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EXPERIENCE WITH TURBINE-DRIVEN POTASSIUM BOILER-FEED PUMPS
FOR THE MEDIUM POWER REACTOR EXPERIMENT*

L. V. Wilson
Design Engineer, Oak Ridge National Laboratory
Oak Ridge, Tennessee

H. C. Young
Development Engineer, Oak Ridge National Laboratory
Oak Ridge, Tennessee

A. M. Smith
Development Engineer, Oak Ridge National Laboratory
Oak Ridge, Tennessee

ABSTRACT

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A series of potassium vapor turbine driven boiler feed pumps with potassium lubricated bearings has been designed, built and tested for a total of 2000 hr of operation at high temperatures. This paper summarizes the problems that have arisen in the course of the design, fabrication, and operation of these units. Approaches to the thermal expansion problems, provisions for lubricant flow to the bearings, speed pickups, and maintenance of moving parts installed in hermetically sealed systems are discussed. Test experience with the units operated with steam and water is related to the test experience obtained in potassium systems. The problems encountered in fabricating tungsten carbide parts, difficulties both with the brittleness of TZM molybdenum alloy and its tendency to gall and seize, together with solutions that have been found effective for coping with these problems are outlined.

Introduction

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One of the basic precepts for the Medium Power Reactor Experiment (MPRE) Program was that, to obtain a system with a high degree of reliability, it is essential that a potassium vapor turbine-driven boiler feed pump using potassium lubricated bearings be developed. Work at Knolls Atomic Power Laboratory** had shown that tungsten carbide running against tungsten carbide in NaK was very resistant to scuffing, and hence should

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**"Materials Compatibility Report," compiled by Bearing and Oil Seal Branch of the Bureau of Ships, USN.

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be suitable for withstanding the boundary lubrication conditions that prevail in startup and shutdown in high temperature potassium. Since this point raised a basic question of materials feasibility, it evidently deserved attention early in the development program.

Other feasibility questions included turbine bucket erosion by wet vapor and cavitation damage to the impeller of a pump operating continuously in cavitation. It was clear that much insight into all of these problems could be provided by the operation of a small potassium Rankine cycle system with a free turbine-driven boiler feed pump, hence work was initiated on the smallest size of system that promised to be meaningful. The turbine pump, referred to as the ORNL turbine pump, for this system was designed and produced by ORNL. For two following larger systems called the intermediate (IPS) and large potassium systems (LPS), two other units, referred to as the Aeronutronic Mark I and Mark II turbine pumps, were designed and produced in conjunction with the Aeronutronic Division of the Philco Corporation.

A number of problems are inherent in the design of any piece of rotating machinery and, as we all know, these problems are compounded in rotating machinery for use in liquid metals. Quite often a problem that should have been recognized and resolved in the early design stages is either overlooked or its importance is underestimated, a single operating condition is specified instead of a realistic range of operating conditions, or some rationalizing assumptions are made that evade a problem. In most cases it is probably more realistic to build a prototype unit, knowing that some problem areas may be overlooked, test the unit in low temperature liquids, and make the necessary modifications before going into the high temperature hardware. Unfortunately, the schedules for other components often require simultaneous fabrication of low temperature prototype and high temperature hardware. Thus although most of the basic development problems in the turbine-pump units showed up during water tests, it was not always possible to make the implied modifications in a potassium unit phased only a little behind the water prototype.

Material Selection

Our design philosophy has been to make the mechanical design of the unit as simple as possible and yet obtain a reliable set of bearings. In approaching the bearing design

problem, tungsten carbide with 12% cobalt binder was selected for the mating bearing surfaces on the basis of the previously mentioned tests by Knolls Atomic Power Laboratory. Its use presents some major design problems since its coefficient of thermal expansion is 3×10^{-6} in/in-°F as compared to 9×10^{-6} in/in.-°F for the stainless steel used in the rest of the system. While it is difficult to design for use of an austenitic stainless steel rotor running in tungsten carbide bearings, TZM molybdenum has a coefficient of thermal expansion very close to that of tungsten carbide with 12% cobalt binder and is much stronger at high temperature than the austenitic stainless steels. Hence it was decided to match the coefficient of thermal expansion of tungsten carbide to that of a molybdenum alloy rotor in the region where clearances are most vital, i.e., in the bearings, and make use of other provisions in the less sensitive zone between the bearing supports and the casing. The use of a molybdenum rotor offered the additional advantage that molybdenum is one of the very few materials that shows promise for operation at 2000°F in potassium vapor.

