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PRELIMINARY STUDY OF THE LIQUID METAL LOOP
AND TEST RIG FOR PHASE II OF THE INVESTI-
GATION OF LIQUID METAL LUBRICATED BEARINGS
AND ROTOR-BEARING DYNAMICS

by

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Date Feb. 23, 1965

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I, INTRODUCTION

This report describes the results of a study that was made of the mercury loop and the rotor-bearing test rig that were shipped to M.T.I. for use in a liquid metal lubricated bearing and rotor-dynamics program sponsored by the Division of Reactor Development and Technology of U.S. AEC. This equipment had previously been developed and used by AiResearch on a program conducted to provide liquid mercury lubricated bearing data for SNAP 2.

This equipment was transferred to M.T.I. specifically for use in connection with Phase II of the liquid metal lubricated bearing and Rotor-Bearing Dynamics Program, currently in progress (Refs. 1 & 2). Work at M.T.I. on the first part of Phase II was subsequently authorized by the U.S. AEC. This part, denoted Phase IIA, was to comprise:

1. Installation of the liquid metal loop and check out of its components.
2. Prove out of the rotor-bearing test rig. This was to determine whether the operational problems which were encountered with the rig at AiResearch and which greatly limited the test data obtained there, could be satisfactorily corrected. Following this, determination would be made whether the existing rotor-bearing test rig could be used in the liquid-metal bearing investigation at M.T.I., or whether a new test rig would need to be designed and fabricated for use with the existing liquid metal loop.

It was further requested by the U.S. AEC that the Phase II test program at M.T.I. be conducted with an alkali metal instead of mercury. Assurance that the loop was operable with an alkali metal was therefore required before its installation at M.T.I. The selection of the recommended alkali metal for the test program was to be made by M.T.I. during the preliminary investigation.

Accordingly, the preliminary study reported here was made, with the following objectives:

1. Review the liquid metal bearing and rotor dynamics program in light of the change to alkali metals.
2. Assess the feasibility of using the existing AiResearch built, single bearing test rig in the alkali metal test program. Review necessary modifications to the test rig or design an alternative unit, if the existing rig is not satisfactory for the program.
3. Determine extent and cost of modifications of the present AiResearch built mercury loop, required for its conversion into an alkali metal loop.
4. Recommend the test fluid.

II. CONCLUSIONS AND RECOMMENDATIONS

The principal conclusions of the study are:

1. Analytical study of the present, AiResearch built, single bearing rotor test rig showed it to have serious problems of rotor bearing dynamics, so that this test rig should not be further used in the current program. These problems include:
 - a. Two shaft criticals, including the free-free critical, occur within the coastdown speed range. These make it extremely difficult to avoid conical whirl or to separate out rotor dynamics effects when obtaining bearing test data.
 - b. Uncertainty exists regarding the location of the flexural critical speed due to the fact that the rotor is built up, with two lightly shrunk on sleeves, whose contribution to the stiffness (section modulus) of the shaft has to be ascertained by special test and, which may not be reproducible.
 - c. Significant errors in rotor displacement measurements are introduced because of shaft bending between the bearing and the location of the measurement probes.
 - d. At least in vertical operation, the unit is limited to the use of a 360° plain circular bearing, since bearing loads are not directly measurable, but have to be calculated by translatory or conical whirl theory. At present, theory and calculation procedures are available only for translatory whirl of the 360° bearing, neglecting film rupture effects.
 - e. When used with alkali metals, only limited penetration of the developed turbulence regime can be achieved, due to the small bearing size (1" diameter).

2. A preferred, two journal bearing test rig was reviewed and is shown schematically in this report. Such a rig can be accommodated in the dry box of the present loop and it is recommended because of -

- (a) Freedom from Conical Whirl
- (b) Adaptability, with minimum changes to all parts of the Phase II test program (i.e. steady state load tests, dynamic load tests and flexible shaft, rotor-bearing dynamics tests).

The journal bearing size will be between 1-3/4" and 2-1/2" to be established during the final design. This will permit:

- (a) ease of instrumentation
- (b) operation in Alkali metals from laminar up to at least 5 or 6 times the value of the Reynolds Number at which developed turbulence is induced in the film.

3. Several components of the current mercury loop can be utilized in the alkali metal loop, including: the vacuum system, the dry box, the present instrumentation and some of the loop piping. The components that will have to be replaced for operation in alkali metal were also listed. These include: the pump, the flowmeters, the drain sump and interconnecting piping. A schematic of the recommended alkali metal loop was given.

4. The preferred test fluid is NaK 78, since:

- (a) its properties are intermediate between sodium and potassium and it will therefore provide representative data on alkali metal lubrication.
- (b) it can be used at room-temperature, thus simplifying the test loop and the test program.
- (c) no evidence could be found to indicate any significant change in its properties with time due to the fact that it is a two constituent fluid.

Liquid mercury could continue to be used and would require no major loop modifications. However, some of its properties are extreme, compared to alkali metals, and this could influence bearing performance, tending to indicate problem areas that would be different from those encountered with alkali metals.

It is therefore recommended that Phase II of the program be continued as follows:

1. Install Nak Loop shown schematically in this report and prove out the components, including the useable components of the present mercury loop.
2. Design, fabricate and prove out the two bearing test rig also shown schematically in the report.

The change-over to NaK and the requirement for a new test rig will increase the level of funding required for the procurement and installation of the test equipment, however this is partly balanced by the reduced cost of the later work, since the new test rig can be used for both the bearing tests and the rotor-bearing dynamic tests.

III. DISCUSSION

1. Review of the Phase II Objectives

Before proceeding with the discussion of the study recently completed, it is helpful to review the objectives of Phase II of the liquid metal lubricated bearing and rotor-bearing dynamics program.

The concept of Process Fluid Lubrication has been adopted for liquid metal, Rankine Cycle turbomachinery for dynamic power conversion as well as for liquid metal pumps, circulators and other equipment. This has been done for improved reliability of design, by eliminating a separate oil lube system and greatly reducing the complexity of the seals required in the system. The use of liquid metals as lubricants, however poses some new problems of bearing operation including:

1. Low kinematic viscosity and, hence, susceptibility to turbulence in the fluid film.
2. High shear losses associated with turbulent bearings and, consequent thermal distortion.
3. Higher attitude angles associated with turbulent operation and reduced damping due to the low viscosity of the lubricants, result in greater susceptibility to rotor-bearings instabilities. (Operation in zero G field in space vehicles aggravates this problem further).
4. Materials compatibility.
5. Lack of adequate film thickness instrumentation for liquid metal environment.

In connection with other work (Ref.6) a comprehensive turbulent lubrication theory has been prepared and its validity for engineering design purposes has been checked by comparison with both steady state and dynamic load test data obtained with easy to use, low kinematic viscosity lubricants (0.65 c.s. silicone oil and water).

This theory can now be applied to rational design of turbulent flow bearings, however verification of the theory with actual liquid metal lubricants is still needed.

The problem of instability can be attacked, by selection of bearing designs that are inherently stable, or that have a very high threshold of instability. The problem of high power loss and consequent thermal distortions associated with turbulent bearings is also a serious one, that needs to be solved in order to realize the advantages of process fluid lubrication in the larger C.R.U.'s.

The approach selected at M.T.I. is to design for operation above the flexural critical speeds of the C.R.U. rotor, whereas all previous, liquid metal lubricated rotors have been designed to operate below the flexural critical speed. By designing for operation above the flexural critical speed, journal size can be greatly reduced, with consequent reduction in bearing power loss, cooling requirements and thermal distortion problems. The journal bearings for such systems would also have to be designed to be either inherently stable or to possess very high instability thresholds.

