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HIGH POWER CRU
SCALING ANALYSIS AND PRE-CONCEPTUAL DESIGNS

* * *

Pre-Conceptual Designs Contributed by Thompson Ramo Wooldridge
Prepared by Atomics International

Exempt from CCRP Re-review Requirements
(per 7/22/82 Duff/Caudle memorandum) *NK*
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I. SUMMARY

An efficient approach to the development of high power CRU's is by extending existing technology. This will ensure early attainment of maximum reliability machinery. Maximum utilization of present technology can be achieved by applying known scaling criteria to a proven concept.

There is no question that present MRPCP technology can be utilized to scale up the existing hermetically sealed, single shaft, liquid mercury lubricated turboalternator concept which has proven its feasibility. If maximum reliability is emphasized with conservative design rather than designing for absolute minimum weight, long endurance can be confidently predicted. Studies of known failure modes on present CRU's show that higher power level units scaled from present technology would have significantly higher inherent operational life. The component wear-out modes innate to such machinery will all be alleviated in the larger machine. Bearing erosion will differ from MRPCP CRU's approximately proportional to shaft speed ratio and inversely to bearing diameter ratio. If a 12,000 RPM, 4-pole alternator rotor is utilized, standard voltages and frequency can be generated, very close voltage regulation can be achieved, and load conversion devices minimized. The rotor will operate at low stress levels so minimum long term creep and subsequent shaft unbalance will be realized. Stator bore sealing will be a much easier task due to an alternator design allowing an ample air gap sufficient for a strong and reliable tubular ceramic seal. The mercury pump system will operate well out of cavitation, again enhancing long term reliable life.

The same stringent manufacturing tolerances pertinent to present CRU's can be utilized and shaft balancing achieved to an even greater accuracy proportional to shaft weight. Therefore, bearing alignment and shaft dynamic interplay will improve considerably over MRPCP units. The application of design practices and considerations to each component as rigorous as those required for present small machinery will provide the opportunity to take advantage of this significant, inherent reliability factor.

The objective of the study was to define sufficient criteria to confidently conduct a systems evaluation. Several alternator and turbine concepts were studied on a preliminary basis. After screening competing concepts, four turboalternator designs, both 15 KWe and 25 KWe CRU's were analyzed at 12,000 and 24,000 RPM. From these, a 24,000 RPM, 15 KWe CRU and a 12,000 RPM, 25 KWe CRU were analyzed in more detail to provide systems analysis criteria.

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II. INTRODUCTION

In order to fill the expected future needs of ever increasing power levels for space and extraterrestrial exploration, the development of second generation higher power level hermetically sealed CRU type turbomachinery will be required. In conjunction with the SNAP reactors now under development, a full power range capability of 3 to greater than 100 KWe can thus be provided by the SNAP Mercury Rankine power plants utilizing the same basic processes and equipment concepts. The best method of developing this high power capability is by maximum utilization of present Mercury Rankine technology. Maximum utilization of this present Mercury Rankine experience and technology can be accomplished in three ways:

- (1) By using multiple units of present machines to increase system total power level.
- (2) By using known scaling techniques to create new, higher power level machinery which is similar to that already developed.
- (3) By using multiple units of scaled-up machines to achieve even higher system power levels than is practical by (1).

Through the use of multiple PCS subsystems, the present 3 to 5 KWe CRU can readily provide up to a 20 KWe capability. This power can be further extended up to about 50 KWe with an attendant increasing performance, weight, radiator area and reliability disadvantage. A new high power CRU of 15 to 25 KWe rating can provide a more optimum power capability of 20 to 100 KWe with greater reliability. The six years of effort on the 3 to 5 KWe machine is directly applicable and forms a firm technical basis for the design, testing and qualification of such a higher power machine. This will make available the high power machinery at the earliest time and, at the same time, will assure development success and provide the maximum possible CRU reliability.

In order to properly scope the system requirements and/or system restraints, further analytical studies of weight trade-offs, performance advantages, associated component development requirements, reliability factors plus electrical output requirements will need to be undertaken. The work included within this report identifies the required development technology, potential performance limitations, key technical problem areas and state-of-the-art experience. Based on other studies now underway, a nominal power rating for the high power CRU machine can be established. Accordingly, a high power CRU development specification and a high power CRU development plan can be defined.

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III. REVIEW OF CURRENT STATUS OF MERCURY TURBOALTERNATOR DEVELOPMENT

A. CRU IVM

Initiated in February, 1962, the CRU IVM program has accumulated 8,000 hours of operational time and more than 42 injection starts. It has been a success in meeting its objectives of providing a component for ground tests of PSM-1 and PSM-3. CRU IVM has demonstrated electrical power output capabilities to 6.5 KWe, and CRU IVM-3A completed an endurance test of 4,135 hours, exceeding initial endurance objectives by an order of magnitude.

B. CRU V

The CRU V program objectives were to design, develop, fabricate, and produce flight qualified units and to simplify assembly and disassembly. Since September, 1963, the CRU V program has accumulated 9,000 hours of integrated operational time. Endurance capability, injection start and off-design performance have been verified. Three CRU V units have run over 2,000 hours each. Change in package design from CRU IVM to CRU V to simplify assembly and disassembly has resulted in a reduction of down time and the achievement of part interchangeability.

C. General Technology

All CRU programs have relied heavily upon component testing to verify designs and provide performance data for subsequent design analyses. The important development areas are numerous and include bearing performance and shaft dynamics, magnetic properties and high temperature characteristics, mercury pump performance, turbine development, alternator development and packaging.

1. Bearing Development

The bearing program for application to the CRU IVM and CRU V was directed toward the following goals:

- a. Stable bearing operation at speed.
- b. Maximum bearing lubricant film stiffness, load carrying capacity.
- c. Maximum bearing clearance and/or clearance compatible with bearing housing distortion of operating temperature.
- d. Bearing designs applicable to a variety of CRU designs.

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Bearing whirl is a vibration phenomenon that is characteristic of high speed lightly loaded bearings. Half frequency whirl is a self-sustaining resonance at approximately one-half of the shaft rotational speed and results in lower load carrying capacity or high shaft displacements. Whirl can be eliminated by stiffening the bearing lubricant film or by using a bearing configuration not so subject to this instability. A change of shaft bearing dynamic characteristics can also prevent the onset of the instability.

Bearing stiffness can be increased by increasing the bearing size (diameter) by lessening the bearing-to-journal clearance or to a lesser extent by increasing lubricant pressure.

Of all CRU components, mercury lubricated fluid film bearings have received the most intensive research and development attention. As a result of component and CRU tests, a good foundation has been laid for the design of mercury lubricated, turbulent flow bearings for any CRU design.

Bearing endurance life is affected by deposition of corrosion products in the running clearance, by corrosion-erosion of the running surfaces, and/or by cavitation-erosion of the running surface. Deposition of corrosion products manifests itself by an increasing supply pressure requirement to maintain a given flow at maximum available supply pressure to the bearing. Although the change in supply pressure-lubricant flow relationship has been observed in endurance tests, lubricant flow has not decreased to a point which jeopardizes the unit.

Erosion of bearing surfaces by corrosion or by cavitation is manifested by a decrease in supply pressure required for a given flow, and by a reduction in load carrying capacity as a result. In addition, particles of sufficient size to score or seize the bearing may be torn loose in the bearing clearance by the erosion mechanism. Complete confidence in bearing resistance to corrosion-erosion independent on a better understanding of the phenomenon and by continued demonstration of bearing life.

Cavitation in bearings is due to rapid transitions from high to low pressure regions hydrodynamically developed in bearings. Two approaches which have been used to minimize or eliminate cavitation effects are to eliminate

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the effect of low pressure regions by grooving as in the three sector bearing or by increasing the number of feed sources to make the bearing more nearly hydrostatic. Each of the methods have been demonstrated as solutions.

2. Magnetic Properties

Rotating magnetic loads are caused largely by an unsymmetric magnetic orientation of the rotor magnet poles during the casting process, and to some extent by the magnetization of the rotor in the alternator assembly. Eccentricity of the magnets relative to the center of rotation of the bearings also causes rotating magnetic loads. Unidirectional magnetic loads may result from eccentricity of the stator relative to the bearing center.

Bearing loads due to magnetic forces have been investigated both analytically and experimentally. New assembly and magnetization techniques as well as magnet quality control and selection criteria have been developed. High temperature magnet degradation and performance limits have been determined. Consistent rotor magnet performance and low magnetic loading have now been attained.

3. Mercury Pump Development

The mercury pump may suffer from long term deterioration effects due to erosion caused by corrosion or cavitation. Simple pump configurations may not be adequate to provide the flow or pressure required by the system. Research testing has been performed to evaluate the erosion effects under the most severe conditions to be encountered in flight in the final CRU V design. Compound pumps and various rotor and diffuser configurations were analyzed on the basis of methods already developed and correlated with test experience.

