assembly, any type of flexural joint will produce a low frequency mechanical resonance in the secondary mirror system somewhere between 10 and 100 Hz. In actual operation, the effective Q of this type of resonance should be damped by the effects of hysteresis and eddy current dissipation in the permanen magnets of the voice coil drivers. Consequently, the resonance should not produce any extremely high Q effects.

Furthermore, by changing the design of the flexural support, we can increase or decrease the effective mechanical resonant frequency to help optimize the performance of the servo-control system. Further calculations which address this issue in detail and model the system response will be part of the initial design when developing an operating prototype.

In this section, we've tried to identify the thermal and mechanical issues which can be reasonably treated in a simple manner and make some preliminary numerical estimates to convince ourselves that our recommendations represent the approach which has the best chance of meeting SIRTF specifications. In the next section we consider explicitly the problems and strengths of using superconducting actuators and discuss further the ramifications of thermal dissipation in the mirror assembly with respect to its required temperature stability.

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VIII. SUPERCONDUCTING ACTUATORS AND THE CRYOGENIC DESIGN

The SIRTF specifications place such stringent requirements on the system that the use of superconducting actuators and the detailed cryogenic aspects deserve special consideration. In this section we want to explicitly address these facets of the design problem. The overall sensitivity specification for the SIRTF telescope further dictates the operating temperature and thermal uniformity constraints placed on the secondary mirror, which are closely tied to its thermal design. While we discussed some aspects of the thermal design in the previous section, we will extend that discussion somewhat in the context of the actuators and cryogenic design of the system.

1. The voice coils. In Section IV we recommended the use of moving coil actuators with a superconducting coil assembly. Before proceeding further, it is legitimate to ask if the use of superconducting wire in the voice coil is really necessary. The answer to that question is probably clear from considering the ohmic heating one might expect from a copper voice coil. We can make a first order estimate based on some copper voice coils available from Kimco, Inc.[7] which have a force constant of order 4.5 newtons/amp, and are constructed of 76 turns of #34 copper wire on a 2.5 cm diameter. Using the resistivity of copper at 6 Kelvin[17] (1.6×10^{-8} ohm-cm) we find a coil resistance of order 4.4×10^{-3} ohms for each actuator. Considering the ninety percent duty cycle and assuming that a current of about 10 amps will be required to deliver the required force, we find the ohmic heating to be of order 40 mwatts per actuator, or about 80 mwatts for a simple chopping motion using only one pair of actuators. Mike Dix of NASA has also made this calculation and has found an even higher value for the expected dissipation.[7]

This is a most compelling argument for using superconducting actuators. Even this rough estimate shows that copper voice coils will dissipate a significant part of the maximum allowable dissipation at the secondary, and that there is probably no chance of meeting the design goal of of 50 mwatts without using superconducting techniques. In addition, our estimates regarding the thermal stability requirements for the mirror from Section VII indicate that any dissipative elements attached to the mirror which produce more than a few milliwatts of heating in the mirror assembly will probably compromise the thermal properties of the mirror. All of these factors strongly support the use of a superconducting actuator.

We have also made some very preliminary estimates concerning the nature of the superconducting coils and the electronics which will be required to drive them. As we mentioned in Section VII, our initial concept is for a rectangular voice coil comprised of about

300 turns of #38 Nb-Ti superconducting wire on a 1 cm by 2 cm rectangular form. This coil will have an inductance of about 1.5 milliHenrys[18]. While the coil inductance will be somewhat altered by the nearby iron in the permanent magnet, its effect should not be drastic since the iron will be essentially held in saturation by the $SmCo^5$ magnet core. For these rough estimates, let us consider further the ramifications of this preliminary design using the value computed above.

We first consider the inherent L/R time constant of the coil itself. Since the dc resistance of the coil is zero, its L/R time constant will effectively be determined by the output impedance of the current source. Since our coil inductance is only of order 1 mH, we should be able to achieve an effective time constant which will not have an adverse impact on the servo-control system.

Another important issue is the force constant for the coil and magnet design, which will determine the total current required to achieve the required force from the push-pull actuator pair. If we scale the expected force constant for this design based on the parameters of the Kimco coils described above, we would expect to achieve a value of about 11 newtons/amp for each actuator. This has the desirable effect of substantially reducing the required currents to drive the actuators. However, the large number of turns also increases the coil inductance and limits the rate at which we can change the current through the coil according to:

$$V = L\dot{i}$$

where V is the applied voltage, L is the coil inductance, and $\dot{i}$ is the rate of change of current in the coil. Since the coil has zero resistance, this rate of change should be constant as long as the voltage V is maintained. (This is true as long as the current source can maintain its output voltage with increasing current. In a normal coil, of course, the rate of change of current decreases as the current increases due to the voltage required to overcome the resistive component of the coil's impedance.)

To get a feeling for the voltage required, we again refer to Dix's calculations[7] which indicate that a maximum force of about 100 newtons is required to drive a moment of inertia of 1.6×10^{-4} $kg\text{-}m^2$ over the required throw for a 90% duty cycle. If the maximum current is to be achieved in about 250 microseconds, this gives a rate of change of about 1.8×10^4 amps/second requiring a voltage across the coil of about 27 volts. This seems quite reasonable. Furthermore, the rate at which changes in the coil current can be accomplished can be enhanced by the use of the "shorted turn" concept[13].

A vigorous analysis of the system's behavior is beyond the scope of this study. Nonetheless, these rough initial estimates appear encouraging, and a more detailed analysis in the first part of a prototype development should clarify all of the most important issues in the actuator design. We do believe at this early stage, however, that superconducting actuators are not only a natural choice for SIRTF, but will almost certainly be required to even approach the current SIRTF requirements.

2. The permanent magnets. The requirements for dissipation in the voice coil magnets are, fortunately, much less severe. Since we do not have a firm magnet design, we cannot yet perform any precise modelling calculations for the dissipation in the magnet material due to hysteresis and eddy current losses. However, we can make some crude guesses to get an estimate for their general magnitude.