Therefore, the major objectives of the current program include:

1. Verification of turbulent bearing theory with liquid metal lubricants.
2. Operation of C.R.U. simulator rotors with turbulent bearings well above the flexural critical speed range and into the "Resonant Whip" operating range (see footnote on Page 18).
3. Comparison of the measured performance data with that predicted by:
(a) turbulent bearing theory, and (b) rotor-bearing dynamics analysis in turbulent bearing regime.
4. Generation of design data to permit future, rational C.R.U. design with liquid metal bearings in both the rigid and the flexible rotor ranges. (Since all rotors possess at least some degree of flexibility, the rigid rotor case is the limiting case, where the bending deflections of the rotor are very small compared with the deflections at the bearings).

In Phase I of the program, which is currently in progress, Items 2, 3 and 4 above are being conducted but utilizing easy to use, low kinematic viscosity fluids.

In Phase II of the program all above items are to be conducted using an alkali metal as the system fluid. The experimental work here will be conducted in three steps as follows:

1. Obtain steady state load data with the liquid metal lubricated bearings operating in the turbulent regime. A rigid rotor will be used in these tests, to prevent introduction of rotor-flexibility effects.
2. Obtain dynamic load data, as above to determine also turbulent, liquid-metal bearing stiffness and damping characteristics.
3. Obtain rotor-bearing dynamic response data with turbulent, liquid-metal lubricated bearings and with a flexible rotor, so that the operating range spans the flexural, system critical speeds, as well as the "Resonant Whip" region.

2. Evaluation of the Existing Rotor Bearing Test Rig

(a) Background

The existing test rig, which was designed and fabricated by AiResearch, is shown schematically in Figure 1. The rig, which has a single 1" diameter journal bearing may be operated vertically, horizontally or at intermediate inclinations (although, to the best of the writer's knowledge, the test work conducted at AiResearch was done entirely with the rig mounted vertically). Shrunk on sleeves were used at the bearing section and in the lower end of the stem shaft, presumably in order to permit changing the journal sleeve, in event of failure of the bearing and consequent damage to the journal sleeve.

The rig was designed for operation at up to 40,000 RPM and it was run at AiResearch, at up to about 30,000 RPM. It is driven by an integral, axial gap motor, especially built by AiResearch for this test rig. The rotor part of the motor is at the top of the "hat" section of the test rig rotor. The intended test procedure was to bring the unit up to a desired speed, then to shut off the power supply and to gather all test data during the coastdown period. The significant data was to include: speed of rotation, rotor displacement (measured at the O.D. of the "hat" section of the rotor), fluid film pressures and temperatures, friction torque and lubricant flow. During vertical operation, the forces imposed on the bearings were those due to residual unbalance and to known, imposed unbalance loads. Care was exercised to insure that the mass center coincided with the bearing center. Presumably, therefore, translatory whirl orbits would be obtained if the unbalance loads were located in phase and symmetrically about the bearing center. Alternatively, conical whirl orbits would be obtained by locating the unbalance weights out of phase with one another, or eccentricially about the bearing center.

The thrust load (i.e. the rotor weight during vertical operation) was to be carried by a jewel bearing located below the stem shaft. Operational problems with the jewel bearing necessitated a change, first to an externally pressurized (hydrostatic) gas bearing and, later, to a hydrostatic mercury bearing.

* This rotor was burnt out, prior to its receipt at M.T.I.

In principle, the test rig had some important, potential advantages, including:

1. "Pure" bearing data could be obtained since the system had only one test bearing and there were no loader bearings, support bearings or seals. Also, since the test data would be gathered on coast down, there would be no extraneous influences caused by the motor.
2. Data over the complete speed range then of interest, (0 to 40,000 RPM) could be obtained rapidly during coast down.

In order to realize these advantages, much effort was spent in developing special instrumentation such as the film pressure transducers. The coast down test data were recorded on a high speed magnetic tape recorder, for reduction at the conclusion of one or more test runs.

In practice, however, several operation problems were encountered (some of which had been predicted by M.T.I. during early reviews of the AiResearch program, as noted in Ref. 2), which prevented the gathering of accurate, fundamental bearing data. In summary, the major shortcomings of the tests included:

1. Rotor misalignment was not eliminated. The orbits obtained in coast down were conical instead of translatory.
2. There was considerable question regarding the attitude angles, again probably due to the misalignment.
3. The friction measurements were influenced by parasitic losses (with the hydrostatic gas bearing, in particular, it was not possible to evacuate the dry box to a low enough pressure to reduce windage losses, because of the flow of pressurized gas to the thrust bearing. The hydrostatic mercury bearing reduced this problem however, it introduced a significant additional friction torque).
4. The film pressure transducers were subject to shorting out. At times in the tests, only two out of the eight transducers were operating.

Efforts were made to correct this through design changes, but a full prove out of the pressure transducers was not achieved.

5. During part of the tests, the centrifugal pump had to be replaced with a piston pump. The latter produced pulsations in the flow and, hence dynamic loads of unknown magnitude.

Of these, the most serious problem is that shaft misalignment and conical whirl, which is characteristic of single journal bearing test rigs. Journal bearing characteristics (load capacity, attitude angle, stiffness, fluid film pressures and others) are very sensitive to alignment. While misalignment can be introduced into bearing calculations, it requires two additional parameters. The task of test data reduction and meaningful interpretation (either for comparison with theory or to produce general purpose design data) is so greatly complicated in the case of misalignment that it is impractical.

Thus, it was important to determine, prior to attempting further experimental work with the existing test rig:

1. the cause of the misalignment and conical whirl,
2. whether it could be eliminated by design changes, such as changes in the polar and transverse moments of inertia of the rotor.

The analysis that is discussed here was made principally for these purposes.

(b) Analysis

A comprehensive analysis of the rotor-vibrational characteristics was made to explore the rotor-bearing dynamics of the test rig and, in particular, the conical whirl phenomenon.

2.1 Critical Speeds Map

The critical speeds of the rotor shown in Figure 1 were calculated as functions of journal bearing stiffness, utilizing the existing M.T.I. computer program which uses the conventional Holzer-Prohl Method (Ref.3).

As previously noted, the test rig rotor has two shrunk on sleeves, one at the bearing location and the second at the lower end of the shaft. These sleeves are held to the basic shaft by a light interference fit (0.0001" to 0.0003" shrink), on very short pilot diameters. Their contribution to the shaft stiffness (and, hence, to its critical speeds) are unpredictable and non reproducible. Therefore, the calculations were made for three different ways in which the sleeves may contribute to the shaft stiffness, including the two limiting ones. These are as follows:

<u>Configuration Number</u>	<u>Effect of Sleeves on Shaft Stiffness</u>
1	Both sleeves contribute to shaft stiffness over their full lengths.
2	Only the short, upper sleeve (at the bearing location) contributes to the shaft stiffness.
3	Neither sleeve contributes to the shaft stiffness.

The calculation results are shown in Figure 2. The first critical speed is the lower curve in this figure. As would be expected it is independent of whether or not the sleeves contribute to shaft stiffness, in the lower range of bearing stiffness, (up to about 10^4 lbs/inch). This constitutes the rigid body critical, where the shaft acts essentially as a rigid mass supported on a fluid film spring and dashpot. Above a bearing stiffness of 10^4 lbs/inch, the shaft flexibility starts to affect the first critical speed.

The second critical speed (upper three curves of Figure 2) is sometimes called the "free-free" mode*. In the lower range of bearing stiffnesses, it depends only on the shaft stiffness. The contribution of the shrunk on sleeves is therefore very important here, so that the "free-free" critical can occur anywhere between about 7600 RPM and 27000 RPM (at low values of bearing stiffness), depending on how the sleeves contribute to shaft stiffness. In any case, it is important to note that two critical speeds, including a shaft flexural mode, will always be encountered in coast down tests from 30,000 RPM or higher. The rotor stiffness will therefore

* The second critical speed is the free-free mode in this case, since there is only one radial bearing. In the case of a two radial-bearing system, the third critical speed would be the free-free mode.

influence the measurements, so that the bearing data obtained directly, will be peculiar to this rotor-bearing system. In order to obtain general purpose data, it will be necessary to subtract out the shaft flexural influences. This is very difficult to do and it will greatly reduce the level of accuracy of the bearing data.