Techniques have been developed to design liquid mercury, centrifugal pumps for discharge pressures and flows in the foreseeable range of MRPC requirements. However, as flow rate requirements increase without corresponding increases in permissible net positive suction head, the resultant demands for increasing suction specific speed require that mercury pump development continue. Suction specific speed is indicative of the cavitation performance of the pump. The current suction specific speed capability of 10,000 for the impeller alone is unlikely to be increased to any degree by changes in the impeller design. The overall jet boost-centrifugal pump combination has a suction specific capability of 30,000 which could be increased by refinements in the design of the jet boost pump.

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4. Turbine Development

Design and fabrication techniques applicable to small turbines have been achieved utilizing high temperature materials with a minimum susceptibility to creep during long term operation. Assembly techniques have been developed to avoid overstressing turbine wheels with shrink-fits. A great deal of analysis has been done to attain an optimum staging and stage work split to assure maximum efficiency from a given design point. A re-search program has been initiated to explore the one unresolved theoretical area of mercury vapor supercooling during expansion. Understanding and application of this phenomena may provide several percentage points greater turbine efficiency due to design improvements.

5. Alternator

Considerable progress has been made on hermetically sealing the stator bore. Coated conductor techniques are being developed as well as Pyroceram and metallic seals applied to the stator bore itself. Demonstration of a rugged and reliable bore seal has not yet been accomplished; but sufficient progress has been made that leakage rates are held to insignificant levels and can easily be vented to space through evacuation techniques. This prevents vapor attack on conductors and the inventory losses to space are measured in hundredths of a pound per year.

6. Packaging

Packaging of CRU V is a distinct departure from that of previous models. While the specific application and system requirements will determine the component arrangement and package configuration, CRU packaging is a technology in itself. Analytical techniques for shaft dynamics, stress and distortion determinations, and transient and steady state thermal analysis techniques have been developed. Instrumentation techniques to establish shaft motion have been verified. Package and internal sealing techniques have been developed for specific CRU designs, but which are applicable generally. Alternator cooling techniques have been developed in this general area.

To summarize, the Mercury Rankine effort has developed a high level of technical competence in general; and, in particular, has demonstrated the feasibility of the single shaft, hermetically sealed, liquid metal bearing turbo-alternator concept on two separate machine designs. In the following discussion, the term "scaling" refers

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essentially to a technology level which has been developed and a concept which has been proven feasible, not to absolute optimum performance attainment or to reliability development. The achievement of these latter two will follow the usual development process wherein weak materials or marginal designs will be improved so as to optimize performance and eliminate early failures (infant mortalities).

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III. SCALING CURRENT CRU TECHNOLOGY TO HIGHER POWER

Scalability of existing technology and application of developed know-how should be weighed heavily in selecting a higher power level concept. A study was made to evaluate the potential scaling capability of present technology to higher powers. The object was to define probable scaling limitations or uncertainties as a function of increasing power level. No attempt was made to achieve a conceptual design of a discrete power level unit, rather general trends were evaluated.

The relative change of significant CRU characteristics as a function of power level is shown in Figure 1. This is predicated on a relative change in shaft speed rather than by step jumps. Figure 2 shows relative application of present component technology to higher power level CRU design.

A. Alternator

Several assumptions were made in pursuing this study. Foremost was the decision to utilize a BEEM alternator at any power level over 5 KW. Further assumptions were:

1. Generate 400-cycle power directly. This restricted speed since either 2 poles at 24,000 RPM or 4 poles at 12,000 RPM are required. Together with the power selection, this assumption essentially dictated machine size and weight.
2. Use standard voltage of 120/208 volts.

Rotor stress was considered the component characteristic limiting for extended high temperature operation due to creep considerations. Although BEEM rotors have been tested at surface speeds of over 500 fps hot and 800 fps cold, the rotor design surface speed was held under 350 fps to provide very conservative design margin for one-year life.

Rotor length to diameter ratio was held essentially constant as was the L/D of the stator. The number of slots over stator diameter and number of turns per slot were also held constant. These, together with the electrical requirements (voltage, power, and frequency), dictated the BEEM rotor configuration.

B. Shaft Dynamics and Bearings

As specified in Paragraph A above, to achieve a required electrical output, with a constant rotor L/D, it was necessary to provide rotor length increases proportional to rotor diameter changes. This change must be compensated by bearing-shaft scaling to maintain the flexible shaft critical speed at a safe margin above operating

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CHARACTERISTICS OF HIGHER POWERED CRU'S
USING CRU V TECHNOLOGY

Figure 1

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RELATIVE CHANGE FROM PRESENT CRU V

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EVALUATION OF HIGHER CRU POWER CAPABILITY

COMPONENT	POWER LIMIT FOR PRESENT TECHNOLOGY	COMPONENT DESIGN RELATIONSHIP TO PRESENT EQUIPMENT (CRU V)	ANTICIPATED PERFORMANCE OR ENDURANCE MARGIN TO HIGHER POWER LIMIT
TURBINE	25-35 Kwe	SAME STRESS LEVEL SAME FLOW PARAMETERS (i.e. U/V, pressure ratio) SAME BLADE & STATOR GEOMETRY SAME FABRICATION TECHNIQUE	SAME
BEARINGS	50 Kwe	SAME PARTIAL ARC GEOMETRY SAME TURBULENT ANALYSIS APPLIES SAME SUPPLY PRESSURE SAME HYBRID OPERATION LOWER ROTATING SPEED	BETTER CAVITATION EROSION TOLERANCE
ALTERNATOR	50 Kwe	DIFFERENT ALTERNATOR (field excited) (similar configurations already operated between 5-90 Kwe) SAME SOLUTION TO STATOR WINDING PROTECTION SAME MATERIALS USED SOME CHANGE IN COOLING DESIGN	BETTER PERFORMANCE (regulation, short circuit stability)
MERCURY PUMP	50 Kwe	SAME GEOMETRY IMPROVED BLADE PROFILE SAME SPECIFIC SPEED LOWER RPM	BETTER, LESS SUSCEPTIBILITY TO CAVITATION & NONCONDENSIBLE
SHAFT (critical speed)	NO LIMIT	SAME MARGIN BETWEEN SHAFT CRITICAL SPEED AND OPERATING SPEED SAME MATERIALS SAME BASIC GEOMETRY SAME BALANCING TECHNIQUE	SAME
HOUSING	- - -	SAME TEMPERATURES & PROFILES SAME PRESSURE LEVELS SAME STRESS LEVELS SAME MATERIALS	SAME

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Figure 2

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speed. Bearings increased in diameter to accommodate the necessary increase in shaft diameter. Bearing L/D was maintained constant so bearing length increased proportionally. The bearing and shaft dynamic parameter characteristics are shown in Figure 1. Up to the maximum power level evaluated (35 KW), there appears to be no restriction in scaling present bearing technology.

C. Turbine

The turbine design was dictated by the alternator design (i.e., size and speed). The present CRU V design point and energy split in a two-stage turbine was retained although it is not optimum at the highest power levels. Power increases to approximately 25 KW can be accomplished by maintaining the same turbine peripheral speed to spouting velocity ratio while increasing mass flow capacity by increasing rotor blade length using the present configuration. At approximately 25 KW, a limit to good two-stage turbine performance was reached. This occurs because the turbine blade height-to-rotor diameter ratio becomes too large for good turbine design. Turbine performance is critical to overall machine and system efficiency and weight, so turbine redesign would be advisable beyond 25 KW. This would involve increased staging, change of work split, partial admission changes, and blade contour and configuration changes; but would represent only minor redesign well within present technology capability.

D. Mercury Pump

The jet-centrifugal pump presents the least problem of any subcomponent considered. The decrease in rotational speed, together with increased size, are all in the right direction to achieve better pump performance and efficiency. Each pump characteristic was analyzed and no limiting parameter was found. Larger size pumps will lend themselves to improved methods of forming which will improve cost and lead time, reliability, and performance. The bubble swallowing capacity of the larger pump will increase proportionally to the flow volume in the pump.

Based on the scaling parameter, suction specific speed, there is a definite advantage in considering the 12,000 RPM, 4 pole machine in that the lower suction specific speed will provide greater margin above cavitation condition or, alternatively, allow pumping at lower NPSH.

E. Packaging

The packaging criteria which led to the CRU V cradle concept remain valid at higher power levels:

1. No shaft disassembly after balancing.
2. Maximum insensitivity to thermal transients.

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3. Optimum axial temperature profile.
4. Symmetrical bearing alignment structure.
5. Adaptable to rapid fabrication, assembly, and disassembly.

It will be necessary to establish the comparative advantages of the basic differences in the alternator cooling and housing assembly concept employed in CRU IVM and CRU V. Two methods of cooling may be considered, either of which has attractive features. Turbine exhaust vapor cooling of the stator provides a low thermal gradient packaging concept, but under detailed analysis it may prove to be inadequate to ensure required stator bore and rotor temperature. System considerations are also quite valid in determining the cooling approach. Vapor cooling increases radiator-condenser requirements, while liquid cooling can be construed as boiler preheating.