First let us assume that the magnet design uses Kimbles's concept of "flux focusing" so that there is a high reluctance path of soft iron surrounding a central permanent core of samarium cobalt[14]. Then using a typical hysteresis curve for commercial soft iron[19], we find that a complete transition of the hysteresis curve at 20 Hertz gives a total hysteresis dissipation of approximately 6 $mwatts/cm^3$. However, for the type of magnets we are considering, magnets are typically designed such that the soft iron is operated with a flux density sufficiently high to saturate the soft iron[13]. Consequently, even when a reverse field from the voice coil is applied to the magnet, the demagnetizing effect is only a few percent. (Note that if the iron is operated in complete saturation, there will be no direct hysteresis loss.)

Hence, we should be able to get a very rough order of magnitude estimate for the hysteresis loss by assuming that the soft iron will be cycled through, say, 5 percent of its hysteresis loop, and computing the total volume of iron subjected to the field from the voice coil. Taking 5 percent of the value computed above, we find a value of about .3 $mwatts/cm^3$ for the hysteresis loss in the iron at 20 Hz. Making a rather conservative estimate for the approximate volume of iron to be subjected to the demagnetizing field (that is, a larger volume than we actually expect to be affected) to be about 50 cm^3, we find a total hysteresis loss of order 15 mwatts for all four actuators.

The question of eddy current dissipation in the magnets is not so easily analyzed with this type of simple approach, since it will depend on the particular current path in the magnet. However, in the region nearest the voice coil, we might assume a current path in the magnet pole tip flowing in response to the demagnetizing flux generated by the voice coil. Again trying for only an order of magnitude estimate, let's assume that the demagnetizing flux penetration depth into the pole tip is of order 10 percent of its thickness, giving a cross-sectional area for the flux penetration of order .025 cm around the coil circumference of about 6 cm. Further assume a cross sectional area for the current path around the pole tip of order 0.25 cm x 0.5 cm. This gives a value for the eddy current dissipation of order 12 mwatts per actuator averaged over the entire chopping cycle, or about 50 mwatts for all four actuators.

This estimate is not particularly comforting, but because of its extremely crude nature, it is not clear whether this estimate is too high or to low. Consequently, a much more detailed analysis will be required when the complete magnet design is developed to insure that the dissipation will meet the system specifications. However, since this may be the most significant source of dissipation at the secondary mount, one further interesting observation can be made regarding the eddy current magnet losses as a function of the chopping amplitude and frequency. In Figure 6 we show the expected eddy current heating versus frequency for several values of the chopping amplitude for a 90 percent duty cycle. The dissipation should decrease as the square of both the amplitude and frequency as shown in Figure 6, where we have scaled the plot to 50 mwatts of dissipation for a peak-to-peak chopping amplitude of 28 arcmin at a 20 Hz chopping frequency.

One final loss mechanism which may contribute to the dissipation in the magnets may be the "shorted turn" provided to enhance the rise time for the current in the voice coil. Since dissipation in the shorted turn will depend strongly on the final parameters of the actuator, we have left this analysis for the more detailed design during the prototype development. However, we can make two relevant comments. First, since the dissipation will occur in the permanent magnets and not in the mirror assembly, it will not be critical in terms of thermal effects in the mirror. Secondly, the shorted turn can be custom designed, or perhaps completely eliminated, if its dissipation is excessive. Consequently, we do not view the shorted turn design as a critical issue at this time.

3. Thermal aspects of the cryogenic design. This discussion is essentially an expansion of our comments in the previous section on the thermal nature of the system. Nonetheless it is appropriate to consider the issue hear at greater length since it presents further support for our selection of superconducting voice coils. The primary point we wish to reemphasize is the problem of dissipation in the voice coils at the end of the drive stem.

To obtain an appreciation for the problem, consider the thermal connection between the mirror assembly and the mirror housing, in terms of the following simple model. To meet the SIRTF specification for thermal drift in the mirror, let us assume that the maximum allowable temperature fluctuation in the drive stem at the flexural pivot is about 10 microKelvin. Let us further assume that the thermal connection is made of a solid copper washer with and outer diameter of 3 cm, an inner diameter of 2 cm, and a thickness of .06 cm, dimensions which are roughly characteristic of our proposed flex washer. Note, however, that we are assuming