The mode shape at the two critical speeds are shown in Figure 3 for two extreme cases (1450 rpm at a bearing stiffness of 1000 lbs/inch and 28,379 rpm at a bearing stiffness of 10,000 lbs/in.) It can be seen that both mode shapes are essentially conical and that the flexural critical speed is an oscillation about the mass center of the shaft.

2.2. Rotor Response to Unbalance Loads

The response of the rotor-bearing system to unbalance loads was calculated for the following two conditions:

1. Speed = 30,000 rpm
Bearing radial stiffness: 5000 lbs/inch
Bearing angular stiffness: 416 in. lbs/radian
Unbalance loads: 0.015 oz.in at each of the
two balance planes; in phase
Rotor configuration 1, i.e., with both sleeves
contributing to rotor stiffness.
2. Same as above, but with rotor configuration 3, i.e.,
with neither sleeve contributing to rotor stiffness.

The calculated mode shapes for the two cases are shown in Figure 4. The projections of the shaft axes on two orthogonal planes (sine and cosine components of the vibration amplitudes) are shown for each case in Figure 4. The two lines represent in each case, the axes of the "stem" and "hat" sections of the shaft. The large amplitudes of the shaft, calculated for the case where the sleeves contribute to bearing stiffness, are due to the fact that the 2nd critical speed for this configuration occurs at 28,000 rpm (see Figure 2) i.e., close to the speed used in the calculations (30,000 rpm). In the second case, the amplitudes are smaller since the 2nd critical speed for this case occurs at about 8000 rpm (see Figure 2) i.e., much below the speed at which the calculations were made.

The significance of Figure 4 lies in the insight it provides into the various vibration amplitudes and mode shapes that are encountered by the present rotor on coastdown. Since, as shown by Figure 2, the rotor will traverse two critical speeds on coastdown from 30,000 RPM or higher speeds (irrespective of the contributions of the shrunk-on sleeves to the rotor stiffness) the vibration amplitudes will initially be fairly small and nearly translatory (as shown by the mode shapes in Figure 4 for the low section modulus case). Then, as the second critical speed is approached, the amplitudes will increase in magnitude and the mode will be largely conical (as shown by the mode shapes in Figure 4 for the high section modulus case). The amplitudes will next decrease, as the shaft coasts down between critical speeds, and increase once again as the first critical speed is encountered. Again, therefore, it is seen that the data obtained with the present rig will be a function of the rotor characteristics, rather than being general purpose bearing design data.

2.3 Rotor Displacement Measurements

The rotor displacement measurements are made at the O.D. of the "hat" section. When these measurements are related to the motion of the journal center in the bearing, there will be an error (that varies with the speed and magnitude of unbalance load) due to the bending of the shaft. As shown in Figure 4, this error is reflected by the angle between the axis of the "stem" shaft where the bearing is located, and the axis of the "hat" section. For the calculation conditions of Figure 4, the difference between the vibration amplitudes in the journal and at the measurement points in the hat section, are of the order of 10% to 30% of the journal displacement. Additional error is introduced due to the bending of the disc section which carries the motor laminations.

2.4 Calculated Bearing Loads

When the rig is run vertically, there is no unidirectional, steady-state load on the journal bearing. The journal bearing is loaded by the introduction of mechanical unbalances of known magnitudes in the two balancing planes. When the speed of rotation is below the first critical speed, the unbalance introduces

a bearing force equal to the centrifugal force corresponding to the mechanical unbalance (in oz-inches) and the speed of rotation. Above the inversion point that occurs at the first critical speed, however, rotation will be about the mass center, instead of the geometrical center. The rotating load in the bearing must then be calculated using translatory or conical whirl theory. While this can be done with presently available theory for the case of a plain (360°) circular bearing assuming that film rupture can be prevented by adequate pressurization of the lubricant, (see e.g. Ref. 4), such theory is not yet available for other bearing geometries. Thus, the current test rig in vertical operation is presently limited to the use of 360° plain circular bearings, unless considerable additional effort is expended on analysis and preparation of computer programs for treating large whirl loci with other bearing types.

From an experimental standpoint, the 360° plain circular bearing is limited to operation of high eccentricity ratios (say about 0.5 or higher) since below that it is highly susceptible to half frequency whirl, which makes it virtually impossible to obtain accurate steady state test data. Good steady state data at low values of eccentricity ratio are, perhaps best obtained by using pivoted pads, since these can be designed to be inherently stable at all values of bearing eccentricity ratio. The envelope dimensions of the bearing housing of the present rig, however, will not accommodate a tilting pad bearing.

5. Penetration of Turbulent Flow Regime

The current test rig has a one inch diameter journal bearing. If mercury were used as the lubricant, and allowing clearance ratios as high as 2.5×10^{-3} inches/inch, it would be possible to achieve values of Reynolds Number as high as 15,000 at a speed of 40,000 RPM. If, however, operation were with alkali metals (in this case, Na, K or NaK), the maximum Reynolds Number would be only about 2500, or less than twice the value required to attain developed turbulence in the fluid film. Thus, only limited penetration in turbulent fluid film region is achievable with a 1" diameter bearing, with alkali metal lubricants. This is not felt to be sufficient when seeking turbulent bearing design data.

In summary, therefore, it is recommended that the present rotor-bearing rig should not be further used to obtain liquid metal lubricated bearing data, for the following reasons:

1. Two shaft criticals, including the free-free critical, occur within the coastdown speed range. These make it extremely difficult to avoid conical whirl or to separate out rotor effects when obtaining test data.
2. Uncertainty exists regarding the location of the flexural critical speed due to the fact that the rotor is built up, with two lightly shrunk on sleeves, whose contribution to the stiffness (section modulus) of the shaft has to be ascertained by special test and, which may not be reproducible.
3. Significant errors in rotor displacement measurements are introduced because of shaft bending between the bearing and the location of the measurement probes.
4. At least in vertical operation, the unit is limited to the use of a 360° plain circular bearing, since bearing loads are not directly measurable, but have to be calculated by translatory or conical whirl theory. At present, theory and calculation procedures are available only for translatory whirl of the 360° bearing, neglecting film rupture effects.

The above limitations are independent of the liquid metal used as lubricant. In addition, if alkali metals are used instead of mercury, there is the additional limitation that operation is possible only up to at most twice the value of Reynolds Number required to insure developed turbulence. This is due to the small (1" diameter) size of the bearing.

3. Recommended Rotor Bearing Rig

To accomplish the liquid metal lubricated bearing test objectives, the test rig must be applicable to generation of accurate steady state and dynamic test data in turbulent regime, without unpredictable, external influences. Specifically, the test rig must satisfy, at least, the following requirements.

1. Both steady state and dynamic loads should be applicable to the bearings. The steady state loads should be directly measurable. The dynamic loads should be readily obtainable from rotor bearing response calculations.
2. The test rig should be capable of accommodating, in sequence, first a rigid shaft for bearing performance measurement, then a flexible shaft for rotor-bearing dynamic response measurements.
3. The rig should be capable of accommodating different bearing geometries.
4. The rig should be operable over a wide speed range that should extend from laminar conditions up to, at least, a bearing Reynolds Number of about 7500 (i.e. about 5 times the Reynolds Number required for developed turbulence).
5. Lubricant supply pressure and flow should be adequate to keep the temperature rise across the bearing low, say, not to exceed 20 F at maximum speed.
6. The instrumentation should be adequate to measure, as a minimum: load, speed, rotor displacement, lubricant temperatures, vibration amplitudes and phase angles (and, hence, bearing stiffness and damping characteristics), over the load and speed ranges of the tests.