Bearing Tests

The first step in implementing the development program was the operating of a bearing test rig that had been used in the Molten Salt Reactor Development Program. The most expeditious approach to the use of this rig was to mount a stainless steel journal on the existing Inconel shaft and run it against a tungsten carbide sleeve bearing. This test consisted of a series of cycles in which the shaft was started up under light load, the load was then manually increased as the speed was increased, then full load and speed were maintained for a period of an hour after which the load and speed were proportionately reduced and the shaft was brought to a stop under the original starting load. In one test, a plain stainless steel journal running against a tungsten carbide bearing in 800°F potassium showed very little change in the power required to drive the shaft after 160 cycles. On disassembly the journal was moderately scuffed. The test was repeated with a chrome-plated stainless steel journal running against a second tungsten carbide bearing. This test was run for 450 cycles at which point it was decided to shift to a continuous endurance test. The test bearing was still operable at shutdown after a total operating

time of 4085 hr, and the journal and bearing were both in excellent condition.

ORNL Turbine Pump

Concurrently with the above work on bearings, work was initiated on the construction and operation of a small potassium vapor turbine-driven feed pump employing a TZM molybdenum rotor with tungsten carbide journal bearing sleeves brazed to the shaft, and tungsten carbide sleeve bearings mounted in a stainless steel housing with provisions to accommodate differential thermal expansion between them. This pump, shown in Figs. 1 and 2, is a fairly small unit having a turbine wheel diameter of 2.75 in. and an impeller diameter of 1 in. No effort was made to design the impulse turbine for good efficiency because more than adequate vapor flow was available from the boiler. The flow passages of the pump impeller are 7/64 in. diameter drilled holes which discharge into a fully concentric collector. Design and operational criteria for the turbine pump are shown in Table 1.

The rotor is supported on two potassium-lubricated, full-journal bearings each having a diameter of 0.5 in., a length of 0.25 in., and a radial clearance of 0.0004 in. The axial loads are carried by radially-grooved, flat-plate thrust bearings. Unfiltered potassium is fed directly into the bearings from the collector. The turbine wheel, pump impeller, and shaft are fabricated from TZM molybdenum (0.5% Ti, 0.08% Zr, 0.03% C). The tungsten carbide journals are brazed to the TZM molybdenum with a nickel-boron-silicon brazing alloy. In this instance, differential thermal expansion problem was handled by mounting the tungsten carbide bearing sleeve on stainless steel cantilever beams designed to be radially "soft" but translationally stiff.

The first step in the development program for this small unit was to build a unit of conventional materials for test in a water-steam system. This was done not only to check out the performance of the pump but also the problems of "bootstrapping" a free turbine-driven feed pump of this sort in a hermetically sealed system and the general problems of stability and control that might be associated with the system. Performance tests on this prototype unit were made in an open steam system (not in a closed Rankine cycle) in order to determine some of the operating characteristics. Fig. 3 shows the head-flow curves for several speeds. The flatness of the curves stemmed from the use of a concentric

collector with a rather generous flow area. Typical performance curves for the turbine-pump unit for various turbine inlet pressures are shown in Fig. 4. By obtaining performance data in this manner it is possible to analyze the problem of the pump performance relative to the Rankine cycle requirements for a range of system power levels.

Several problems arose during the developmental water tests of the unit. It was necessary to alter the bearing lubricant paths in order to assure adequate lubricant supply and to control lubricant leakage to the turbine cavity. The outside diameter of the impeller suction shroud was reduced to reduce the axial forces on the thrust bearing. Originally the turbine pump had only unidirectional thrust capability; this arrangement was found to be inadequate for a wide range of system operating conditions. A thrust bearing for carrying reversed loads was added.