An attractive packaging concept is shown in Figure 13. This would utilize Sunflower, CRU IVM and CRU V technology to achieve a weight reduction, but may compromise several of the design criteria previously stipulated. Figure 13 should not be construed as depicting a preferred concept nor as a conceptual design.

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V. HIGH POWER PRELIMINARY CONCEPTS

TRW conducted an analytical study during the latter quarter of Fiscal Year 1964. The effort was to investigate higher power turbo-generator concepts in order to establish weight and efficiency parameters. These were then used by AI to conduct a systems evaluation oriented to selecting the next level of power to develop. No finalized conceptual design or design selection was attained, since this was not the object of the study. However, the possible approaches were screened and the following provisional concept characteristics were selected for more detail examination:

Power	15 KWe	25 KWe
Voltage	120/208	120/208
Frequency	400 cps	400 cps
RPM	24,000/12,000	12,000/24,000
Poles	2/4	4/2
Alternator Efficiency, N_a	.88/.89	.90/.88
Pump Power, KW	1.40/1.20	1.38/1.06
Bearing Losses, KW	1.90/1.00	1.03/1.90
Machine Efficiency, N_m	<u>.52</u>	<u>.57</u>
CRU Weight, lbs	<u>150/190</u>	<u>330/*</u>

A. Alternator Concepts

A preliminary evaluation of candidate alternators was made to define the most appropriate type. The following paragraphs contain a brief discussion of these various types.

1. Conventional Synchronous

This machine has been utilized for many years in commercial applications, usually at very low RPM. In this machine, the magnetic field is generated by coils wound on rotor poles. Current to generate the magnetic field is normally conducted through the rotor field coils through conductive brushes contacting slip rings mounted on and rotating with the rotor. This machine is compact, light in weight, and can be very reliable provided speeds do not exceed several thousand RPM and periodic maintenance can be provided for brush adjustment and replacement. In the intended application, rotational speeds will be high and the wound field coils are not practical from strength and dynamic balance considerations. In addition, conductive brushes are not environmentally compatible. The conventional synchronous generator is not considered acceptable for the proposed requirements.

*No 24,000 RPM, 25 KWe CRU weight was obtained, since further detailed analysis could not be justified on the limited program after 12,000 RPM. 25 KWe was essentially dictated by rotor stress and pump cavitation considerations.

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2. Rotating Rectifier Synchronous

The conventional synchronous machine can be greatly improved by the use of a brushless exciter system utilizing rectifiers mounted on the rotor. This system eliminates the requirements for brushes and raises the rotational speed capability. Although this design maintains the compactness of the conventional system, the rotating windings still are weak points at high speeds and the radiation environment precludes the use of solid state devices. It also requires the addition of a second generator winding on the same shaft.

3. Flux Switch Alternator

The flux switch alternator consists of a laminated rotor and stator. Two field coils with opposite polarities are located between the two AC windings. The direction of the magnetic flux which links the AC windings depends on the rotor position in the stator. An AC cycle will be completed when the rotor travels one full rotor pitch. The alternator frequency is determined by the number of rotor poles and shaft speed.

$$f = \frac{\text{number of poles} \times \text{RPM}}{60}$$

The advantages of this machine are high reliability, simple construction. It can be used for high shaft speeds.

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The disadvantages are:

- a. Poor voltage regulation. A series capacitor may be used to improve alternator voltage regulation.
- b. High harmonic content in the output.
- c. Only single phase operation is possible unless three separate alternators are used.
- d. A minimum air gap is required to reduce alternator leakage flux. The increase of air gap due to bore seal will reduce the efficiency and considerably increase the alternator weight.

4. Inductor Alternator (Homopolar)

In the conventional 4-pole inductor alternator, the stator consists of two sets of laminations equipped with a conventional AC winding. One set of laminations sees only north poles. The other set only south poles. A magnetic steel housing encloses both sets of laminations and the field winding. All the north poles are located on one end of the rotor and the south poles on the other end.

The magnetic field produced by the field coil travels axially in the magnetic housing to one stack of stator laminations, crosses the gap and enters the rotor south pole. Further, the flux travels axially in the rotor to the north pole, again crosses the gap and returns to the stator.

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The voltage generated in the AC winding is produced by flux which varies only from a maximum to minimum but does not pass through zero flux. Magnetic steel is utilized to only half of its capacity in either a positive or negative flux direction (but not both) and nearly twice as much magnetic material is thereby required. The longer parts result in increased weight and size in addition to a low efficiency. Low efficiency is brought about by three factors: increased copper losses, increased windage losses and increased core losses. All three are a direct result of a large stator and rotor. An important aspect of this type of machine is that the total number of poles are split between each end of the rotor. That is, a 4-pole machine has two poles at each extremity of the rotor and a 6-pole machine has three. Accordingly, dynamic and static balance are inherent in the rotor except when two poles are used.

Advantages are:

- a. The elimination of brushes, slip rings, and rotor windings.
- b. A simple rugged rotor construction allows operation of the alternator at high shaft speeds, but windage may be high.

Disadvantages are:

- a. The stator laminations are not fully utilized and require 100% more stator iron than conventional wound rotor alternator. Therefore, weight is greater.
- b. The increase of stack length requires more copper for the AC windings which increases losses.
- c. In a 2-pole design, the magnetic field will produce an unbalanced rotating radial pull on both ends of the rotor and, therefore, a 2-pole design cannot be used.

5. Bekey-Robinson Alternator

This is a DC excited salient pole alternator having stationary field coils attached to the end bells.

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The flux generated by the field coils passes through the shaft and the poles of the same polarity, crosses the air gap and enters the stator from where it emerges opposite the next pole. After passing radially through the air gap, it enters into the pole face, travels axially in the pole face to the outer rims, from where the field coil housing provides the return path. The north and south poles are on the surface of a smooth cylinder and are separated by non-magnetic inserts.

The advantages of the Bekey alternator are:

- a. Stator laminations are fully utilized; therefore, stator iron and AC winding copper weight is reduced.
- b. The smooth rotor surface helps to reduce windage loss.

The disadvantages are:

- a. Complicated rotor construction is required.
- b. The rotor has poor mechanical strength.
- c. There are two parasitic gaps in the flux path on each side of the rotor.
- d. There is increased rotor leakage flux.

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- e. Shaft critical speed would be reduced by the required field winding location. This is especially true at the turbine end of the rotor.
- f. CRU assembly would be very complicated because of field coil location. Field power leads would be difficult to route.

6. Lundell

The pure Lundell configuration, which has advantages in some applications, is considered undesirable primarily because of higher leakage and the subsequent high weight per KVA which results. The high rotor leakage also has adverse effects on transient performance and voltage recovery times. There is thrust loading on the bearings in this design and this becomes appreciably greater on larger machines.

7. Inductor-Lundell

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The Inductor-Lundell design combines the features of a pure Lundell and an inductor. This type of machine has an inductor section at each end of the shaft and a Lundell section in the center. At each end of the rotor, voltage is induced in the conductor by a magnetic flux alternating from a given value to zero. The plus flux (or north pole) is on one end and the minus flux (south pole) on the other. At each end, the Inductor-Lundell also utilizes only half of the capacity of the magnetic steel. However, in the Lundell section, the voltage in the stator conductors is induced by a flux alternating from plus (north) to minus (south) and the magnetic steel is utilized to full capacity. Combining the Inductor and Lundell characteristics into one rotor allows the relatively high leakage flux of the Lundell section to be utilized in the inductor section and also eliminates parasitic air gaps. The major disadvantages of the design are the higher losses, higher weights and lower efficiencies. This type of design is more difficult to manufacture due to the interweaving of the Inductor and Lundell sections.

8. Permanent Magnet

The permanent magnet alternator is brushless and has no rotating windings. However, permanent magnet alternators have poor regulation unless combined with some form of booster generator to provide voltage control. The magnets operate at low flux density and force the rotor design to be unusually large, resulting in severe mechanical problems. The permanent magnet in the rotor can also produce significant side-bearing loads.

The reluctance, hysteresis, and induction alternators, although brushless, are capable of supplying real power only and must receive reactive power for excitation purposes from the AC lines, thus requiring parallel operation with synchronous alternators or excitation from a separate supply.

9. Static Field AC Salient Pole (NADYNE or BEEM)

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A combination of the benefits of a synchronous machine and the inductor machines can be achieved in a design known as the NADYNE or BEEM. The high speed, high temperature, mercury atmosphere, long life and high reliability requirements can be met by this design which has static conductor windings (sealed from the mercury coolant) and a solid rotor of high strength material which requires no coolant. An important characteristic of this concept is that it is inherently balanced both dynamically and magnetically for any number of pole pairs. Because of this, no limitation exists on speed found in other possible approaches discussed above.