A two bearing, horizontal test rig is recommended for this, as shown schematically in Figure 5. It is electric motor driven through a quill shaft and loaded by means of a loader bearing located midway between test bearings. A test rotor supported in two bearings is preferred because it can readily be designed to operate aligned and without conical whirl.

The test sequence will be divided into three parts, as follows:

1. Steady State Bearing Tests

Here, only steady state loading will be used and the shaft will be rigid (i.e. its first flexural critical will be at least 50% above the maximum running speed). These tests will establish the steady state characteristics of the liquid metal lubricated bearings i.e. load, eccentricity-ratio and attitude-angle relations.

2. Dynamic Load Bearing Tests

Here combinations of steady state and dynamic loads will be applied. Again a rigid shaft will be used. These will determine the dynamic characteristics of the liquid metal lubricated bearings (i.e. fluid film stiffness and damping) from measurements of vibration amplitudes, phase angles and whirl orbits, all as functions of bearing and lubricant properties and operating eccentricity ratios.

3. Dynamic Rotor-Bearing Response Tests

Here, the steady state load will be the rotor weight and tests will be conducted at different values of the dynamic load. A flexible shaft will be used and the test range will span the flexural critical of the shaft and will be carried into the "Resonant Whip"* region. The data from this test will provide dynamic rotor-bearing design data in the regions where the shaft is operating at or above its flexural critical speeds, as well as in the "Resonant Whip" range.

* "Resonant Whip" (sometimes called "Resonant Fluid Film Whirl") is a resonant vibration of a journal in a fluid film bearing which occurs at approximately twice the first (flexural) critical speed and persists to higher speeds. The frequency of vibration is approximately equal to the first critical speed of the rotor-bearing system. Unless stable operation can be insured beyond this threshold, it limits the degree of shaft flexibility that can be used in the design. Since a major problem associated with turbulent bearings is high power loss and consequent thermal distortions, it is a principal objective of this program to demonstrate feasibility of operation at above twice the first, flexural system critical speed and gather design data in this range. This will permit future design of flexible shaft, liquid metal lubricated C.R.U.'s with reduced problems of high power loss and thermal distortions.

The first two sets of tests will be conducted using two pivoted pads in each test bearing. Pivoted pads were selected because:

1. They are inherently self aligning (if spherically pivoted) and will follow all shaft motions. Therefore, they satisfy one of the basic assumptions of the lubrication analysis, i.e. the eccentricity ratio is constant over the length of the bearing.
2. The tilting pad bearing has a zero attitude angle which gives rise to its inherent freedom from half frequency whirl and, if the load is directed midway between pivots, bearing stiffness can be measured directly from the load-displacement measurements. Because of its inherent stability, meaningful tests at low eccentricity ratios can be conducted.
3. The partial arc (a tilting pad is a pivoted partial arc where the direction of steady state load is exactly defined, since it has to pass through the pivot). is a simple basic form of a hydrodynamic bearing and readily lends itself to theoretical analysis.

The test bearings will be mounted into housings containing clearance seals and drain passages at each end. The seals are required to maintain the bearing cavity flooded at all times and to keep the test fluid (NaK) from contaminating to any large degree the environment outside of the bearing housing. Two proximity transducers mounted 90° apart will be located on the outside of the housing to measure shaft to housing displacement with varying loads and speeds. Additional proximity transducers will be mounted inside the housing (conceptually shown in Figure 6) to measure pad to housing displacement. The measuring surface will be under a "trapped" cover gas to keep the liquid metal from "shorting out" the transducer. Use of the proximity transducers located in two places will give data on both: (a) shaft to housing and, (b) shaft to pad, displacements. Thermocouples will be located within the housing to measure pad lubricant inlet and outlet temperatures. Either pressure or level instrumentation will be employed on the housing to insure that the bearing cavity is full of fluid at all times.

The load is applied to the shaft through a grease packed shielded self-aligning rolling element bearing. Load is provided by means of a pneumatic cylinder pressurized with either argon or nitrogen. The loading system will be designed with a low enough spring constant so that it will not influence shaft orbits resulting from applied unbalance. Load is to be measured by calibrated strain gages located within the loading system.

The shaft will be driven by a variable input frequency motor through a splined quill shaft designed to impose minimum radial or moment type restraint on the test shaft. The shaft size should be in the order of 1.75 to about 2.5 inches in diameter to approximate the size of future C.R.U.'s. The exact shaft size will be determined after a more thorough design analysis and will be based on such factors as:

1. leakage flow
 2. steady state load required to achieve the desired range of eccentricity ratios, up to say about 0.7, and its effect on the structural rigidity of the system.
 3. ease of instrumentation
 4. lubricant supply pressure and flow requirements,
- and others.

The test rig will be operated within the dry box under a cover gas of either argon or nitrogen maintained at atmospheric pressure. Because of space limitations within the dry box, the drive motor will be located outside, with the shaft entering the dry box through a labyrinth seal.

The only modifications necessary for the third series of tests will be

1. removal of the static loading system
2. replacing the rigid shaft with one of flexible design with distributed masses to simulate turbomachinery C.R.U.'s
3. the addition of two upper pads to each test bearing to make them complete and capable of sustaining loads in all directions.

4. Modification of the Existing Mercury Loop, for its Conversion to Alkali Metals

(a) Utilization of Present Loop with Mercury

The mercury test loop was received at M.T.I. and inventoried, as noted in the Appendix to this report. While some parts of the loop were damaged either prior to or during shipment (see Appendix), the loop is useable. It can be re-assembled and de-contaminated for future use as a mercury loop, substantially, in accordance with our previous proposal (Ref. 2) which was prepared after notification that the loop would be shipped to M.T.I., but prior to its receipt here.

The major items of instrumentation, read out and recording equipment that were furnished with the loop and test rig are also listed in the Appendix. These are not sufficient for the conduct of the test program, since much of the measurement, read out and recording equipment that were used in the previous tests were not part of the AEC owned equipment that was shipped with the loop and test rig.

(b) Conversion to Alkali Metals

In accordance with requested change in the program from mercury to alkali metal lubrication, the present loop components were reviewed to determine the necessary modifications.

Many of the current mercury loop components can continue to be used with alkali metals. These include: the complete vacuum system, the dry box, the present instrumentation, read out and recording equipment, as well as some of the inter-connecting piping.

The components that are not suitable for operation with the alkali metals that have been considered (sodium, potassium and NaK), are listed below, together, with the reasons for recommending a change of these components:

1. The loop pump (Chempump) is a canned motor centrifugal type employing Teflon sleeve bearings. The probable reaction between potassium oxide and Teflon makes the bearing undesirable. The bearings can be replaced

with plane cylindrical bearings by the manufacturer. However, used with an alkali metal, the pump is limited to about 3.5 G.P.M. at 30 psi developed head (as opposed to 450 psi with mercury) and the inlet head requirements are increased from 3 ft. to 4 ft. necessitating lowering the relative position of the pump.

2. The flowmeters are turbine type with the rotating member mounted on carbon bearings. The largest flowmeter is limited to approximately 0.3 GPM whereas test flows, approaching 5 GPM per bearing will be required for alkali metal bearings. Furthermore, the use of carbon as a bearing material in an alkali metal is questionable because of the mass transfer and reaction problems in the critical bearing area.
3. The plumbing and drain sump are not inherently self draining, which is not a serious problem with mercury. However, when using the highly reactive alkali metals it is imperative that no pockets of liquid metal remain in the loop after draining.
4. The plumbing is primarily of 400 series stainless steel which, when welded in the absence of a preheat and subsequent stress relief, results in a very brittle weld joint. Since an alkali metal loop should be of welded construction, the use of 400 series stainless steel should be avoided.