Because of the smallness of the unit and the high rotational speed it was found to be desirable to maintain extremely close quality control on the bearing surfaces. The squareness and eccentricity of these surfaces was held to about .0001 TIR or better - usually much better. We are not too sure about the importance of the bearing surface finishes, which were held to about 4 rms, since corrosion, erosion, and the abrading effects of starting and stopping probably affect the finish.

Upon completion of the prototype tests in water, the first of the turbine pumps was installed in a small potassium system. Potassium vapor from the boiler and separator passes through the turbine throttle valve, the turbine, and into the condenser. The condensate enters an inert gas trap where any non-condensables may be removed. Liquid potassium is subcooled downstream of the trap before entering the pump suction. Liquid from the pump discharge is then fed back through the preheater to the boiler. A parallel boiler feed line containing an Electrodynamic pump is used to supply the boiler feed during the system startup and is valved out of the system once the turbine pump is in operation. Procedures for startup and operation of the system are discussed in more detail in a companion paper* presented during this conference.

*R. E. MacPherson, "Potassium Rankine Cycle Operating Experience for the MPRE." Proceedings of the First Annual Rankine Cycle Space Power System Specialists Conference, AIAA, Cleveland, Ohio, October 26-28, 1965, (Confidential).

After a turbine pump is installed, the entire system is evacuated, flushed with a cleaning charge of potassium, and then filled with the operating charge of potassium. The boiler feed is supplied by the Electrodynamic pump until the desired system conditions are obtained, at which time the turbine pump is brought onto the line.

A typical cavitating condition will have the pump oscillating at a frequency of about 10 cycles per second between a 35% and 75% cavitation coefficient (C) which is defined as

$$C = \left[\frac{P_H - (P_d - P_c)}{P_H} \right] \times 100$$

where P_d = pump discharge pressure, psia

P_c = condenser pressure, psia

P_H = non-cavitating head capability of pump, psi.

A cavitation coefficient of 75% means that the pump is providing 25% of its non-cavitating head at that speed. None of the impellers have shown any evidence of cavitation damage to date although most of the test durations were short.

A total of eight ORNL turbine pumps have been installed and tested in the facility. A summary of the individual tests is shown in Table 2. Of the eight units installed, seven were successfully started and operated in potassium for various time intervals ranging from 15 minutes to 327 hr before it became necessary to terminate the test. One turbine (Unit No. 4) could not be started after installation. During the 327-hr test period of Unit No. 6, one continuous run of 147 hr was obtained without any adjustments being made to the system. Total operating time on the seven units tested has amounted to 540 hr with 129 starts and stops. All of the 72 starts for Units 1, 2, and 3 were "bootstrap" starts; i.e., without the use of the Electrodynamic pump.

Failure of the threaded portion of the molybdenum shaft on Unit No. 8 has led to the design of elastic washer of Cb - 1 Zr alloy that will be used on future units. The purpose of this washer is to maintain a constant axial force on the turbine wheel and thus prevent the possibility of over-tightening the retaining nut during installation.

After several unsuccessful attempts were made to operate Unit 4, particulate matter was found to have locked a thrust bearing. Various attempts at filtering the potassium were not successful in succeeding units, since particulate matter apparently caused bearing seizure in Units 5 and 6. It would have been desirable to filter the bearing lubricant supply, but because of the internal flow path from the impeller periphery, this was impossible without a major redesign of the turbine pump.

Aeronutronic Mark I Turbine Pump

When the preliminary tests of the steam-water prototype of the ORNL turbine pump turned out favorably, the design of a new model of turbine pump suitable for a larger potassium system was roughed out and the design requirements were defined. Since we were very short of design personnel at the time, and since we wished to take advantage of the government investment in the development of mercury and potassium vapor turbines, a set of performance specifications was prepared and sent to the organizations working in this field. As a result of this solicitation a contract was placed with the Aeronutronic Division of the Philco Corporation for the fabrication of one unit for operation on steam and water and two units for operation on potassium. The contract did not call for any guarantees for operation on potassium; ORNL assumed full responsibility for this phase of the work.