The NADYNE or BEEM alternator consists of a conventional synchronous alternator stator (the iron and conductors are utilized 100%), a stationary DC excited field winding, and a magnetic circuit which includes a rotating gap and direct flux field poles for generating the output voltage. The magnetic field circuit results in a conventional synchronous alternator stator design while certain other types require essentially a double stator, one section for north pole generated voltages and one for south pole generated voltages. The magnetic field circuit of this alternator results in minimum leakage fluxes due to its configuration and arrangement of field excitation coils. The magnetomotive force generated by a north or south pole is only that required for a single generator air gap, while other types of alternators require the magnetomotive force of a north or south pole to be that required for two generator air gaps in series. This configuration results in a short stator, a magnetic flux path longer than that on a synchronous machine, but shorter than that on an inductor machine. As the windage loss is less, efficiencies will be higher than those of a conventional synchronous machine, and the machine will be lighter, smaller and more efficient than the inductor types. Field excitation coils of this alternator may be designed to be concentric or wound around separate DC excitation poles, whichever is selected as being the optimum for a specific alternator design.

The several generator types are summarized in Table I with pre-design analysis estimated weights for 15 to 25 KWe capability.

The BEEM alternator was selected as the most suitable for this application because of:

- a. Reliability
 - low DC winding temperature
 - low rotor temperature
 - simple rotor design
 - high shaft critical speed
- b. Performance
 - low windage
 - 100% iron utilization
 - low pole face losses
 - low AC winding I^2R loss
 - high efficiency
 - low weight

TABLE I

SUMMARY OF GENERATOR TYPES

Generator Types	Rotating Field Winding	Excitation	Brushes	Generates Reactive Power	Approximate Weight (lbs)
Salient Pole	Yes	DC slip rings	Yes	Yes	95
Non-salient Pole (turbogenerator)	Yes	DC slip rings	Yes	Yes	95
Rotating Rectifier (1 or 2)	Yes	DC rectified from rotating exciter arm	No	Yes	100
Inductor	No	DC	No	Yes	140
Lundell or Inductor-Lundell	No	DC	No	Yes	130
Bekey-Robinson (form of Lundell)	No	DC	No	Yes	120
Permanent Magnet	No	No	No	Yes	
Salient Pole Inductor (NADYNE or BEEM)	No	DC	No	Yes	110

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B. BEEM Alternator Considerations

1. Rotor

TRW's experience in BEEM alternator applications has been for the most part limited to relatively low temperature operation. Machines are currently being operated at 350°F. Since the alternator rotor temperature for a high power CRU in MRPCP will be close to 800°F, considerable study was required to insure that the rotor design could meet the requirements. Since the alternator rotor is an integral part of the CRU shaft, it is necessary that for the full 10,000 hour required life-time that the rotor have maximum dimensional stability so that unbalance of the rotating shaft will not develop.

Therefore, for conservative design, the alternator rotor stresses should not exceed 0.1% creep in 10,000 hours. This figure is commonly used in commercial turboalternator machinery and was felt to offer the guarantee that the CRU would have sufficiently good balance at the end of 10,000 hours. The actual magnitude of creep is not the thing of most concern in the design of the rotor. Creep occurring in a high temperature rotor will not be symmetrical. Therefore, a small amount of creep could cause a sizable amount of rotating load.

The stress in the alternator rotor is a factor in choosing the shaft speed. Figures 3 and 4 illustrate the conditions that define the shaft speed. If the creep strength of the rotor material is known, these curves will tell whether shaft speed will be 12,000 RPM or 24,000 RPM. Stresses shown in Figure 3 are simple hoop stresses; therefore, a detail rotor design may slightly alter the results from this curve.

Independent work in progress at TRW on the high temperature magnetic properties of various materials pointed to AISI 4340 as desirable material. Therefore, with AISI 4340 as the magnetic material, a search was made for the spacer material which had the most compatible coefficient of linear expansion. Stress analysis of rotor assemblies with various materials clearly showed that for most material selections the maximum stresses developed in the rotor came from differential expansion between the two rotor materials. The spacer material finally chosen was Haynes 25.

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FIGURE 3

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ALTERNATOR SHAFT SPEED
VS
OUTPUT AT CONSTANT ROTOR SURFACE VELOCITY

OUTPUT - KVA

FIGURE 4

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10,000-HOUR CREEP STRENGTH OF SAE 4340 AT 800°F

<u>Total Plastic Strain</u>	<u>Extrapolated 10,000-Hour Creep Strength (psi)</u>		
	<u>Min.</u>	<u>Ave.</u>	<u>Max.</u>
0.1%	11,000	14,500	17,500
0.2%	approx. 18,700		

FIGURE 5

ROOM TEMPERATURE MECHANICAL PROPERTIES OF ROTOR MATERIALS

Specimen and Treatment	0.2% Yield Strength (psi)	Yield Point (psi)	Ultimate Tens. Strength (psi)	Reduction in Area (%)	Location of Fracture
SAE 4340 - Double Tempered at 1250°F	—	121,600	124,400	41.6	SAE 4340
SAE 4340 - Double Tempered at 1250°F; Stress-relieved at 600 and 1100°F	—	120,600	125,100	51.0	SAE 4340
Indiana Gear Weld No. 3 Haynes 25 to SAE 4340 (Test Bars - 0.160 inch dia.)	45,800	58,800	67,300	66.2	SAE 4340
TRW - Research Lab Weld Haynes 25 to SAE 4340 (Test Bars - 0.505 inch dia.)	82,200 ^(a)	103,500	115,200	14.1	Haynes 25
TRW - Production Weld Haynes 25 to SAE 4340 (Test Bars - 0.505 inch dia.)	82,700 ^(a)	104,800 ^(a)	114,500	55.6	SAE 4340
(a) Haynes 25					

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To proceed on a rotor design, it was necessary to know the creep strength of AISI 4340 since it was known that Haynes 25 had the higher creep strength of the two materials. Considerable confusion exists in the literature for 0.1% creep in 10,000 hours at 800°F. The literature shows strength values from approximately 10,000 psi to in the vicinity of 40,000 psi.

Within the scope of the MRPCP advanced technology studies, creep tests were initiated in May of 1964 on AISI 4340 in the heat treat condition that could be used in the rotor. The creep and strength data collected to date is shown in Figures 5 and 6.

It was concluded fairly early to utilize the electron beam weld technique to join the rotor pieces. This technique offered minimum weld affected zone and also the minimum of mix of the two rotor alloys. Because of the weld depth necessary for rotor strength electron beam offered the greatest chance of a reliable joining process compared with tungsten inert-arc welding, the only other available state-of-the-art process.

In order to determine the properties of an electron beam weld between the rotor materials, specimens were fabricated for mechanical property evaluations. The test results are shown in Figure 6. If funds permit next fiscal year, creep rupture and creep testing will be conducted on the weld specimens.

However, for the purposes of design analysis and before the above data was available, it was decided to use the "best" value in the literature for AISI 4340 creep data at 800°F. This was 16,000 psi. Figure 3 is intended to be a reference as to the stress effect of shaft speed and rotor diameter based on simple hoop stresses in an alternator rotor. CRU IVM and CRU V design point is shown. A dotted line representing 400 ft/sec rotor surface speed is also shown which indicates a stress level of approximately 19,000 psi. The 24,000 RPM, 15 KW CRU is shown on this plot as well as a 12,000 RPM, 15 KW machine. Also shown is the 25 KW, 12,000 RPM point.

Figure 4 shows the relationship between shaft speed and alternator rating in KVA for various rotor surface velocities. Assuming a conservative limiting rotor surface velocity of 400 ft/sec, which is felt to be very reasonable, then it suggests that 15 KW is the maximum alternator rating at 24,000 RPM. Above 15 KW, the shaft speed should be 12,000. It should be remembered that 400 fps surface speed is based on conservative stress levels suitable for at least 10,000 hours life.

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In Figure 3, it can be seen that the 15 KW, 24,000 RPM rotor will have the highest stress levels of the three machines considered. It also has certain other potential problems inherent in the 2-pole design which tended to make the successful design of a 2-pole alternator rotor more challenging. The configuration of the 2-pole design causes dynamic unbalance of the rotor. This is due to the fact that the density of the material of the symmetrical spacer is considerably higher than the density of the pole pieces. The approach to compensate for this dynamic unbalance was to put holes in the end faces of the non-magnetic separator.

2. Bore Seal

The BEEM alternator rotor offers more inherent flexibility for the application of different types of bore seals than does the permanent magnet alternator used in CRU V and CRU IVM. A major reason for this is the fact that the magnetic air gap between the rotor surface and the ID of the lamination stack of the stator can be designed to have as much as 1/8 inch radial clearance. The two most promising bore seal types are: 1) the alumina cylinder with metal bellows connections to the stator proper at each end of the alumina cylinder, and 2) a metallic bore seal formed in a bellows configuration.