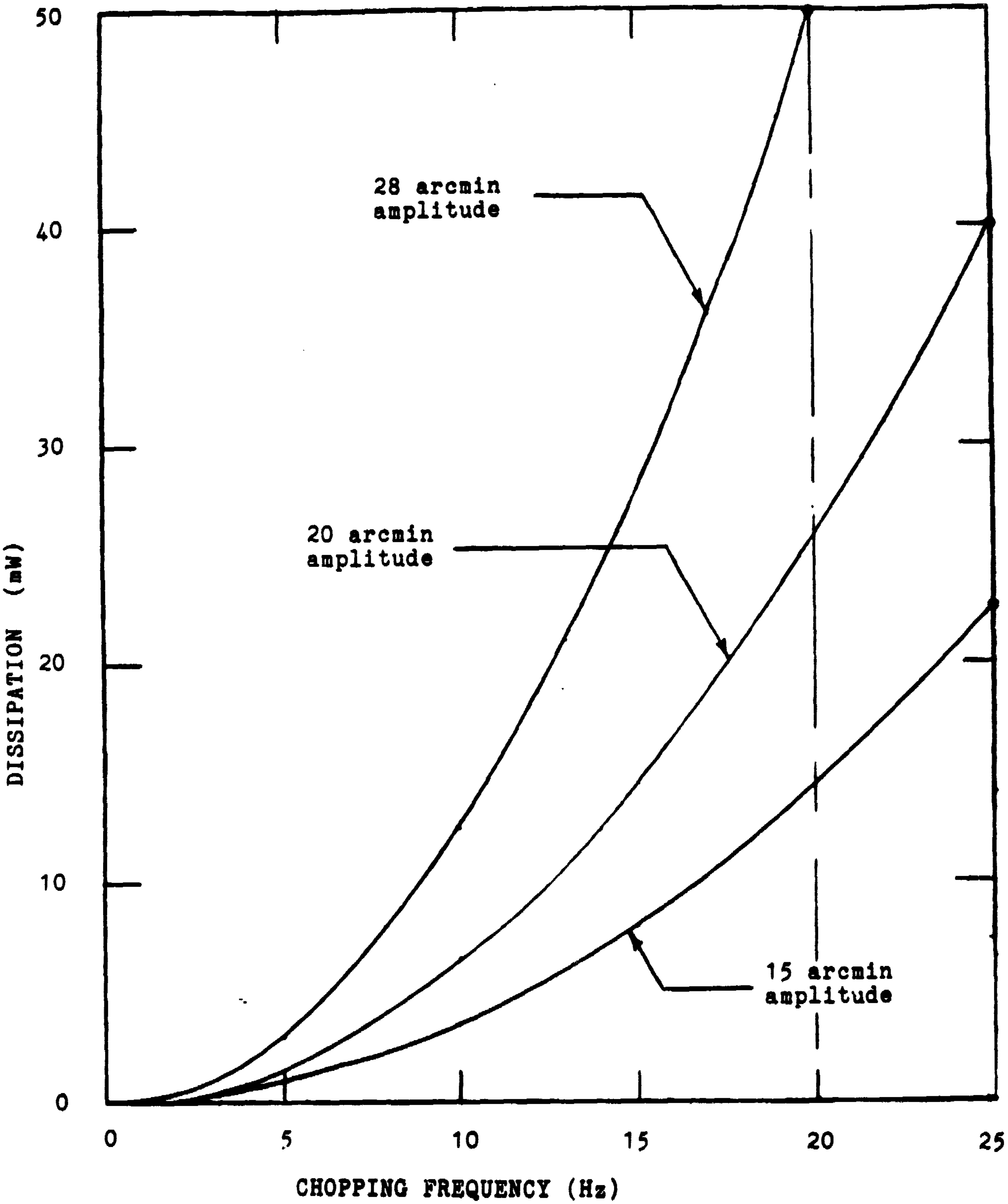

Figure 6. Estimated eddy current losses in the permanent magnets as function of chopping frequency for three different chopping amplitudes

pure copper rather than the beryllium-copper alloy from which the actual flex washer will be fabricated to illustrate the magnitude of the problem.

Using the thermal conductivity of pure copper, we find that a steady heat flow of 0.3 milliwatts from the mirror stem through the copper washer will produce a temperature difference of 10 uKelvin between the inside and outside diameters of the washer. Also note that the thermal conductivity of copper is about a thousand times greater than that of beryllium-copper. The most obvious conclusions from this exercise are the necessity to literally eliminate dissipation in the mirror assembly, and to develop a mirror stem design which has the proper thermal impedance characteristics and thermal connections to the mirror housing.

In summarizing the discussions in this sections we are led to mention some of the potential difficulties to be faced in actually constructing any system to meet the specifications we are considering. First, the system requires that energies of order 100 mwatts be delivered to the system to drive the mirror assembly, yet the thermal stability specifications for the mirror require energy stabilities corresponding to a microwatt or less. The ramifications of this will be the virtual elimination of dissipation in the mirror assembly, and a thermal design which must deal with temperature gradients of microwatts. These values are reminiscent of the thermal isolation requirements encountered in working a temperatures of a few milliKelvin.

With respect to the actuators, we must deal with very small superconducting wire (to reduce the mass of the coils and keep their moment of inertia small) which must operate in a high magnetic field and carry relatively large currents at a temperature only about 25% below the critical temperature of the wire. Furthermore, the nature of the moving coil actuators will require that reliable high-current joints be constructed between the superconducting actuators and the power leads which will connect those coils to the drive electronics.

There are also several other problems to be solved which are characteristic of instrumentation which must operate at low temperature, such as the issues of the heat capacities, thermal conductivities, and differential contraction of materials. The successful resolution of this entire ensemble of difficult problems will require a careful implementation of a variety of cryogenic techniques and some subtle low temperature engineering.

IX. IMPORTANT ISSUES TO BE RESOLVED

While solving many of the thermal and cryogenic problems in the secondary mirror system, there are several issues in our proposed design which must be resolved at an early point in the development of a prototype. For some of the potential obstacles, there appear to be viable, straightforward alternatives which should reduce the associated risk. Not all of the issues, however, can be readily sidestepped. Nonetheless, we believe that overall our design has an excellent chance of solving the remaining problems.

The development program will have to resolve several major issues to provide a performance demonstration for the secondary mirror drive system. We have identified the following items as the most items to be addressed.

1. Two-axis chopping and control. Using a combination of two independent chopping axes to generate an arbitrary chopping axis probably carries the greatest risk of all the SIRTF specifications. The major difficulty we expect to encounter is the possible interaction between the servo-control systems for the two chopping axes which may produce instabilities in the behavior of the mirror control. The resolution of this question will undoubtedly require a combination of laboratory measurements on the actual system and knowledgeable theoretical modeling of the servo-system behavior.

We have already given some thought to this problem. For example, the rectangular voice coil geometry was selected to help optimize the actuator performance yet allow two-axis motion of the single drive stem. Also if capacitative sensors are used to provide position information for the servo-control system, they can be geometrically configured to be insensitive to chopping about an orthogonal axis. More in-depth considerations will be required as a detailed prototype design is developed. These comments are intended to emphasize the importance we place on this issue, and the potentially difficult problems presented by cross-talk between the two independent chopping axes.

2. The control system. As a separate issue from the question of developing a full two-axis chopping capability, the complete set of SIRTF specifications place stringent requirements on the system. This is reflected in our concern for optimizing the system, especially its moment of inertia. Nonetheless, even with our design we feel that our goal of meeting the full range of numerical specifications set forth for SIRTF is still very much in question.

As we have emphasized throughout this report, the greatest uncertainty lies in the difficulty of meeting the settling time specification at maximum chopping amplitude and frequency. Dix's calculations[7] indicate that our servo-control/actuator combination will approach the required performance, but we have yet to fully account for the practical experimental difficulties we will undoubtedly encounter.

3. Thermal effects in the secondary mirror. Clearly another important problem will be the thermal isolation of the mirror and the elimination of dissipation in the mirror assembly. One problem in dealing with the thermal character of the mirror is that experimental measurements to verify its performance will be difficult, even once the final prototype unit has been fitted with a real mirror. Hence, we feel that it will be essential to present convincing evidence for the thermal nature of any proposed system before making final design decisions.