The Aeronutronic Mark I turbine pump (see Figs. 5 and 6) was designed to meet the requirements of an intermediate boiling potassium system. It has a turbine wheel diameter of 6.3 in., an impeller diameter of 2.75 in., and operates at a design speed of 8000 rpm. The turbine is a single-stage, partial-admission, impulse type driving an open faced pump impeller. Design and operational data for this turbine pump are shown in Table 1. The rotating component is supported on two full journal bearings each having a diameter of 1.0 in., a length of 0.5 in., and a radial clearance of 0.0017 in. The thrust bearings are of a hybrid type - being pressurized at the inside diameter and outward through radial grooves extending into the flat thrust face. The grooves are dammed at the outside diameter to prevent excessive through-flow of potassium into the turbine cavity. The pads between the grooves generate hydrodynamic films.

Potassium for lubricating the bearings is taken from the pump discharge line and passed through a 10-micron sintered metal filter before entering the bearing housing. The flow enters an annulus between the two journal bearings where it is divided in two and passes through each journal bearing-thrust bearing combination in series and thence into the turbine cavity and the pump cavity.

The materials are identical to those selected for the ORNL turbine pump; i.e., the shaft, turbine wheel, and impeller are fabricated from TZM molybdenum and all bearing surfaces are tungsten carbide (12% cobalt binder). The journals are ground for a sliding fit on the molybdenum shaft rather than being brazed to the shaft as was done on the ORNL turbine pump. The resulting multiplicity of fits was not conducive to obtaining and reproducing good concentricity and squareness at assembly.

Differential thermal expansion between the turbine housing (type 304 stainless steel) and the bearing housing (molybdenum) was taken care of by allowing the bearing housing flange to slip under the heads of the bolts that hold the bearing housing in place. This was later felt to be unsatisfactory due to (1) the lack of control of slip forces, (2) the lack of concentricity of the rotary element within the housing, and (3) the possibility of damage to the mounting flange and static seal resulting in excessive flow of potassium into the turbine cavity.

Water tests were run that indicated an excessive thrust load on the turbine end thrust bearing. The pressure differential between the turbine cavity and the pump suction for a zero axial force on the thrust bearings was deduced from the rate of change in the flow of bearing lubricant into the turbine cavity as a function of this differential pressure. Fig. 7 shows typical curves of this leakage flow versus the differential pressure. The intersection of the two portions of the flow rate curve was interpreted as the differential pressure at which the axial thrust reversed directions. Vanes were machined into the back of the impeller to reduce the pressure there and consequently to reduce the thrust load to an acceptable level. Unfortunately, time did not permit this change to be incorporated in Unit No. 1, the first unit that was operated in potassium. It was, however, incorporated in Unit No. 2.

The turbine vapor supply and discharge lines enter the turbine cavity through the cover flange which has the turbine nozzle ring mounted to its inside surface. This design was found to be quite inconvenient because it required cutting and rewelding the vapor lines in order to maintain the turbine pump rotary assembly.

Typical water test pump performance curves are shown in Fig. 8. Fig. 9 shows typical water cavitation data which agrees with predicted values for the required NPSH at the point of incipient cavitation.

Vibration measurements on the Mark I turbine pump housing taken during water tests revealed a large amplitude at a frequency corresponding to one-half the shaft speed. By raising the water temperature from 80°F to 200°F and thus reducing the viscosity of the water lubricant to one-third of its original value, the amplitude of vibration was drastically reduced at 6000 rpm. The reduction in amplitude was much less at the design speed of 8000 rpm. This prompted a serious consideration of bearing half-speed whirl. As a result, Unit No. 2 of the Mark I design was modified by reducing the weight of the turbine wheel in order to better distribute the bearing loads. Also a more sophisticated analysis of the bearing characteristics and rotor dynamics was made for the Mark II design.