One of the potential advantages of utilizing a ceramic tube or metallic bellows bore seal as opposed to the conventional glass or pyroceram currently used on CRU V's is the feasibility of having the bore seal fabricated and utilized as a separate assembly. This allows the bore seal to be developed as a separate component for vibration and thermal shock by utilizing fairly inexpensive simulated alternator frame assemblies. This cartridge concept also may permit easy replacement of bore seals that are damaged in handling or in developmental test of actual CRU's. It is assumed in this concept that the bore seal is weld sealed at each end of the bore of the cylinder and possibly cemented with a suitable material to the ID of the alternator lamination stack. This latter cementing is required to remove the heat of magnetic losses that are inherent in metallic bore seals. The ceramic bore seal which will not have losses develop within the seal will not require cementing to the lamination stack.

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C. Turbine Performance

An analytical survey was made of turbine performance to be expected in a high power CRU. The ground rules for the turbine design were strict adherence to previous MRCP designs. Therefore, the turbines used in the three CRU analyses in this report were two-stage impulse. This was due to the level of effort allowed by the limited high power CRU program. It is recognized that any follow-on program to develop a discrete new power level machine should not be restricted to present MRPCP turbine types. It is pertinent to reiterate that the study objective was to establish CRU weight and efficiency characteristics which could then be confidently factored into a system evaluation.

Table II summarizes the preliminary turbine study results. As can be seen in the figure, a supersonic single-stage turbine (S.S.S.S.) was also analyzed. This was done to explore the potential of other than MRPCP types. Additionally, though the results are not shown here, a five- and six-stage turbine was analyzed. This turbine had full admission on all stages and showed a design efficiency of .73 for a 12,000 RPM, 40 KWe CRU. No CRU design integration or other additional work was done for these turbines since it was beyond the scope of this effort.

It is obvious that the supersonic turbines offer simplicity of construction but raise the question of endurance capability. Endurance may be reduced from greater erosion caused by the higher velocity vapor flow and the higher theoretical moisture. Also, the stresses caused by higher rotor speeds of supersonic turbines may pose design problems for 10,000 hours life.

	CRU V	CRU IVM	Sunflower	Supersonic	Supersonic
<u>Parameter</u>	<u>2nd Stage</u>	<u>2nd Stage</u>	<u>3rd Stage</u>	<u>SCAP</u>	<u>MRP</u>
Mean rotor velocity, ft/sec	390	401	314	571	556
Average design quality assuming equilibrium expansion	.92	.91	.88	.80	.89

As can be seen from this table, the rotor speed is 40% greater than CRU IVM and CRU V, 2% less than SCAP but the moisture level is 1% less than the third stage of Sunflower which demonstrated adequate life.

There is no question that the endurance of turbines for high power CRU can be the same, or better, than present MRPCP. The larger size of the machine may allow moisture separation devices to be used. As turbine buckets increase in size, the sensitivity to erosion is reduced since the critical leading edge becomes thicker. Or the same leading edge thickness can be used to obtain higher efficiency with the same more than adequate endurance capability as present MRPCP.

TABLE II

MRPCP TURBINE ANALYSIS SUMMARY

	CRU V	CRU IVM	15 KW	15 KW	25 KW	35 KW S.S.S.S.
Speed, RPM	36,000	36,000	24,000	12,000	12,000	12,000
Overall Efficiency, t-s	.543	.583	.70	.64	.63	.67
Total Shaft Power, watts	5,130	5,320	23,800	22,400	32,300	42,500
Total Flow, lbs/min	17.8	17.5	57	59	95	110
ΔH Available	30.2	29.7	30.6	30.6	30.6	30.6
Ideal Power, watts	9,440	9,130	34,000	35,000	51,000	63,000
<u>First Stage</u>						
Rotor Velocity, fps	314	350	390	390	405	556
% Admission	30	33	50	25	40	20
U/C	.37	.405	.47	.47	.47	.47
Stage Final Power, watts	2,100	3,130	12,400	12,000	17,600	42,500
Stage Efficiency	.62	.61	.73	.69	.68	.67
<u>Second Stage</u>						
U/C	.55	.47	.47	.47	.47	
% Admission	.93	100	100	50	100	
Rotor Velocity, fps	343	350	390	390	405	
Stage Power	3,030	2,190	11,400	10,400	14,700	
Stage Efficiency	.61	.62	.74	.70	.69	

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D. Bearing Performance

1. Large Diameter Bearing Reliability and Endurance Capability

Turbomachinery units of higher power Mercury Rankine power systems require larger journal and thrust bearings than the ones developed for the MRPCP (SNAP 2). For two specific power levels, 15 KW (20 KVA) and 30 KW's (50 KVA), mercury lubricated bearings were designed by extending the present SNAP 2 bearing technology through the use of dimensionless performance parameters which were obtained from the 5/8 inch diameter bearing development. It is the intent here to discuss and make preliminary predictions of the reliability of operation and the endurance capability of large bearings relative to the 5/8 inch bearings.

Larger bearings can be compared to .625 inch bearings by evaluating:

Load capacity	Cavitation erosion
Power loss	Crudging
Temperature rise	Particle interference
Stability	

Generally, as machine size increases, operating speed has to be decreased for reasons of mechanical strength. A range of operating speeds versus power output is given in Figure 7a.

The approximate journal bearing diameters that are required for the various power levels are shown. These are approximate since they will depend somewhat on bearing type. This graph indicates that as bearing size increases the lower operating speeds must be taken into account. A base for comparison is Reynolds number. This is shown in Figure 7b for a constant clearance modulus of $C_D/D = 0.002$, where:

$$R_e = \frac{\rho U C_D}{2 \mu g}$$

All the proposed higher power turboalternators will have journal bearings operating at increased Reynolds numbers and far into the turbulent regime.

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FIGURE 7a

FIGURE 7b

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Additional differences in operating conditions include rotating load to unidirectional load ratio. Because of balancing ability, lower shaft speed, alternator magnetic loads and increased shaft weights at higher power levels, this ratio decreases. A plot is made of this ratio versus power level that can be expected based on preliminary design (Figure 8a).

2. Load Capacity

To predict the load capacity, the following dimensionless plot was used:

$$\frac{P^1}{\mu N^1} \left(\frac{C_D}{D} \right)^2 R_e^{-.5} \quad \text{vs.} \quad \frac{6 W}{\Delta P} \left(\frac{D}{C_D} \right)^2$$

for eccentricity ratio of .2, .4, .6, and .8.

where:

- $P^1 = W/LD$
- $W = \text{load capacity}$
- $D = \text{diameter}$
- $L = \text{length}$
- $\mu = \text{viscosity}$
- $N^1 = \text{speed, rps}$
- $C_D = \text{clearance}$
- $R_e = \frac{\rho U C_D}{2 \mu}$
- $U = \text{surface speed}$

This plot was based on 5/8 inch diameter bearing tests with constant L/D ratio, an empirical basis. To evaluate the range of application, one inch journal and thrust bearings were tested under MRPCP Advanced Technology funding to determine the validity of bearing design analysis techniques and generality of 5/8 inch diameter bearing tests. The journal bearings were 3-pad compensated with pads and orifices scaled up to give the same degree of compensation as the 5/8 inch. The thrust bearings had nine spiral grooves. The bearing dimensions were:

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Rotating Load
Unidirectional Load

FIGURE 8a

FIGURE 8b

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<u>Journal Bearings</u>		<u>Thrust Bearing</u>	
Diameter, in.	1.00	Outer Diameter, in.	1.00
Length, in.	0.65	Inner Diameter	0.375
C _D , Clearance	0.0024	Annular Orifice Clearance	0.005
Orifice Diameter	0.032	Annular Orifice Length	0.145

Balanced shaft tests were run and data compared to predicted values based on extrapolation by dimensionless plots of 5/8 inch bearing data. Figure 9 shows the load versus eccentricity comparison between data and prediction for three speeds, 18,000, 12,000, and 6,000 RPM. The actual load capacity slightly exceeded the predicted value at higher speeds but was somewhat less at the lower speed. Operation was stable up to 30,000 RPM.

Complete analyses and reporting of the large bearing tests has not yet been obtained, but sufficient information has already been gained to assure that mercury lubricated bearing technology is generally applicable over a regime of sizes, speeds and power levels. Indications from this are that these dimensionless load parameters are usable at least to a bearing diameter of 1.5 inch.

MRPCP (5/8 inch diameter) bearings have an operating minimum film thickness of about 0.0002 inch and a bearing power loss (total) approaching 15 to 20% of net power output. The bearings were designed for the 25 KW unit to operate with a minimum film thickness of about 0.0005 inch, and a friction power loss of only 5% of the net output.