In this context, we can make one point with respect to our proposed design. All of the major dissipative elements in the system are connected to the mirror only through the drive stem, and there will be essentially no thermal conduction through the mirror. Furthermore since all of the thermal sources and sinks in the mirror assembly are localized and can be well defined, we should be able to construct an excellent thermal model for the entire mirror assembly. Consequently, we believe that careful thermal modelling of our system in the design stage should provide the required confidence in its predicted thermal performance.

4. Resolution of system resonances. Another potential problem in this system is the requirement for a relatively wide bandwidth for the servo-control system, and the possibility of the mirror/actuator system generating mechanical resonances at frequencies within that bandwidth. We discuss this point in some detail in Section VIII where we identify three potential resonances in our system - oscillations of the tail stem, oscillations in the mirror structure itself, and the natural resonance associated with the mirror system inertia and the restoring force generated by the flex pivot. While only the last of these three resonances should significantly affect our design, its greatest impact will probably be felt in trying to meet the pointing accuracy and settling time requirements. We make some simple estimates regarding these resonances in Section VIII; a more detailed analysis will have to be performed when a prototype device is designed.

5. Position and velocity sensors. Mike Dix of NASA has made some preliminary investigations into this problem which indicate that a capacitive position sensing system may provide the requisite sensitivity for this application, if the excitation and detection electronics are properly designed[7]. We have also discussed the possibility of using optical position sensors and possible configurations for velocity transducers which would provide direct velocity information for the servo-control system. This may well be a necessity for SIRTF since differentiating a

position signal can introduce additional errors, not inherently related to the position sensors, into the servo-control loop.

In addition to our preliminary discussions with NASA personnel regarding different types of sensors, we have also begun to accumulate information on commercially available proximity sensors which might be suitable. Beyond these very preliminary steps, however, we have given little further consideration to this aspect of the design. Consequently, while we believe that this issue carries somewhat less risk than the others we've discussed, it will have to receive significant attention during the prototype design, including an extension of the laboratory tests Mike Dix has been performing.

6. The flex washer. The only item that truly represents a developmental effort is the design of the flex washer, which is the only substantial departure of our design from some other previous articulated mirror designs[10,11]. An important point here is that, even if the concept should fail catastrophically, a simple alternative is immediately available in the form of a gimballed mounting scheme based on conventional flex pivots. There is nothing in our design which inherently prohibits the gimballed mount - but a gimballed mount will be much more difficult to implement in a system which requires that the moment of inertia be kept small. Consequently, while the fabrication and testing of the flex washer will require a nominal investment to determine whether it is suitable for our application, its inherent simplicity and contribution to the thermal design of the system make it attractive if it can meet the system specifications.

7. Another important issue which must be addressed at the beginning of the prototype development is the exact design and material selection for the secondary mirror. There are several aspects to this problem, as we list below, all of which must receive substantial attention at the beginning of the prototype development.

a. The mirror design must be optimized to minimize its moment of inertia while providing the requisite rigidity for the structure.

b. The question of distortions in the mirror during cooldown must be investigated, and a decision reached concerning the material selection for the mirror.

c. Additional work is needed to more fully investigate the problem of possible thermal fluctuations being generated in the mirror by dissipative effects in the mirror drive stem. This work needs to specifically address the problem of interfacing the mirror and drive stem.

Compiling the above list of important issues for developing a secondary mirror prototype has prompted us to make some preliminary numerical estimates for some of the issues which can be treated, at least initially, in a more simple-minded manner. In the next section we present some of these very preliminary calculations along with some additional thoughts on our conceptual design. In Section IX we set forth a program outline for actually developing an operating prototype of our proposed secondary mirror system.

X. THE PROTOTYPE DEVELOPMENT PROGRAM

In reviewing our discussion of the major issues which must be resolved in developing an operating prototype for the secondary mirror system, we make the somewhat disturbing observation that those issues which appear to contain the greatest uncertainties (pointing control and two axis chopping capability) could normally not be resolved until the actual operating model has been constructed and subjected to experimental measurements. For precisely this reason we feel that it will be extremely important to construct a preliminary brassboard model of the device at the earliest moment, with the proviso, of course, that the model can represent the real system closely enough to provide meaningful data.

In our schedule for the development program presented below, we have tried to address this problem by identifying those items which must be done before a simple model can be constructed and completing this work as quickly as possible. By scheduling these tasks near the beginning of the program, we hope to quickly reach the point where we can construct a simple model to operate at room temperature which can be used to resolve some of the important questions concerning the servo-control system. These tests can then proceed in parallel with other tasks which deal more with the general prototype design and the cryogenic aspects of the system.

In Figure 7a we present a time chart for the first part of the project which identifies significant milestones and shows the approximate calendar schedule for their completion. The most important goals to be achieved early in the program include:

a. A detailed analysis for an optimal mirror design.

b. The design and testing of the flex washer joint.

c. The design and testing of superconducting actuators.

d. Identification and testing of appropriate position and velocity sensors.

e. The design and construction of a room temperature model to resolve issues concerning the servo-control system.

Included implicitly in the above tasks is the design work which will help determine the precise design of the mirror system, such as the exact geometry for the magnets and actuator coils, more detailed estimates for hysteresis and eddy current losses in the system, thermal and mechanical modelling of the structure, and the details of the mechanical design for interfacing the various components of the mirror assembly.