In addition, two additional components are required to convert the loop to alkali metal operation.

1. A lubricant purification system consisting of a cooled, through-flow cold trap and filter to remove alkali metal - oxides and other extraneous particles.
2. A gas purifier to remove trace amounts of oxygen and water vapor in the system cover gas, to insure that no reactions take place as a result of gas contaminants.

The revised loop is shown schematically on Figure 7 which shows the addition of the lubricant and gas purification components. It is proposed that the existing Chempump be replaced with an EM pump because of its proven reliability and the performance limitations of the Chempump. The cost of a new EM pump for this application is comparable with the conversion cost of the existing Chempump. This, together, with Chempumps limited performance and probably lower reliability in alkali metals, makes the EM pump the far more desirable. Also, EM flowmeters are proposed to replace the flow limited turbine types because of their proven reliability in this type of application.

Quotations have been requested for the new components and the estimated cost of conversion of the loop to alkali metals will be covered in a separate letter.

5. Selection of Test Fluid

The liquid metal test facilities in question were designed to gain basic journal bearing information using liquid mercury at up to 350°F as a lubricant. The size and speed range together with the dynamic loading selected for the rig was primarily to complement the SNAP 2 program. The technical offices of the AEC have strongly indicated that future bearing studies should be conducted using an alkali metal so that more useful design data for future power systems and other applications can be generated.

As a test lubricant, it is recommended that NaK 78 be used. The primary basis of this recommendation results from NaK's low freezing point which allows for low temperature experimentation - in the order of room temperature to 150°F. Since, from a lubricant property standpoint, there are no drastic differences between the alkali metals, it is suggested that the one least likely to compromise the experiments be selected. NaK is the only readily available alkali metal which is liquid at room temperature. Its properties are intermediate between those of sodium and potassium.

Use of sodium or potassium would require that all experiments be conducted at temperatures in the order of 325°F such that effective cold trapping could be employed. Furthermore a number of controlled heating elements would be required on the various loop and rig components to prevent local freezing or cold trapping and to facilitate start-up. Although the additional heating equipment is not particularly costly it is undesirable because:

- (a) it requires constant attention during the bearing experiments.
- (b) it complicates and lengthens the start-up and shutdown operation, hence increasing the cost of collection of the experimental data.
- (c) the required insulation complicates loop maintenance and it significantly increases loop decontamination hazards.

Furthermore, the rig becomes more difficult because of the increased need to temperature compensate the bearing components and the instrumentation. Since the test bearing will behave essentially the same with any of the alkali metals, the simplicities afforded by NaK makes its choice attractive.

Attempts were made to determine whether any change in the properties of NaK 78 would occur with time, due to its being a two constituent fluid, which would

produce uncertainties in the fluid properties and, hence, in the test data. Liquid metal specialists in Mine Safety Appliance Co., and NASA, Lewis Research Lab. as well as the materials engineers at M.T.I. were consulted on this point. No evidence to the effect that there would be any such change of the physical properties was found. Particularly in view of the fact that operation will be in the room temperature to 150°F range, i.e. very much below the B.P. of the constituents or of any disassociation point, it appears that NaK 78 is entirely suitable for the proposed test program.

Consideration was also given to conducting, at least the initial parts of the tests in liquid mercury in order to minimize loop conversion costs. Liquid mercury has extreme properties when compared to the alkali metals. Some of these are summarized in the table below, compared with NaK 78 (Data from Ref. 5).

TABLE 5-1
Comparison of Properties

	<u>Hg</u>	<u>NaK 78</u>
Absolute Viscosity (cp)	1.21 at 100°C	0.468 at 103.7°C
Density (gm/cm ³)	13.352 at 100°C	0.847 at 100°C
Kinematic Viscosity (c.s.)	0.091 at 100°C	0.55 at 100°C
Specific Heat (cal/gm/°C)	0.03279 at 112°C	0.227 at 100°C (interpolated value)
Surface Tension (dynes/cm)	454 at 112°C	110 to 120 at 100°C

In view of these differences, problems associated with hydrodynamic lubrication are likely to be much more severe with mercury, than with alkali metals. For example:

1. The large difference in kinematic viscosities will produce much higher values of Reynolds Number with mercury lubrication.
2. The large difference in densities, may make fluid inertia effects significant in the case of mercury lubrication but negligible in the case of alkali metals.
3. The large difference in specific heats will produce much higher

lubricant film temperatures with mercury than with alkali metals.

4. The difference in surface tension, as well as in the wettability of the fluids will influence the rupture regions in the film and, possibly, also the occurrence of cavitation and cavitation damage.

In conclusion then, it appears that the extrapolation of data obtained with liquid mercury, to lubrication with alkali metals may well introduce extraneous problems that will not be encountered in the latter case.

6. Work Statement and Schedule for Installation and Prove Out of the
Alkali Metal Loop and Rotor-Bearings Test Rig.

The purpose of Phase IIA of the program was to set up and to prove out both the liquid metal loop and the rotor-bearings test rig, prior to conducting the liquid metal lubricated bearings and rotor dynamics test program. The actual liquid metal experimental work would then be conducted in a separate part of the program.

The (Ref. 2) proposal on Phase IIA listed the following work items:

1. Installation of the test facility (test rig, mercury loop and instrumentation.
2. Prove out of the existing test rig (utilizing easy to use fluids, to determine whether the problems encountered with the rig at AiResearch could be corrected).
3. Program review to conclude whether or not the existing test rig could be used for the liquid metal tests.

In view of the request that alkali metals be used in place of mercury for the test program, a study of this was made prior to commencing installation of the loop and it has been reported here. In addition, a detailed analytical study of the existing rotor-bearing test rig was made to determine its usefulness in the liquid metal program and the results have also been reported here.

Based on the conclusions of these studies, we recommend the following:

1. Install a NaK loop, as shown schematically in Figure 7, and prove out the loop components. This will include the existing and new components as noted in Section II-4 of the report.
2. Design, fabricate and prove out in silicone oil a two bearing test rig as shown schematically in Figure 5.

These work items and the revised schedule are described below.

Revised Work Items

1. Reporting and test planning - monthly letter type progress reports will be submitted. A test plan will be prepared prior to initiation of loop and rig check-out operations which will include descriptions of test objectives, data required to meet objectives, method of data analysis, and form of analyzed data. A portion of the test plan will contain instruction and precautions for operating the alkali metal loop and rig.
2. Rig mechanical design and layout - the design and layout of the test rig and drive assembly will be completed with particular emphasis on "purity" of data and minimum revision for rotor dynamics work. A design review will be completed prior to initiation of detail drawings.
3. Rig detailing and checking - detail drawings of the test rig will be completed, checked and released for fabrication.
4. Rig fabrication - the components of the test rig will be fabricated both at MTI and subcontractors plants. All critical inspection will be accomplished and recorded at MTI.
5. Rig assembly and check-out - the rig components will be assembled and alignment verified. Static load-deflection tests will be conducted. The rig will be operated using silicone oil as a test fluid to establish the integrity of the rig. The actual rig instrumentation, and all practical loop instrumentation will be used for this test.
6. Loop Modifications - performance specifications and envelope requirements will be prepared such that major components and subassemblies can be procured from qualified sources. Schematics and loop installation drawings will be prepared.

7. Procurement of Loop Components

8. Loop installation and check-out - the dry box will be modified to fit within the existing MTI test cell and all loop equipment will be installed. All plumbing will be tested for integrity, electrical installation checked and instrumentation calibrated as required. All instrumentation and controls (but not the actual NaK circulation assembly) will be used for the rig checkout operation. Following the rig checkout, the loop will be operated (with the dry box by-passed) to establish the integrity of the NaK system.
9. Report - a report covering the significant features of the loop and rig design along with the data analysis from the rig assembly check out test will be prepared.