The intermediate potassium Rankine cycle system was not specifically designed to obtain turbine pump performance data; however, since an electromagnetic pump was used for system startup, it was possible to operate the turbine pump in a closed loop, comprised of the condenser jet pump recirculation system, and obtain some non-cavitating pump performance on one flow resistance line. These data correlated very well with water test data and increased the confidence in predictions of the cavitation coefficient.

Unit No. 1 of the Mark I design operated for a total of 18 hr and was subjected to a total of six starts and stops. The bearing drag, as measured by the turbine nozzle differential pressure required for starting, became progressively higher with each startup. The difficulty stemmed from having the vapor throttling valve in the turbine exhaust line because this gave a high bearing load under starting conditions.

Unit No. 2 basically had two modifications, both of which have been mentioned previously, that would affect the operation of the unit. The turbine wheel was lightened to improve the loading conditions for the journal bearings and vanes were machined into

the back of the impeller to reduce the thrust bearing load. In addition, the system was modified to locate the vapor throttling valve in the turbine inlet line.

Since being placed in operation, Unit No. 2 has accumulated a total of 1400 hr of operation with 9 starts and stops. Table 3 shows typical operating conditions for the turbine pump over a range of system operating conditions. Notice that, for non-cavitating operation, the pump suction pressure is relatively high, while for cavitating operation the pump suction pressure is reduced.

Speed measurements of the potassium units are made with a high temperature electromagnetic pickup made by Electro Products Laboratories and rated for 1000-hr of operation at 800°F. The interruption of its magnetic field is provided by small slugs of Hyperco 27 located in the impeller outside diameter. Another slug of Hyperco 27 is brazed into the turbine pump housing to enhance the magnetic coupling between the pickup and the rotating Hyperco slugs. The pickup is connected to an appropriate amplifier and counter. Its use has provided satisfactory speed measurements at about 700°F. However, after 1000 hr, there are indications that the generated voltage had decreased more than 50%.

Aeronutronic Mark II Turbine Pump

Extensive test experience with the seven ORNL turbine pumps and two of the Aeronutronic Mark I units made it clear that a new design should be prepared based on the Mark I design but with numerous modifications. A new contract was negotiated with Aeronutronic for the redesign work and the fabrication of the stainless steel and TZM molybdenum parts for five turbine pumps. We are procuring tungsten carbide parts in rough form from vendors and are having them finished to meet the tight dimensional tolerances and surface finish requirements that we have found to give satisfactory performance. These five units have been designed so that they can be used interchangeably in any one of the three potassium Rankine cycle systems of widely different sizes employed in our development program. This approach has the advantage that it cuts the cost for the engineering design, procurement and development, as well as the costs associated with the inventory of spare parts. In addition, considerable operating experience can be obtained with one design.

The Mark II turbine pump, shown in Figs. 10 and 11, is basically a modification of the Mark I unit designed to meet the needs of the large potassium system. The increased power is obtained by using a two-stage, partial admission, impulse-type turbine. Additional flow capacity is provided by using two diffusers from a concentric collector. The impeller is a Barske wheel, which is completely open; i.e., no front or back shroud, just four radial vanes. Operational and design criteria for the turbine pump is shown in Table 1.

As the design of the Mark II turbine pump progressed it became apparent that the new design could, by virtue of its improved turbine performance characteristics, be operated as a replacement for both the ORNL and the Aeronutronic Mark I turbine pump in their respective systems as well as in the large boiling potassium system for which it was designed. In the small potassium system it would require all the boiler vapor flow to obtain about one-half of the turbine pump design flow at design head and speed, but in the intermediate potassium system it could be operated at design conditions.

Many features in the Mark II design were the result of experience gained in designing, fabricating, maintaining and operating both the ORNL and the Mark I turbine pumps. Based on this experience, several precepts were established that were to serve as guidelines in the design of the Mark II turbine pump. Some of the more notable precepts and their backgrounds are as follows:

Access. We felt that since the turbine pump was one of the more likely components in the system to give trouble in the early stages of the development program, it should be designed so that the entire rotary assembly, i.e., turbine wheel, shaft, impeller, bearing housing and retaining flange, could be removed and a new one installed without cutting any of the lines to the unit. Removal of a rotary assembly from the intermediate potassium system was a considerable inconvenience, as noted previously, requiring several new weld joints and making it difficult to avoid contamination of the system.