Since the accuracy or precision of fabrication is independent of size in this size range, it can be seen that there will be a considerable improvement for high power CRU's in fabricated alignment relative to the operating minimum film thickness.

In addition, the boundaries of the high power CRU thermal map are the same as previous CRU's (turbine inlet temperature, turbine exhaust temperature, lubricant temperatures), but the available space is greater. Therefore, high power CRU's offer more flexibility in design to avoid distortion effects of the thermal gradients. The designer will simply have more freedom to obtain symmetry and optimum section sizes compared to the previous miniaturized CRU's. The result will be less thermal distortion for high power CRU's than for CRU IV's and CRU V's which, when coupled with the larger minimum film thickness, means better reliability.

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BEARING LOAD vs ECCENTRICITY RATIO

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Figure 9
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Therefore, load on the bearings will be supported safely by the larger units. Reliability from this viewpoint will be better than present MRPCP. The decreased rotating load to unidirectional load ratio does not come into play here because the sum of all loads is used for determining minimum film thickness.

3. Power Loss and Lubricant Temperature Rise

Power loss prediction for journal bearings is very accurate even in the turbulent regime. This is established by computing power loss and modifying by a turbulent friction factor as given by Smith and Fuller. This method correlates very well with the SNAP 2 bearings. The 1.0 inch diameter bearing power loss experimental data has confirmed this accuracy.

Specifically for the 25 KW unit (12,000 RPM concept, 1.5 inch diameter bearing), predicted power loss is 450 watts and flow is 15 lbs/min. Accompanying temperature rise is 52.8°F. This temperature rise is consistent with good design practice for CRU's.

4. Stability

The stability of a rotor-bearing system is a function of bearing stiffness as well as shaft stiffness. Up to about the first critical speed, stiff shaft analysis is accurate. The bearing stiffness is read off the load-eccentricity curves.

For the 25 KW unit, the computations yielded the following results:

- a. For 12,000 RPM concept, operation is below the first mode critical speed.
- b. For 24,000 RPM concept, operation is above the first mode critical speed.

There are no problems expected at the pass-through point of the first critical speed. These bearings have very high damping coefficients so that no large amplitudes are encountered. A prediction of the damping ratio can be obtained from an empirical plot of a dimensionless damping number D versus eccentricity (obtained from the small bearing tests).

$$D = \frac{CCr^3}{\mu_{RL}^3} \frac{4}{\pi} = 2$$

at eccentricity ratio of 0.6.

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C = damping coefficient

Then for a 25 KW unit (alternator bearing)

$$\begin{aligned} C &= \frac{2\mu RL^3\pi}{4Cr^3} \\ &= \frac{2 \times 1.5 \times 10^{-7} \times .75 \times .975^3 \times \pi}{4 \times .0015^3} \\ &= 48.5 \text{ lb sec/in} \end{aligned}$$

The critical damping coefficient is:

$$\begin{aligned} C_c &= 2mW_n \quad \text{where } m = \frac{1}{2} \text{ shaft mass,} \\ & \quad W_n = \text{first critical frequency} \\ &= \frac{2 \times 38}{386} \times \frac{15000}{60} \\ &= 49 \end{aligned}$$

This bearing is critically damped. Therefore, the amplitude of shaft motion at shaft critical will be small. Computations and test data also show that half-frequency instability is also nonexistent.

5. Cavitation Erosion

The most difficult thing to predict about larger bearing performance is its cavitation erosion resistance. The presently accepted model for cavitation in a journal bearing is similar to that in other flow devices. In a device with a flow constriction, cavities were found to form in the region of minimum pressure in the constriction. When the pressure in this constriction is reduced to the vapor pressure of the fluid, according to Bernaulti's equation, the liquid boils and forms voids containing vapor. As the cavities are carried downstream, they meet increased fluid pressure and subsequently collapse. The collapse is very rapid with the development of high local pressure. The resulting high-pressure force is believed to hammer the surface, eventually causing metal to be detached. This form of destructive action is known as cavitation erosion.

Actually, in liquids the inception of cavitation is associated with the growth of nuclei (submicroscopic in size) containing vapor, undissolved gas, or both, which

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are present either within the liquid or in crevices on suspended particulate matter or on bounding walls. Heat transfer effects play an important role in many cases of cavitating liquid.

Another very important factor in the inception of cavitation is the effect of turbulence. Very little data is available on either the magnitude of the turbulent pressure fluctuations or inception in turbulent regions. Some approximate data for critical cavitation number for a series of discs is given by Kermeen, McGraw, and Parkin: "Mechanism of Cavitation Inception and the Related Scale-Effects Problem", Trans. ASME 77, 1955, and is reproduced in Figure 8b. Here, the critical cavitation number σ_i is given versus R_e , where

$$\sigma_i = \frac{P - P_{\min}}{\frac{1}{2} \rho U^2}$$

$P_{\min}$ = minimum local pressure in fully wetted flow

The increase in operating Reynolds number from 5,000 for MRPCP to 15,000 for 25 KW output is negligible (about 10%) as far as incipient cavitation is concerned.

Since erosion will take place where desinent cavitation occurs (i.e., where the pressure collapses the bubble), it is reasonable to assume that regardless of the bearing size the desinent cavitation peripheral area is very small and is not affected by bearing size. Therefore, a larger bearing has less percentage of its total area exposed to cavitation erosion. The ratio of exposed percent areas is approximately inversely proportional to diameter ratios,

$$\therefore \frac{\text{cavitation}_{\text{small}}}{\text{cavitation}_{\text{large}}} = \frac{D_L}{D_S}$$

In addition, cavitation is a cyclic phenomena occurring once every shaft revolution. Therefore, total cavitation damage, being a function of cycles rather than time, will be as the direct ratio of shaft speed, e.g., 12,000/36,000 or 1/3 as many cavitating occurrences in a 25 KW CRU as in present MRPCP units for a given operating time.

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6. Crudding

"Crudding" in liquid mercury lubricated bearings is a very definite but little known factor. It is known that crudding reduces flow through a bearing similar to a decrease in clearance, builds up on stationary as well as moving surfaces although to a lesser degree on moving surfaces, builds up to a limiting value at which time periodic cleaning action seems to take place. However, the mechanism of crudding, the governing parameters, the point of self-cleaning are all unknown at present. Only qualitative reasoning can be used to determine what to expect with operation of larger bearings.

It can be safely assumed that with the larger turbo-alternator units (Mercury Rankine systems), the same type of crud generation will take place, the same concentration will be present, and that the same behavior will be experienced. If this is true, the build-up rate will be the same as in the smaller bearings (in thickness per hour) for the same flow conditions. But since the larger bearings have larger clearances, the crudding effect may be less noticeable for a longer period of time.

The bearings in the larger units, however, operate at higher Reynolds numbers. It can be argued on the basis of a smaller laminar sublayer at higher Reynolds numbers that there is less of a chance for deposit of entrained crud.

Another factor affecting the rate of crud depositing can possibly be cavitation. Cavitation of the lube would leave behind the dissolved crud. Since cavitation is expected to be less, crudding should be less than the CRU V. When the self-cleaning action will begin and whether the flow at this point will be sufficient to carry away the heat remains unanswered. Plugging of the orifices is not expected since it has never been observed with the smaller bearings (with smaller orifices).

7. Particle Interference

Since the orifices and clearances of the larger bearings are also larger, materials, etc. being the same, particle tolerance will be proportionately greater.

A conclusion can be made from the foregoing that large mercury lubricated bearings applied to large CRU'S will be more reliable than MRPCP CRU's. This analysis must be considered preliminary regarding cavitation erosion and crud effects since complete knowledge of these phenomena is unavailable.

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E. Mercury Pump

The significant finding in the study of mercury pumps for high power CRU is the comparison between 12,000 RPM and 24,000 RPM. The 24,000 RPM CRU mercury pump will operate within the impeller cavitation region under the conditions specified. The 12,000 RPM CRU will not.

The useful flow required by the pump was calculated as follows:

Useful flow = boiler flow + lubricant flow + allowance for potential undefined auxiliary uses

$$\begin{aligned} 1. \text{ Boiler flow} &= \frac{\text{CRU output, KW}}{\eta_{\text{CRU}}} \times \frac{1}{\Delta h} \times .948 \times 60 \\ &= 1.94 \frac{\text{CRU output, KW}}{\eta_{\text{CRU}}} \text{ lbs/min} \end{aligned}$$

Previous studies predict:

$$\eta_{\text{CRU}} \geq .5 \quad \text{or} \quad \leq .6$$

Example:

$$\dot{\omega}_{\text{boiler}} = \frac{16 \text{ KW}}{.52} 1.94 = 60 \text{ ppm}$$

2. Minimum bearing flow is based on the requirement that the temperature rise in the bearing not exceed 50°F due to bearing friction alone. This leaves a margin for absorbing additional heat losses from seal leakage and conduction in the lubricant.

$$\dot{Q} = \dot{\omega} c_p \Delta T$$

$$\dot{\omega} = \frac{\dot{Q}}{c_p \Delta T}$$

$$\dot{\omega} = 34.5 Q_L \quad (\text{where } Q_L = \text{bearing losses in KW,} \\ \dot{\omega} = \text{lbs/min})$$

$$\begin{aligned} \therefore \dot{\omega} &= 34.5(1.9) \\ &= 66 \text{ lbs/min} \end{aligned}$$

when total bearing losses are 1.9 KW. This is the minimum flow, therefore, if a plus 20% minus 0% is used.

$$\dot{\omega}_{\text{bearing}} = 66 \text{ lbs/min to } 79 \text{ lbs/min}$$

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3. An auxiliary need for high pressure mercury might be to recharge a restart reservoir, a jet drain pump, etc. In order to provide for an undefined requirement, an allowance of 10 lbs/min is provided.