REVISED SCHEDULE

	Weeks																																
	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27	28	29	30	31	32	
1. Project Mgt. Reporting & Test Planning																																	
2. Rig Mechanical Design & Layout																																	
3. Rig Detailing & Checking																																	
4. Rig Fabrication																																	
5. Rig Assembly & Checkout																																	
6. Design of Loop Modifications																																	
7. Procurement of Loop Components																																	
8. Loop Installation & Checkout																																	
9. Report																																	

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1. "Development of Liquid Metal Lubricated Bearings," MTI Proposal 63RD115 (Rev. 1), August 20, 1963.
2. "Memorandum on Liquid Metal Bearing Tests," by B. Sternlicht and E.B. Arwas. (Memo issued to Mr. N. Grossman, U.S. A.E.C., March 11, 1964.)
3. "IBM 704 Computer Program: Unbalance Response and Critical Speeds of a Rotor with Flexible Bearing Supports," MTI Report 62TR3. (See also "A General Method for Calculating the Critical Speeds of Flexible Rotors," by M.A. Prohl, Trans. ASME, 1954.)
4. "On the Translatory Whirl Motion of a Vertical Rotor in Plain Cylindrical Gas Dynamic Journal Bearings," by C.H.T. Pan and B. Sternlicht, ASME Paper 64-Lub-4.
5. "Reactor Handbook, Vol. 1, Materials," 2nd Edition, Edited by C.R. Tipton, Jr., Interscience Publishers Inc.
6. "Topical Report on Turbulent Lubrication", edited by E.B. Arwas, M.T.I. Report 64TR67, November 30, 1964.

Fig. 1 - Rotating Assembly Liquid Metal Bearing Test Rig

Fig. 2
Mercury Test Rig Rotor-Bearing
Critical Speed

Fig. 3
Mercury Test Rig
Critical Speed
Normalized Mode Shape

Fig. 4
Mercury Test Rig
Shaft Vibrational Mode Shape

Fig. 5
Schematic of Two Bearing Test Rig

Fig. 6
Installation of Proximity Probes
In Liquid Metal Test Bearing

APPENDIX A

1. Summary Review of Loop and Test Rig, As Received at MTI

This section of the Appendix A summarizes the major components of the mercury test facilities as received from AiResearch by broadly outlining the makeup, condition and required equipment of the major components.

1. Mercury Loop - The loop consists of the dry box with supporting structure and platform together with the necessary circulating equipment as outlined below:
 - a) Mercury pump with appropriate switching but without pressure instrumentation
 - b) Mercury turbine type flowmeters but without readout equipment
 - c) All required mercury valves
 - d) Drain reservoir with level indicator and readout panel.
(It is understood that the level indicator is inoperative)
 - e) Mercury to water heat exchangers
 - f) Dump tank and appropriate valving
 - g) Loop pressure transducers but no signal conditioning or readout equipment.

The loop came with the interconnecting wiring and conduits. However, due to the difference in installation locations between MTI and AiResearch, these must be replaced. The dry box structure, platform and several loop plumbing welds were broken in shipment requiring partial disassembly of the loop at MTI.

2. Vacuum System - The vacuum system used to evaluate the loop and dry box consists of:
 - a) Roughing and diffusion pumps together with appropriate switching

- b) Vacuum transducers and readout equipment
- c) Miscellaneous valves, fittings and pipe. (Much of the vacuum plumbing was tailor made for the AiResearch installation.)
- d) The components for purification of the inert gas supply were not supplied

3. Miscellaneous Maintenance Supplies

- a) Helium leak detector
- b) Mercury vacuum cleaner
- c) Steel clothes lockers and shoe racks
- d) Resperators
- e) Mercury vapor detector
- f) Mercury distiller
- g) Mercury

4. Test Rig - The bearing test rig was received in a partially assembled condition and is complete with several spare parts. Pertinent observations concerning the test rig are summarized below:

- a) The drive motor is burned out and partially disassembled
- b) The Photocon proximity and pressure transducers are in various states of disrepair. The signal conditioning equipment for these items have been checked out and are operative
- c) The two Bently proximity transducers are complete with amplifiers and power supply and appear to be in operating condition
- d) There is a fixture for calibration of the dynamic pressure transducers, the condition of which is unknown. The signal conditioning equipment is in working order but no transducers were supplied

- e) The motor-alternator for the drive motor together with the control panel was supplied. However, it has not been checked out.

5. Readout Equipment - The major readout equipment associated with the rig and loop instrumentation supplied to MTI is limited to:

- a) One dual trace oscilloscope
- b) Two frequency meters
- c) Temperature indicator with switching assembly

2. Equipment Inventory

The following twelve pages are a complete itemized inventory of all AEC owned property received from AiResearch on 17 August, 1964 (Items 1-263) and 4 June, 1964 (Items 263-287) for use on the subject contract.

Some of the equipment shown on the inventory record is damaged, or mercury contaminated, and the extent of repair has not been fully determined. The condition of many of the operating components such as valves, transducers, meters, etc. is unknown. Therefore, some of the equipment listed here may require repair and maintenance prior to its use in a workable liquid metal loop.

INVENTORY OF AEC OWNED PROPERTY
RECEIVED FROM AIRESEARCH FOR USE ON RD323

MTI No.	Description	AEC No.	Ref AiResearch So No.	Quan.	Remarks
1	Balance Adaptor SK-8513-31			2	
2	Socket Hd Cap Screw		527	12	
3	3/8 - 20NF X 7/8 Cap Screw		526	4	
4	52 Cinch Jones Tie Point		656	3	
5	Offset Screw Driver		319	2	
6	Metal "O" Ring (o250-1-PF)		86	4	
7	Spacer (SK 85B-11)		49	11	
8	Packing Set W/Teflon		538	4	
9	1/2" SS Tubing (SK85B-14)		786	1	
10	1/14" + 7/16" Tygon Tubing		39	4ft	
11	Metal "O" Rings (0625-2-3)		89	6	
12	P47-85-B-269 Lug		24	3	
13	Screen		622	1	
14	Dowel Pin P47-85B-23-5		21	2	
15	Plug		1	8	
16	Rotor Point SK-85B42			1	
17	Tape Measure		17	1	
18	Rotor Points SK-85B142			3	
19	Metal "O" Rings (0500-1-43oPF)		524	21	
20	Bearing Discs SK-85B-5			2	
21	Cone Points (P47-85B-13)		11	4	
22	Cup Bearing (PAP-102339)		512	2	
23	Carbide Bearing K-94 (1 1/8 x 1 3/4)		64	1	
24	Special Screw (AN505-10-9)		4	4	

MTI No.	Description	AEC No.	Ref A Research So No.	Quan.	Remarks
25	Bearing Support (P47-85B-18)		547	1	
26	Balance Adaptors SK-85B-32			2	
27	Special Screws (AN565-A6-H4)		5	6	
28	Cap Screws (1/2-20 x 1 1/2)		94	5	
29	90° Elbow Union (810-9-440C)		111	2	
30	Fittings (SK-85B-23)			2	
31	Arbor (P47-84A-207)		505	1	
32	Bearings (P47-85B-22)		507	2	
33	Cap Screw (1/2-20x1 7/8		95	1	
34	56B - Cinch Jones Tie Point		658	3	
35	53B - Cinch Jones Tie Point		657	4	
36	MS3106A-28-22P Connector		590	1	
37	1/4 - 20NCX 1 1/16 Cap Screw		525	1	
38	PAP-102339 Cap		512	1	
39	400-9-440C 90° Elbow Union		109	3	
40	Bearing Plate (SK-85B-43			2	
41	P47-85B-22 Bearing Sleeve		548	1	
42	1/4" w3/8 sh Socket Hd Screw		90	5	
43	Bearing (SK-85B-41)			2	
44	ANSO5-3-3 Screw		3	2	
45	Bearing Support P47-85B-18		547	1	
46	P47-85B-15-5 Spacer Sleeve		15	3	
47	.007 x 3/4 (Steel Trancort Strip		520	900ft	
48	1/4" Tube Bender		323	1	
49	3/8" Tube Bender		325	1	
50	Phillip Screw Driver		313	2	
51	12" Adj Wrench		343	1	
52	15" Tube Cutter		350	1	
53	1/8" Drive Socket Set		324	1	
54	8" Adj Wrench		338	1	