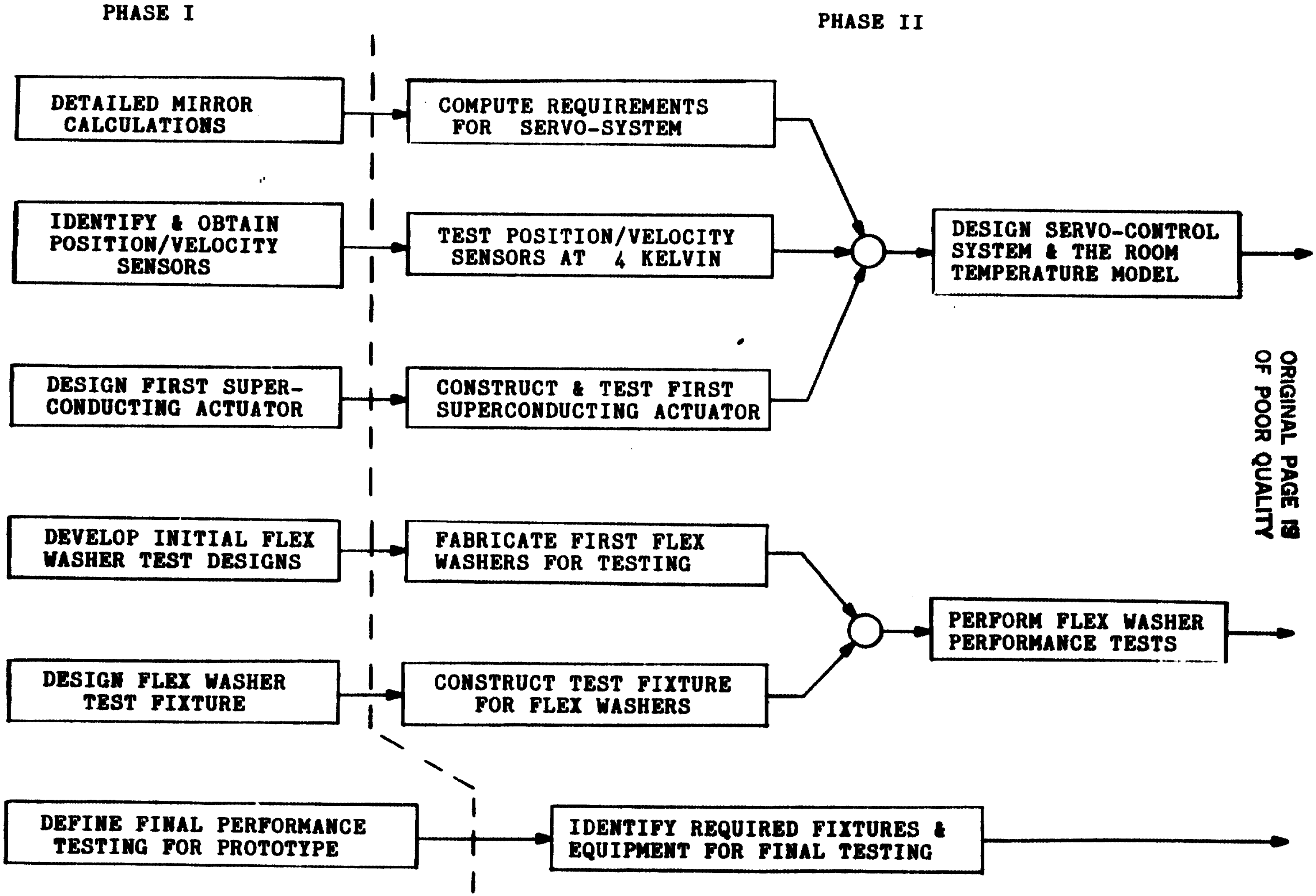
PHASE I
PHASE II
DETAILED MIRROR CALCULATIONS
COMPUTE REQUIREMENTS FOR SERVO-SYSTEM
IDENTIFY & OBTAIN POSITION/VELOCITY SENSORS
TEST POSITION/VELOCITY SENSORS AT 4 KELVIN
DESIGN SERVO-CONTROL SYSTEM & THE ROOM TEMPERATURE MODEL
DESIGN FIRST SUPER-CONDUCTING ACTUATOR
CONSTRUCT & TEST FIRST SUPERCONDUCTING ACTUATOR
DEVELOP INITIAL FLEX WASHER TEST DESIGNS
FABRICATE FIRST FLEX WASHERS FOR TESTING
PERFORM FLEX WASHER PERFORMANCE TESTS
DESIGN FLEX WASHER TEST FIXTURE
CONSTRUCT TEST FIXTURE FOR FLEX WASHERS
DEFINE FINAL PERFORMANCE TESTING FOR PROTOTYPE
IDENTIFY REQUIRED FIXTURES & EQUIPMENT FOR FINAL TESTING