If the samples chosen here were representative of an actual CRU:

$$\dot{W}_{\text{total}} = 60 + 79 + 10 = 149 \text{ lbs/min}$$

When an actual jet inducer/centrifugal impeller pump system is designed, a problem arises in the 24,000 RPM system. For non-cavitation performance, the 24,000 RPM pump cannot supply more than 140 ppm useful flow and will operate in a slightly cavitating mode. At 12,000 RPM for both 15 KW and 25 KW CRU's, the pump system performance will be non-cavitating, and would be definitely superior.

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VI. 15 KW AND 25 KW CRU CHARACTERISTICS

A. 15 KWe CRU

Figure 10 is a pictorial presentation of the 2-pole BEEM alternator rotor concept for a 24,000 RPM, 15 KW CRU. A very conservative surface speed of 390 fps was used giving a rotor diameter of 3.7 inches and a stack length of 2.0 inches.

A concurrent analysis was made of a 15 KW, 12,000 RPM, 4-pole rotor unit. Parasitic losses would be lower due to lower RPM, but turbine efficiency would also be lower so that overall machine efficiency would be equivalent to the 24,000 RPM unit. The design information is summarized below:

	<u>24,000 RPM</u>	<u>12,000 RPM</u>
Alternator Power (KWe), gross	18.0	18.0
Alternator Efficiency	.88	.89
Alternator Input Power	20.5	20.2
Bearing Power Losses	1.9	1.0
Pump Power	1.4	1.2
Total Shaft Power	23.8	22.4
Mechanical Efficiency		
$\frac{\text{KWe}}{\text{KW shaft}}, \eta_m$	.75	.81
Turbine Efficiency, η_t	.70	.64
CRU Machine Efficiency, $\eta_m \times \eta_t$	.52	.52
CRU Weight, lbs	150	190

Both machines were predicated on the present CRU V design point, i.e., $P_{\text{inlet}} = 115$ psia, $T_{\text{inlet}} = 1150^\circ\text{F}$, and $P_{\text{exhaust}} = 9$ psia.

1. Alternator

Preliminary evaluations emphasized the alternator design and electrical characteristics for two reasons:

- a. The alternator represents the only component which is new to the Mercury Rankine Program integral turboalternator concept.
- b. The alternator is the governing component of the CRU once desired electrical characteristics are specified, i.e., frequency, voltage and power.

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2-POLE ROTOR CONCEPT

24,000 RPM

FIGURE 10

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The selection of the standard 400 cps frequency dictated a rotating speed of either 12,000 or 24,000 RPM with an alternator rotor of 4 or 2 pole construction, respectively. The selection of standard voltages of 120/208 together with the power level and speed determine alternator size, since these characteristics must satisfy the following equations:

$$DL = \frac{V}{NnB}$$

and
$$D^2L = \frac{P}{NB} \left(\frac{D}{nI} \right)$$

where: D = rotor diameter
L = stack length
V = voltage
P = power, KW
N = speed, RPM
n = number of winding turns
B = flux density
I = current, amperes

A summary of 15 KW alternator design analyses is tabulated in Table III.

2. Turbine

The generalized study indicated that the 24,000 RPM CRU weighed about 27% less than the 12,000 RPM CRU. This was due to superior turbine efficiency and compactness. This appeared to be a significant weight advantage. Therefore, the 24,000 RPM unit was selected as the best choice at the 15 KW level and other prime components were scaled from CRU V to fit this alternator. The turbine was scaled using the same work split. There were several general reasons for the significant increase in turbine efficiency over CRU V:

- a. Larger turbine dimensions result in high Reynolds number throughout the turbine resulting in a 2 point improvement over present nozzle and rotor efficiencies.
- b. Shaft leakages tend to remain constant, while flow is increased by a factor of nearly 3. This results in a 3 point improvement in the first stage and over 6 points in the second stage.

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~~CONFIDENTIAL~~TABLE III15 KW ALTERNATOR DESIGN ANALYSES

	24,000 RPM	12,000 RPM
Rotor Diameter, inches	3.7	4.84
Stack Length	2.0	2.437
Alternator O.D.	8.0	9.0
Alternator Length	7.625	8.75
Rotor Surface Velocity, fps	388	253
Weight, lbs	77	100
Magnetic Pull, lbs/bearing	30.5	57
Losses, watts		
AC Winding	707	744
Field Coil	665	630
Stator Teeth	65	94
Stator Core	106	94
Rotor Surface	110	80
Stray Load	100	80
Windage	310	160
Voltage Regulation	100	100
TOTAL LOSSES	2158 watts	1982 watts

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- c. The interstage collector pressure drop will be decreased by a factor of 2 by using a greater axial collector length. This provides nearly a 3 point improvement.
- d. The 24,000 RPM rotor speed allows a more nearly optimum relative rotor Mach number.

3. Bearings

The bearings were scaled from CRU V based on 3-pad design. The analysis gave the following values:

Alternator Bearing

Diameter = 1.25" Power Loss = 1100 watts
 Length = .813"
 Speed = 24,000 RPM
 $C_D = .0025"$
 Load Capacity at 300 psi = 165 lbs with .0004 min. film
 Stiffness = 172,000 lbs/in at 300 psi
 Damping Coefficient = 62 lbs sec/in
 Lube Coolant Flow = 9.16 lbs/min at 300 psi

Turbine Bearing

Diameter = 1.0" Power Loss = 450 watts
 Length = .65"
 $C_D = .002"$
 Load Capacity at 300 psi = 73 lbs with .0004 min. film
 Stiffness = 120,000 lbs/in at 300 psi
 Damping Coefficient = 39.0 lbs sec/in
 Lube Coolant Flow =

Thrust Bearing

$D_o = 1.25"$ Power Loss = 350 watts
 $D_i = 0.50"$
 $TAC = .0025"$
 Length = 0.15"

TOTAL LOSSES = 1.9 KW

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4. Mercury Pump

The pump analysis resulted in design criteria compared to MRPCP machinery as follows:

	15 KW CRU 24,000 RPM	CRU V 36,000 RPM
Specific Speed	1130	1000
Suction Specific Speed	8720	7200
Useful Pump Flow, lbs/min	162	45
Impeller NPSH, psia	28.6	36.0
Power Consumption, watts	1400	285
Jet Nozzle Size, inch	.048	.036
Pump Rotor Diameter, inch	0.734	0.50
Inlet Diameter, inch	0.28	0.20
Number of Vanes	6	6

The pump impeller will operate in the incipient cavitation region. A complete, detailed conceptual design may reveal that the flow requirements used are overly conservative. Since the pump should provide non-cavitating performance at 140 ppm useful flow, a 15% decrease in required useful flow would put the pump in the non-cavitating region.