MTI No.	Description	AEC No.	Ref AiResearch So No.	Quan.	Remarks
55	10" Adj Wrench		340	1	
56	Nut Driver Set		310	2	
57	10" Tuber Cutter		342	1	
58	1520A (Arc Joint Pliers)		357	2	
59	7 1/4" Lineman Pliers		337	2	
60	Rubber Mallet		317	2	
61	6 3/4" Slip Joint Pliers		336	2	
62	Electrician Pliers		312	2	
63	4 oz. Ball Peen Hammers		333	2	
64	12 oz. Ball Peen Hammers		344	1	
65	10" Pipe Wrench		341	1	
66	Punch and Chisel Set		318	2	
67	15" Adj Wrench		351	2	
68	1/2" (11 Pc) Drive Socket Set		330	1	
69	Hack Saws		315	2	
70	Screw Driver Set		316	2	
71	1/2" Tube Bender		328	1	
72	1/8" to 2" Pipe Cutter		321	1	
73	3/8 (17 Pc) Drive Socket Set		326	1	
74	3/8 to 1" Wrenches (Open & Box)		327	1	
75	6" Needle Nose Pliers		335	2	
76	1 1/16 Combination Wrench		331	2	
77	1 1/8 Combination Wrench		332	2	
78	Pipe Wrench (14")		349	1	
79	Nylon Shoes		117	12	
80	Gas Detector & case & Tubing (+10 Tubes)		302 & 305	2	
81	P47-85B-67 Instrument Port Cover		175	1	
82	Tin Snip		345 - 347	3	

MTI No.	Description	AEC No.	Ref AiResearch So No	Quan.	Remarks
83	86-SP Hd Heater		660	2	
84	SK-85B-15 1/2 Tube S S		787	1	
85	CT1641V 1" Micrometer & Case		296 - 298	1	
86	CT 1642V 2" Micrometer & Case		297 - 299	1	
87	EG 187-AT Conax Electrode Gland		174	3	
88	Type "M" Terminal Clamp		447	2	
89	P47-85B-38 Instrumentation Bracket		540	1	
90	CR-40397 Cartridge		306	10	
91	P47-85B-15-3 Bearing and Spacer		14	3	
92	185-290-1 Transducers Absolute Press		179	2	
93	77MIP11 Amphinal 1-11Pin Socket		659	1	
94	CR-73654 Filters		307	4	
95	2550 Connectors		535	11	
96	299 Panel Rack		534	1	
97	MS3106B-28-15S Cannon Connector		591	2	
98	MS3102A-28-22S Cannon Connector		587	1	
99	MB3106B-28-21SK Cannon Connector		573	2	
100	DOX826233 Solenoid Air Valve "110" Volt		498	1	
101	P47-85B-16 Speed Gear		16	1	
102	A-2018 Terminal Strap		446	2	
103	Molybdenum Bar		59	101#	
104	(304 s/s) Reducer 2 1/2" x 2"		151	1	
105	P-47-85B-38 Bracket		29	2	
106	MS3102R-28-18P Cannon Connector		588	1	
107	8008-7308 Tube		497	1	
108	MS3102A-28-21P Cannon Connector		585	1	
109	Teflon Inserts for CFH-T7-1/2-1-65		760	1	

MTI No.	Description	AEC No.	Ref AiResearch So No.	Quan.	Remarks
110	MS3102R-28-18S Common Connector		589	1	
111	CPHT7-1/2-1-6S Carbide Bearing		759	1	
112	(316S/S) Adaptor 2" oxx2" NPT		150	1	
113	No. 16 x 10" Single Jack Chain		107	98'	
114	Hand Knobs - Swing Bolts - Washers		406, 412, 413	8ea	
115	1150-H-1 1% Microammeter (0-500psi)		180	2	
116	7763 Connector (Male) Twist-Lock		682	12	
117	200W Solder Gun		355	1	
118	Molbdenum Ring (1 1/4 x 2 x 3/4)		65	3	
119	GTC - 100 Vacuum Gage		115	2	
120	GPH - 100A Ionization Gage Cold		114	1	
121	GTC - 100 Thermocouple Gage		427	2	
122	1 1/2" Dia Molybdenum		552	1	
123	SC-4 Calibrated Lock		433	1	
124	7769 Cover		683	12	
125	Molybdenum Ring (4 x 4 3/3 x 3/4)		72	1	
126	316P561-13s/n C x H Motor		436	1	
127	3/4" ODX 120W Tubing		138	9"	
128	P47-8513-67 Instrument port		712	1	
129	5" Dual Bar (SAF-660 Bronze)		528	6 1/2"	
130	3/4" Thick Plate (347s/s)		137	161b	
131	CUS-100 - P-W Valve s/s Bellows		112	1	
132	18" Pipe Wrench		352	1	
133	1206AMXU 1/2" Seal Valve		441	1	
134	P-312 Vlier Toggle Pads		122	4	
135	CR-855559 Respirator		308	8	
136	Sawing Vice (3/16" to 3 3/8")		311	1	
137	1 1/2 x 2" Bar (347s/s)		145	17"	

MTI No.	Description	AEC No.	Ref AiResearch So No.	Quan.	Remarks
138	6" x 6 3/4" Bar (Inconel)		104	1	
139	P47-85B-50 Electrical Components		795	16	
140	54808 Hand Pump		699	1	
141	(430SS) Tubing (3/4" x .095W)		455	32'	
142	7N253oSF Dry Box Gloves		218	2	
143	30N20A Hoffman Type I Panel Enclosure		643	1	
144	P47-85B-69 Vapor Detector Panel		732	1	
145	PS-606 Power Supply	165623		1	
146	DS-600 Meters Dynagage	165611		1	
147	P47-85B-51 Powerstat Panel		714	2	
148	" " " " "		595	1	
149	Dynagage - Power Supply DG600-PS605	165624		1	
150	Meters, Dynagage DG600	165592		1	
150	" " "	165593		1	
150	" " "	165594		1	
150	" " "	165595		1	
152	200 ABR Audio Oscillator		697	1	
153	P47-85B-53 Alarm Control Panel Assy		792	1	
154	P47-85B-63 Temp and Sw Assy		794	1	
155	117-1A Mercury Vapor Detector	165442		1	
156	Meter Control Panel and Chassis P47-85B-59)	165600		1	
157	Microammeter Panel and Chassis		790	1	
158	P47-85B-65 Magnetic Clutch Panel and Chassis	165599		1	
159	Amplifiers, Model 568	165556		1	
"	" " "	165557		1	
"	" " "	165558		1	
"	" " "	165559		1	
"	" " "	165560		1	
"	" " "	165561		1	