Figure 7a

PHASE III

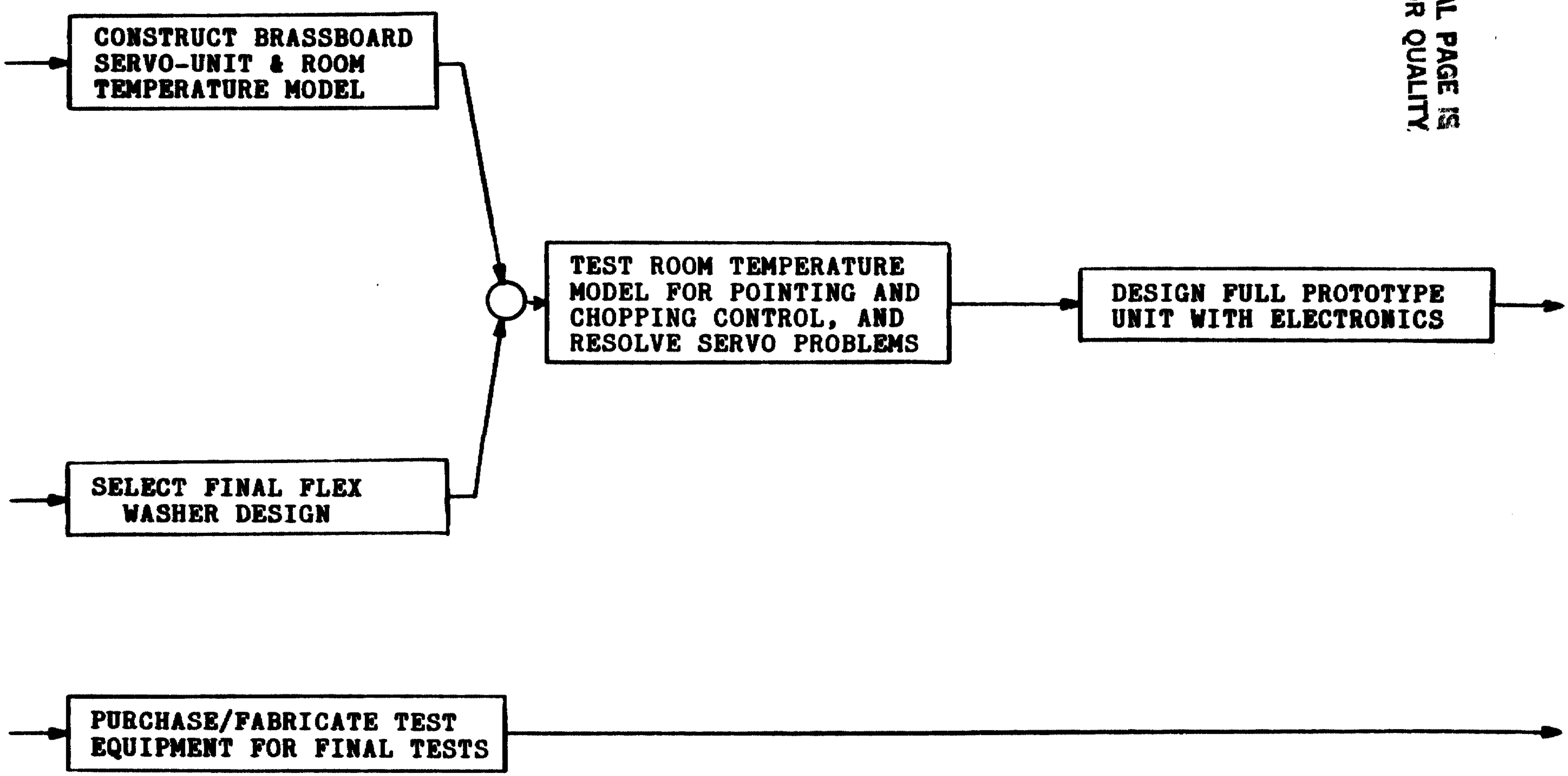

Figure 7b

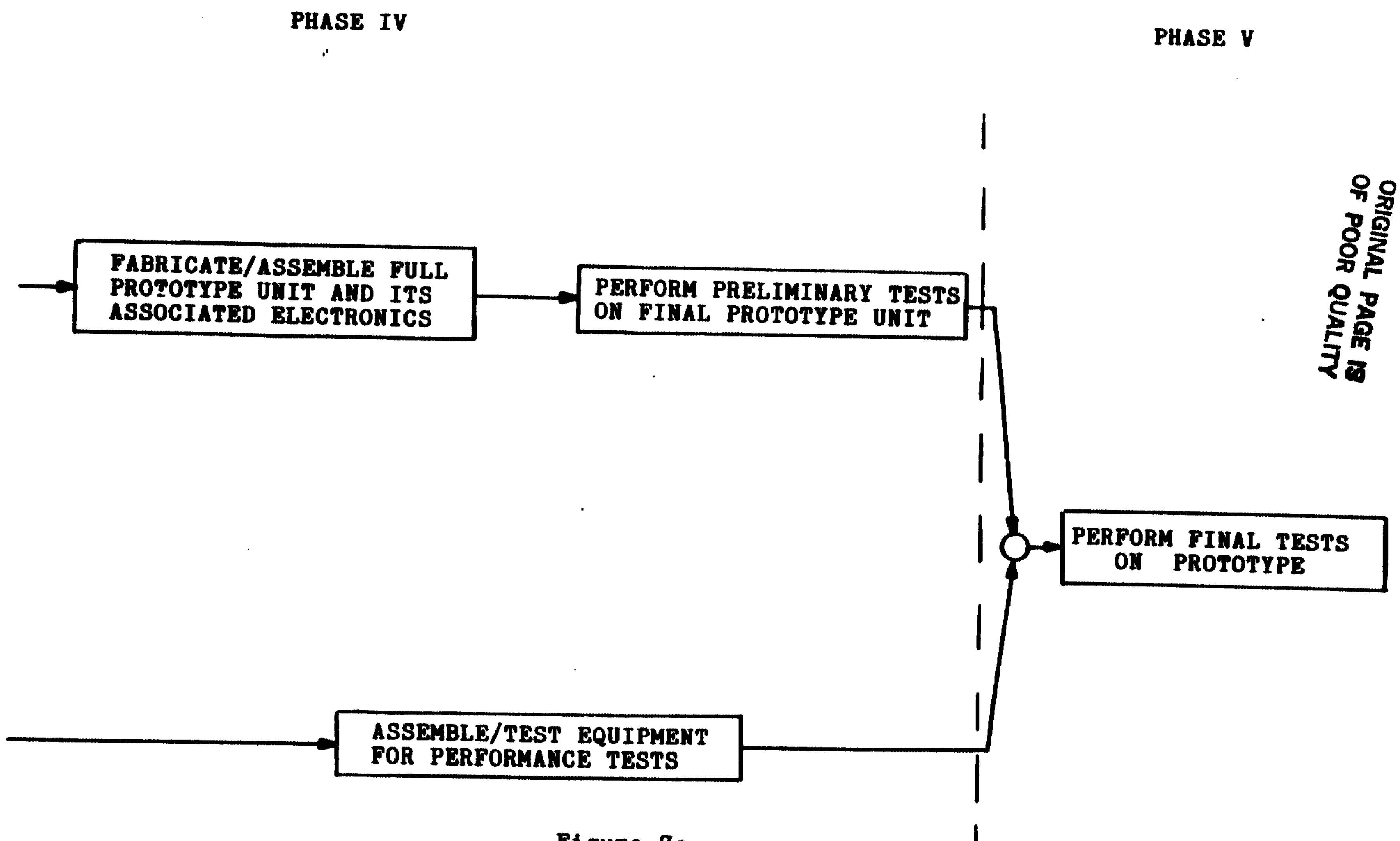
PHASE IV
PHASE V
FABRICATE/ASSEMBLE FULL PROTOTYPE UNIT AND ITS ASSOCIATED ELECTRONICS
PERFORM PRELIMINARY TESTS ON FINAL PROTOTYPE UNIT
ASSEMBLE/TEST EQUIPMENT FOR PERFORMANCE TESTS
PERFORM FINAL TESTS ON PROTOTYPE

Figure 7c

In practice the device for testing the in-situ performance of the flex washer design can also provide the room temperature test apparatus for examining the behavior of the servo-control system. Then, as soon as the position and velocity sensors are available, and we have some reasonable estimates for the moment of inertia of the system, we can construct an operating room temperature model and begin testing the servo-system. This will be extremely helpful since it will provide information about the control system at an early point in the program.