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G. 25 KWe CRU

Preliminary analyses were also conducted at TRW to evaluate a 25 KWe CRU. After system evaluations indicated that 25 KWe or greater would be a very attractive power level, further and more detailed analyses were made and are incorporated herein. Both 12,000 RPM and 24,000 RPM designs were analyzed as well as variable power level. The preliminary design indicated that a 35 KW alternator could be matched with an appropriate turbine and pump to give 25 KWe. With an insignificant weight penalty, the machine would have future growth potential to 35 KW by applying a non-scaled turbine to the alternator within the same package. This could be done by changing work split, number of stages, or design point, any one of which might prove to be needlessly restrictive if detailed systems analyses so indicate. The following initial design point and machine characteristics were established as preliminary design criteria:

25 KWe Preliminary Design Analysis Criteria

Gross Electrical Power	29.0 KWe
Speed Control and Degradation	2.8
Allowance for NaK Pump	<u>1.2</u>
Net Power	25.0 KWe
Power Factor	0.8 to 1.0
Frequency	400 cps
Frequency Control	± 1%
Voltage (3Ø, Wye)	120 V line to neutral 208 V line to line
Voltage Regulation (0-100% load, 0.8-1.0 power factor)	± 1%
Shaft Speeds	12,000 RPM
Turbine Inlet Pressure	125 psia
Turbine Inlet Temperature	1150°F
Turbine Exhaust Pressure	9 psia
Turbine Flow (approximately)	100 lbs/min
Approximate Bearing and Seal Load	1.88 KW
Approximate Bearing Flow Rate	75 lbs/min
Mercury Pump Suction Pressure Available	6 psia maximum
Mercury Pump Discharge Pressure	500 psia
Mercury Pump Efficiency	40%
Turbine Efficiency	63%
Alternator Efficiency	90%

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1. Alternator

As before, initial emphasis was placed on alternator designs. Figure 11 shows the 4-pole rotor concept chosen for this machine. Two designs were analyzed: the single stator stack BEEM which has already been built and tested for other applications, and a modified BEEM with two stator stacks. The preliminary design alternator characteristics are:

25 KWe Alternator - 12,000 RPM

	<u>Single Stack</u>	<u>Two Stack</u>
Rotor Diameter, in.	6.42	5.10
Stack Length	3.25	5.0
Alternator O.D.	11.25	9.125
Alternator Length	9.50	13.875
Rotor Surface Velocity, fps	336	267
Weight, lbs.	190	170
Efficiency, η_a	.90	.90

The two-stack BEEM may merit further future study, but for preliminary design purposes, the slight weight advantage is more than outweighed by the 50% increase in stack length. This requires greater bearing diameter to maintain shaft dynamics, so parasitic losses are substantially higher. Therefore, the further preliminary design concentrated on the single-stack BEEM.

A further analytical study was made of a 25 KW CRU to operate at 24,000 RPM with a 2-pole rotor design similar to that shown in Figure 10. The alternator preliminary design characteristics are as follows:

25 KWe Alternator - 24,000 RPM

	<u>Single Stack</u>	<u>Two Stack</u>
Rotor Diameter, in.	5.0	4.17
Stack Length	2.5	4.25
Alternator O.D.	10.625	8.625
Alternator Length	9.75	13.75
Rotor Surface Velocity, fps	525	435
Weight, lbs.	180	160
Efficiency, η_a	.88	.88

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4-POLE ROTOR CONCEPT

12,000 RPM

4 POLE ROTOR CONSTRUCTION SCHEMATIC

FIGURE 11

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The 24,000 RPM alternator is unattractive from the high surface velocity requirement. This casts doubt on long life due to excessive creep. The slight weight advantage is compensated by the 2 point decrease in alternator efficiency.

2. Turbine

The scaled turbine design emphasized the attainment of 25 KWe with a very conservative approach. Predicated on a turbine efficiency, $\eta_t = .63$, which is virtually certain of attainment, the 25 KWe power level will be achieved. The turbine efficiency improvement over present CRU V machinery can be achieved because:

- a. Larger turbine dimensions and high pressure levels result in higher Reynolds numbers throughout the turbine resulting in improved nozzle and rotor efficiencies - 2 points.
- b. Shaft leakages tend to remain constant while flow is increased by a factor of five - 3 points - first stage, - 7 points - second stage.
- c. The rotor speed of the second stage will be more optimum to reduce the relative rotor Mach number.
- d. The interstage collector pressure drop will be decreased by a factor of 2 (- 3 points) by allowing a greater axial collector length.

The turbine design parameters are shown in Table IV.

A detailed analysis of the 24,000 RPM, 25 KWe turbine was not done due to the time and funding restrictions and since the alternator appeared to be very questionable at this speed. The 24,000 RPM turbine would be several points higher in efficiency, but the accumulative parasitic losses would essentially equalize this so that machine efficiencies of the 12,000 and 24,000 RPM units would be equivalent.

By increasing flow area or turbine inlet pressure to increase vapor flow and subsequent turbine work, the 25 KW CRU will have future upgrading capability to 30-35 KWe, similar to upgrading of CRU V from 3.5 to 5.0 KWe.

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TABLE IV

25 KWe TURBINE DESIGN CHARACTERISTICS

	12,000 RPM
Overall Efficiency, t-s	.63
Total Power, watts	32,300
Total Flow, lbs/min	95
Δh Available	30.6
Ideal Available Power, watts	51,000
<u>First Stage</u>	
Percent Admission	40
Nozzle Blade Height	.18
Nozzle Axial Chord	.22
Nozzle Efficiency	.93
Rotor Blade Height	.19
Rotor Axial Chord	.25
Rotor Efficiency	.84
Diagram Efficiency	.82
Average Rotor Moisture	.96
Dia. η_s , Moisture Corrected	.79
Dia. η_s , Moisture Tip Leak Corrected	.75
High Pressure Seal Corrected η	.73
Disk Friction, watts	450
Stage Final Power	17,600
Stage Final Efficiency	.68
Δh Ideal	15.6
<u>Second Stage</u>	
Nozzle Blade Height	.25
Nozzle Blade Chord	.25
Nozzle Efficiency	.88
Correction for Interstage Mixing	.98
Rotor Blade Height	.27
Rotor Blade Chord	.25
Rotor Efficiency	.75
Stage Diagram Efficiency	.75
Average Rotor Moisture	.92
Moisture Corrected	.69
Disk Friction, watts	140
Power	14,700
Δh Ideal	14.4

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3. Bearings

Bearing analysis established the following bearing sizes and power losses:

	<u>12,000 RPM</u>	<u>24,000 RPM</u>
Alternator Bearing		
Diameter, inch	1.5	1.25
Length, inch	.975	.813
C _D , inch	.003	.0025
Power Loss, watts	430	1100
Turbine Bearing		
Diameter, inch	1.25	1.0
Length, inch	.813	.65
C _D , inch	.0025	.002
Power Loss, watts	200	450
Thrust Bearing		
D _O , inch	1.5	1.25
D _I , inch	.56	.50
TAC	.003	.0025
Power Loss, watts	400	350
TOTAL LOSSES		
	1030 watts	1900 watts

Figure 12 indicates the critical speed versus bearing stiffness analysis of the 25 KW, 12,000 RPM machine showing the unit will operate well between the first and second critical speeds.

4. Pump

The pump analysis resulted in design criteria and comparison to present machinery as shown in Table V. The scaling of the mercury pump presents no problem. Suction specific speed and NPSH are lower for 12,000 RPM giving far greater margin to cavitation, and machining is much easier due to increased size. The 24,000 RPM pump would have less margin to cavitation than CRU V because of its high specific speed.

The 25 KWe CRU and possible future upgraded unit characteristics which resulted from these analyses are summarized in Table VI.

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25 KWe CRU CRITICAL SPEED VERSUS BEARING STIFFNESS

FIGURE 12

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~~CONFIDENTIAL~~TABLE V25 TO 35 KWe MERCURY PUMP DESIGN CRITERIA

	25 KWe 24,000 RPM	25 KWe 12,000 RPM	35 KWe 12,000 RPM	CRU V 36,000 RPM
Specific Speed	1230	615	710	1000
Suction Specific Speed	9540	4770	5400	7200
Useful Pump Flow, lbs/min	170	170	230	45
Impeller NPSH, psia	31.8	14.6	16.5	36.0
Power Consumption, watts	1060	1380	1550	285
Jet Nozzle Size, in.	.050	.050	.100	.036
Pump Rotor Diameter, in.	0.75	1.50	1.50	0.50
Inlet Diameter, in.	0.30	0.30	0.346	0.20
Number of Vanes	6	6	6	6

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~~CONFIDENTIAL~~TABLE VI25 TO 35 KWe CRU CHARACTERISTICS

	25 KWe	35 KWe
Shaft Speed, RPM	12,000	12,000
Alternator Losses at 90% and 92% Efficiency, watts	2,800	3,200
Bearing Power, watts	1,030	1,030
Pump and Seal Losses, watts	1,380	1,550
Shaft Power, watts	32,300	42,500
Turbine Efficiency, η_t	.63	.67
Required Turbine Flow, lbs/min	95	110
Mechanical Efficiency	.83	.85
CRU Machine Efficiency	.57	.59
CRU Total Weight, lbs	330*	330

*Total CRU weight predicated on utilizing 35 KWe alternator.
Designing to 25 KWe alternator limit will save $\approx$ 10 pounds.

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Figure 13 depicts a possible alternate configuration to the CRU V type for a higher power CRU, in this case the 25 KWe design. This machine has desirable features which have been performance verified by CRU V, CRU IVM, and CRU V Sunflower testing.

- a. Internal component geometries are scaled from CRU V.
- b. The turbine end housing employs the Sunflower design which had no NaK pump and the best bearing support configuration.
- c. The main housing provides for either vapor or liquid cooling, whichever appears most attractive from systems considerations.
- d. The mercury pump is scaled from CRU V with the jet pump external to the housing.
- e. No new manufacturing processes or fabrication techniques are required.

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FIGURE 13

TYPICAL CONFIGURATION OF A 25 kwe CRU

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