MTI No.	Description	AEC No.	Ref AiResearch So No.	Quan.	Remarks
159	Amplifiers, Model 568	165562		1	
"	" " "	165563		1	
"	" " "	165564		1	
"	" " "	165565		1	
160	Power Supply "NFBR"	165551		1	
161	Indicator, Brown, Vert 156 x 12P	165443			
"	Switch Assy 24PT # 78284	165585			
162	Powerstat Motor Driven (15M12S6Dt-25)	165450		1	
163	" " " "	165451		1	
164	Transformers 15DVA (15KVA548-C-11515A)	165634		1	
165	" " "	165635		1	
166	1/2" FloLine Tubing		496	200'	
167	5" Jaw Vise	165612		1	
168	H.D. Heater (8708-SP)		690	12	
169	Mercury Collector, Model 9002	165440		1	
170	Air-Packs, Scott, 6000C-B4A	165437		1	
170	" " "	165438		1	
170	" " "	165439		1	
171	Basket and Rack Assembly + 21 Baskets			1	
172	Spectrometer, Helium MS-9AB	165583		1	
173	2 1/2 x 0.083W Tubing		475	4'	
174	1/2 x 0.040"W. Tubing (430s/s)		453	22'	
175	Valve Vacuum PR 400M	165614		1	
175	" " CUS200LW	165615		1	
176	" " PR-400M	165620		1	
177	" " " "	165619		1	
178	Pump Diffusion PMC-720	165553		1	
179	Electrical Equipment Cabinet (P47-85B-41)		713	1	
180	Locker Assembly Steel 52222		358	20	

MTI No.	Description	AEC No.	Ref AiResearch So No	Quan.	Remarks
181	3/4" x 24" x 482 (Buna "N" Material)		752	47#	
182	Pipe 8" (Sch.10) (321s/s)		159	11"	
183	Complete Chamber Assembly and Sample Chamber 117-1A		187 & 205	1 ea	
184	6" Dia. x 6 5/8" Molybdenum Titanium Bar		530	1	
185	P4785B-34 Probe Test Fixture		366	1	
186	.032 Dia. Copper Wire		58	67#	
187	Mercury Decon. Powder		300	50#	
188	VTS-431 Regulators		127	2	
189	KD-30 Vacuum Pump	165550		1	
190	" " "	165631			
191	Power and Relay Inst. Panel Assembly		796	1	
192	Cases for 568 Amplifier		400	10	
193	Glass Silicone Insulation (.011 x 36 x 42)		577	1	
194	Cases for 117-1A Vapor Detector		301	1	
195	Furnace Model L-1		377	1	
196	Detector, Amplifier	165625		2	
197	" "	165626		1	
198	1/6 HP Motor-(Component of Vapor Detector)		788	1	
199	Drill Motor 1/2" S/B Tagged 80-111		394	1	
200	Gage, ionization GPH-100A	165555		1	
201	Cable Termination Box		536	6	
202	Pyrex Brand Glass (8" Dia. x 1/2)		756	2	
203	6" OBX x 120W Tubing		158	8"	
204	P47-85B-61-15 Control Panel		722	4	
205	" " " 19 " "		724	1	
206	" " " 17 " "		723	1	
207	" " " 23 " "		725	1	

MTI No.	Description	AEC No.	Ref AiResearch So No.	Quan.	Remarks
208	P47-85B-61-57 Panel Alum		728	1	
209	" " " 27 " "		727	4	
210	" " " 59 " "		729	8	
211	" " " 25 " "		726	6	
212	36" Paper Cutter		353	1	
213	3/8" Thick Plate (347s/s)		133	1	
214	1 1/16 " Hostelloy Weld Rod		693	20#	
215	Meca Plate .012 x 36 x 12		523	1	
216	PMC-720 Diffusion Pump		123	1	
217	Pump, Corblin Diaphram	165638		1	
218	1 1/4" x 0.065 W Tubing (430 S.S.)		459	40'	
219	2 1/2" x 0.187 W Tubing		462	48'	
220	6" Channel Steel		466	20'	
221	1" x 8" Flat Bar (1018)		458	1'	
222	Pipe(Sch. 40) 1"		457	84'	
223	Manifold Systems for Vapor Detector		789	1	
224	Vari-Drive, 15 HP P47-85B-56	165552		1	
225	Alternator 10 K.V.A. P47-85B-56	165589		1	
226	Chempump Model CFHT 7 1/2-1-65	165454		1	Part of MTI 263
227	Flowmeters 6-10 l/min.	165566		1	"
"	" "	165567		1	"
"	" "	165590		1	"
"	" 6-30 /min.	165568		1	"
"	" "	165569		1	"
"	" "	165591		1	"
228	Transducers Teleflight Press 185/295-1	165622		1	
229	" " " "	165621		1	
230	Valve, Vac CVS-100 RW	165616		1	

MTI No.	Description	AEC No.	Ref AiResearch So No.	Quan.	Remarks
231	Valve, Vac CVS-100 RW	165617			
232	Control Panel and Chassis (P47-85B-52)		791	1	
233	Mercury Distillery and 3 Valves (P47-85B-66)			1	
234	PT4 Probes		514	6	
235	PT3-PRP-5614 Transducers			1	
236	PT10-3732 Transducers		380	2	
237	D-152 Detector Amplifier		368	2	
238	H-3-042-6 Temp Probe and Cables		376	2	
239	7ea #PZ-6 Pressure Sensors		381	8	
240	H-1074-3 Sensor Heads and Cables		374	2	
241	H-1-084-3 Sensor Heads and Cables		375	2	
242	PT4 Photocon Transducers with Shields and Calibration S Tubes		379	4	
243	Disc 4100-5		405	13	
244	Tube 4-4020-5		392	10	
245	Tube 1-1010-5		286-287	50	
246	Crystals .100 x .040		385	100	
247	Tube 2-2020-5		391	21	
248	Apron 10825-1		361	1	
249	WeldRod 19-9		365	31 b	
250	Freq Meters 500Br	165570	397	1	
251	" " "	165571	"	1	
252	1/16 x 36 x 42 epoxy Glass		61	1	
253	2" x .188W		70	6"	
254	5" x 9" Molybdenum + 1/2 ZR		101	1	
255	Mercury		418	358#	
256	Oscilloscope	165572		1	
"	"	165574		1	
"	"	165573		1	
257	PT-3 Transducer		378	1	

MTI No.	Description	AEC No.	Ref AiResearch So No	Quan.	Remarks
258	MS 3102A-22-14P Connector		582	1	
259	2 1/2" Weld Cap		278	2	
260	522B Counter		698	1	
261	SK85B-35		745	1	
262	3/8 x 035 Tube (43053)		452	25'	
263	P47-85B-5 Loop Assembly		799	1	Less 1 of Item 19 1 of Item 17
264	Bearing, Liquid Metal P47-85B-22		19	5	
265	Base Plate P47-85B-23		20	1	
266	Rotor/Bearing Support with SK-85B-59		52	1	
267	Inconel X Shaft Assembly		10	2	
268	Bearing Disc SK-85B-44		33	1	
269	Socket Head Cap Screw		100	12	
270	Plug-Passage SK-85B-8		46	6	
271	Deflector Adapter with P47-85B-10		12	1	
272	Bearing 1 PC in P47-85B-10		7	3	
273	Nut, Locking		13	1	
274	Bearing Plate SK-85A-59		34	1	
275	Cover Plate		47	2	
276	Metal O-Ring with P47-85B-10		38	2	
277	Metal O-Ring with P47-85B-10		86	6	
278	Slotted Flat Head Screw		47	4	
279	Bearing Housing		8	1	
280	Rotor Stator Assembly		27	1	
281	Socket Head Capscrew 1 inch		32	4	
282	Support Assembly		809	1	
283	Shaft		25	1	
284	Washer		25	2	

MTI No.	Description	AEC No.	Ret AiResearch So No	Quan.	Remarks
285	Socket Hex Head Bolt W/Assembly Support		93	2	
286	Ball Lock Pins		6	2	
287	Base Support		520	1	