We've divided the program into essentially five separate phases which reflect the chronological order of the work. Discussions with NASA personnel indicate that a total developmental schedule of about 10 to 12 months should be consistent with the current schedule for the overall SIRTF system development. During Phase I we would try to address the most important design issues for the system, and begin constructing the test apparatus for evaluating the flex washer concept. We would also need to identify the required equipment and test fixturing needed to perform more complete system tests once the complete prototype has been constructed.

In Phase II most of the critical preliminary experiments would be performed on individual elements of the system. These experiments and measurements will insure that each the critical components of the system can meet their design specifications, and will provide performance information on each component to be used in developing the final prototype design in Phase III. The experimental work will include performance and reliability testing of the flex washer concept, initial testing on a superconducting actuator to characterize its electronic behavior, and testing and evaluation of different position and/or velocity sensors which might be suitable for our application.

We believe that it is important at the beginning of Phase III to develop a room temperature model to be used for preliminary tests on the questions of pointing and control of the mirror. To this end, one of the last tasks in Phase II is to develop a preliminary design for the servo-control system for the prototype, and construct a 4-element voice coil assembly which can be used to investigate the behavior of the two-axis chopping and control problems in a room temperature test device. The additional effort required for the room temperature experiments would be nominal, since the mechanical apparatus for these experiments would the same test fixture which had been previously constructed for testing the flex washers.

This is also a natural point for constructing a preliminary model of the servo-control system for the prototype. By addressing the electronic control system at an early date, using the results of experiments on the individual components, we would have the preliminary date from a brassboard servo-control unit to provide data for the more detailed prototype control system design to be done in the last part of Phase III.

While the room temperature test device would not have the full bandwidth capability of the more complete version, it will provide valuable information on the issues of pointing accuracy, and possible interaction between the two chopping axes. Furthermore, in terms of the development effort itself, it will be extremely difficult to work out all the problems in the control system using only a cryogenic model. We believe that the small additional effort required to set up a room temperature model of the two-axis actuator system will be repaid many times over in the convenience of solving problems in the servo-control system with a room temperature test model.

Using the information derived from the Phase II effort, then, we would expect to develop construct and test the room temperature model at the beginning of Phase III. Then, using the data obtained from these experiments, the last part of Phase III would be devoted to developing detailed design for the full operating prototype. At this point we should have demonstrated the performance of the superconducting actuators, the position and velocity sensors, the flexural joint, and the electronic control system. In addition, we should have obtained some preliminary information on the performance of the servo-control system.

Phase IV would be completely dedicated to the construction and preliminary testing of the full prototype unit. The preliminary tests should indicate the more obvious problems with the final design, and we expect that the any substantial engineering changes would also be carried out during Phase IV.

The final stage of the development program would then simply be devoted to testing and evaluating the full prototype unit. In addition to examining the pointing control and chopping performance of the model, we would also be investigating its thermal performance. Proper test instrumentation attached to the device could provide information on eddy current dissipation in both the mirror assembly and the voice coil magnets. We could also make some preliminary measurements of thermal drift in the mirror itself. This final phase of the development program would also include any engineering effort required to analyze and correct any deficiencies in finished prototype.

In this type of development effort, we would expect there to be a very close relationship between the contractor and NASA personnel who are familiar with the SIRTF program. In particular, NASA personnel have dealt for many years with the problems of developing space qualified hardware and launching complex, high-precision instrumentation into space. Furthermore, since the performance of the secondary mirror system is critical the the success of the project, the NASA personnel who are most familiar with SIRTF's scientific goals and requirements should remain in close contact with this effort.

To try to summarize our development program and the above discussion more concisely, we have itemized the various tasks as follows. These tasks, which are grouped into the four phases as discussed above, represent a summary of the development program. Figures 7a, 7b, and 7c show the approximate scheduling of the tasks.

Phase I Tasks:

1. Perform computer-aided modelling calculations to determine optimum shape for mirror which will produce the required rigidity and minimize its moment of inertia. This should include thermal and mechanical modelling of the mirror drive stem and voice coil mounting structure as well as more detailed estimates for dissipative losses in the structure.

2. Identify and purchase or construct appropriate position and velocity transducers with detection electronics which can be tested for possible use in the secondary mirror servo-control system.

3. Develop an initial design for a superconducting voice coil actuator and a simple drive electronics which can be used to characterize the actuator. This task must also include the design for a simple fixture suitable for testing the actuator in a cryogenic environment.

4. Develop a specific design for the flexural washer joint with consideration for possible materials to be used, its reliability and expected operating life, and the thermal conductivity the joint will provide to the mirror.

5. Design a simple test fixture suitable for testing the flex washer joints. The test station must be designed to allow testing of the pointing accuracy, positioning reproducibility during chopping, and stability of the rotation axis. Since this test station will also be used to test a room temperature model of the prototype to resolve problems in the servo-control system, its initial design should be compatible with this additional function.

6. Define the performance testing which will be required for the final prototype. It will be important to define the testing to be performed on the final prototype to allow sufficient lead time to obtain the necessary testing equipment and construct any fixturing which may be required.

Phase II Tasks:

7. Using data for the optimized mirror design from Task 1, develop final performance specifications for the actuators, position, and velocity sensors. These calculations

represent preliminary work which will culminate in the complete servo-control system to be developed in Phase III.

8. Purchase or construct those components required for the position and/or velocity sensors which were identified in Task 2, and construct any minor test fixtures which will be required to perform some preliminary tests of the actuators. We expect that these tests would be minimal. For example, we might wish to perform some brief preliminary experiments simply to insure that the sensors behave as expected in a cryogenic environment.

9. Construct and characterize the behavior in a cryogenic environment of the prototype superconducting actuator from Task 4. These tests will also help define the requirements for the actuator drive electronics for the final prototype.

10. Construct the apparatus from Task 5 to be used both for testing the flex washers and for testing the servo-control system with a room temperature model of the prototype.

11. Fabricate and test the flexural washer samples based on the designs developed in Task 4 using the test apparatus constructed in Task 10.

12. Design a set of room temperature actuator coils and a preliminary brassboard control system which will be suitable for operating a room temperature model of the two-axis chopping drive using the flex washer test fixture.

13. Identify the test equipment and fixturing which will be required for the prototype performance testing. Most of these requirements will depend on the exact testing to be done which was defined in Task 6.

Phase III Tasks:

14. Identify the most promising design for the flexural joint, and fabricate a flexural joint which is suitable for use in the room temperature model.

15. Construct the room temperature model of the two-axis drive system and the preliminary brassboard control system designed in Task 12.

16. Perform experiments and measurements on the room temperature model constructed in Task 15 to determine the ability of the system to meet pointing stability specifications and to identify possible problems in the implementation of the two-axis drive system.

17. Using data obtained from measurements on the mirror, the actuators, the position and velocity sensors, and the room

temperature model, design the complete operating prototype of the secondary mirror control system.

18. Begin the purchase and/or fabrication of the test apparatus which will be required to adequately characterize the system in the final sequence of performance tests.

Phase IV Tasks:

19. Fabricate the entire operating prototype for the secondary mirror control system, including the final design for the electronic servo-control system.

20. Perform preliminary testing of the finished prototype to identify any obvious design problems, and modify the system as required to correct the problems.

Phase V Tasks:

21. Perform the entire sequence of testing which has been previously identified to verify the performance of the final prototype unit. These tests should be designed to fully characterize the mechanical, electronic, and thermal behavior of the system, and we expect that the measurements will require that some reasonable instrumentation be attached to the mirror to make the necessary measurements.

In the above task breakdown we have tried to address in fairly general terms all of the work which will have to be done to develop the desired system. We have not assumed any particular assignment of the different tasks, only the general order in which they must be accomplished, since some parts of the work must naturally be done first. However, some of the tasks require very specific skills or capabilities, such as the optimization of the mirror design in Task 1, for example. In managing this developmental project, it may be desirable in some special cases to assign a single task or group of tasks to an individual or group which has very specialized knowledge about the specific problem to be solved.

XI. CONCLUSION

To conclude this final report we would like to quickly summarize our most important conclusions regarding the overall program. First, we feel those issues discussed in Section II to be the most important problems which must be addressed by the secondary mirror design.

1. Reliability and simplicity.

2. Thermal dissipation in the mirror. This issue is graphically demonstrated in Section VIII.

3. Distortion in the mirror, both during cooldown and as a result of the chopping accelerations.

4. Total thermal dissipation at the secondary mount. Even if there is no electronic dissipation at the secondary mount, the calculations of Section VIII make us cautious about deemphasizing the importance of this issue.

5. Thermal Management. Again, the calculations of Section VIII graphically demonstrate the vital importance of proper thermal design in the system.

6. Overall compatibility with a cryogenic environment. This is closely related to that of reliability. The system simply must be able to function reliably at liquid helium temperature, and must be able to withstand repeated thermal cycling between room and cryogenic temperatures.

7. Reaction mass. The pointing stability requirements for the telescope virtually demand a reactionless system at the secondary mount. The system must provide that feature in some reasonable manner.

In Section III we discuss at some length the issue which we believe is a corollary to the above requirements - that the secondary mirror system requires that the absolute minimum moment of inertia be rotated at the secondary mirror. This view is further supported by our estimates in Sections VII and VIII which identify some of the major problems that will have to be faced in meeting SIRTF specifications. In particular, we are convinced that the stringent chopping specifications can be met only by a system which first minimizes the total mass and moment which must be moved, then incorporates the most efficient possible actuators to produce the required torques.

The requirement for minimizing the moment of inertia also means that we will want to use a material for the mirror which has the greatest possible strength to weight ratio. However, the final decision on this will probably rest with those who do the detailed calculations on the mirror design.

The actuators may also have a significant impact on the total moment of inertia of the system, and these are exactly the points we addressed in Section IV in our discussion of actuators. In brief:

1. We believe that moving coil actuators are the most natural selection for the secondary mirror drive system.

2. By using superconducting voice coils and electrically insulating materials for the coil carriages, dissipation in the mirror assembly can be nearly eliminated.

3. Any actuator in which any dissipation occurs must be kept away from the mirror and mounted such that its dissipative elements do not produce thermal dissipation in the mirror assembly itself.

4. The actuators themselves must not contribute excessively to the moment of inertia of the system. This is consistent with our concern for minimizing the torques and energy which are required to drive the system.

The importance of minimizing the total energy which must be delivered to the system is further emphasized (if indeed further emphasis is needed) by the somewhat disturbing calculations in Section VIII, regarding the difficult problem posed by the temperature stability requirement for the mirror in light of its tiny heat capacity at 7 Kelvin. It is not at all clear that this specification can be met when we consider that the energy required simply to move the mirror as required must be a million times greater than the allowable fluctuations in its enthalpy.

In considering the full range of issues which must be addressed in the secondary mirror design, we've been led to the design concept described in detail in Section V. While the design certainly leaves open the question of its ability to meet all of the SIRTF requirements, we believe that the discussion in Section VI demonstrates that it does indeed address nearly all of the most critical issues in a reasonable manner. The most significant unresolved questions are described in Section IX.

In Section X we have tried to outline a program for developing a complete laboratory model for the system which can be used to investigate some of the critical issues surrounding the secondary mirror, and provide some firm experimental data upon which further decisions can be made. Since the secondary mirror system is one of the most vital subsystems of SIRTF, it is important that these questions be resolved in a timely manner. That is the primary goal of our proposed program.

Finally, it should be clear from the results of this study that the successful resolution of the entire set of cryogenic and thermal design problems posed by the SIRTF secondary mirror system as currently specified will require the utmost care in both the design and prototype development programs. As we pointed out at the end of Section VIII, some of the thermal design aspects of the system will have to consider problems which are normally only encountered at temperatures two or more orders of magnitude below the operating temperature of the secondary mirror. Consequently, we expect that a variety of experimental measurements will be required during the prototype development to optimize the system, and insure that the final prototype unit will have a reasonable probability of being suitable for the SIRTF telescope. The effort will certainly have to be undertaken by people who have had extensive experience in the techniques and problems encountered in developing low temperature apparatus.

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