Assembly and Disassembly. The bolts holding the rotary assembly in place should be located outside the outer diameter of the turbine wheel so that the entire rotary assembly could be removed from the turbine housing without removing the turbine wheel from the shaft.

Screw holes should be provided for jacking the rotary assembly out of the turbine housing should that be necessary.

All mating assembly surfaces should be selected to eliminate material combinations with bad galling characteristics. It was found in the Mark I unit that the clean TZM molybdenum turbine wheel bore sliding on the clean TZM molybdenum shaft was extremely susceptible to galling while being assembled and disassembled. This material combination in particular was to be avoided in the Mark II design. Wherever possible, sliding fits should consist of tungsten carbide against itself or against molybdenum.

To facilitate the disassembly of the rotary assembly, the shaft and nut screw threads should receive special treatment. The threads (National Coarse) should be ground and lapped and the roots and crests of the thread profile should be given a radius to reduce surface asperities and discontinuities that would cause high contact stresses and subsequent galling, especially under the combined effects of high temperature and potassium. All retaining nuts (TZM molybdenum) should be carburized and provided with slots to make it easier to cut or break them off should galling or sticking occur. The threads on the cap screws (304 stainless steel) holding the rotary assembly into the turbine housing (304 stainless steel) should be nitrided to reduce galling.

Operation. To avoid bearing loading conditions conducive to bearing failure, the center of gravity of the assembled rotating parts is located well inboard of the two journal bearings and the casings are designed so that the fluid dynamic forces on the turbine and impeller should be oriented to give satisfactory bearing load conditions over the full operating range.

Differential Thermal Expansion. To accommodate the differential thermal expansion of the tungsten carbide bearing sleeve and the type 304 stainless steel turbine housing, a set of conical surfaces should be used. The apices of the two cones have a common point of intersection on the centerline of the shaft so that the differential thermal expansion between the two parts results in slippage along the conical surfaces with no change in the contact pressures. The contact pressure on the conical faces is maintained by means of a preloaded spring washer that will provide a force of approximately 100 lb.

A test of a pair of preloaded conical surfaces (tungsten carbide versus nitrided stainless steel) in potassium has been initiated. An earlier test of TZM molybdenum versus stainless steel in potassium consisted of thermally cycling the test pieces from 1050°F to 1450°F through 42 cycles and from 200°F to 1450°F through 4 cycles. The damage to the surfaces was slight.

Bearings

The rotor of the Aeronutronic Mark II turbine pump is supported on two potassium lubricated bearings each having a diameter of 1.0 in., a length of 0.75 in. and a radial clearance of 0.0008 in. To provide some operating margin against bearing whirl the radial clearance was decreased from the 0.0017 in. used in the Mark I design. In addition, each bearing has two axial grooves located in the upper half. These are located 45° on either side of the vertical centerline to suppress half-speed whirl. This groove geometry was chosen because there is some question as to the exact direction of the bearing loads over the entire operating range of the turbine pump. The axial grooves do not extend over the entire length of the bearing in order to reduce the lubricant flow rate, particularly into the turbine cavity.

The loads on the journal bearings are made up of three components:

1. Radial resultant of the tangential impulsive forces acting on the turbine wheel.
2. Radial hydraulic thrust on the impeller.
3. Weight of the rotary element.

There is considerable control over the resultant bearing forces through the location of the turbine nozzle blocks and the two pump diffusers. In accordance with the precepts established for the Mark II design, the center of gravity of the rotary components has been located inboard of the two journal bearings by reducing the weight of the turbine wheel and increasing the span between bearings. Using the shaft made entirely of tungsten carbide also helps a little. The Mark II design has the turbine nozzle blocks and pump diffusers oriented to give equal journal bearing loads of about 3 lb at the design conditions.

An analysis of the bearing characteristics and rotor dynamics was made by Mechanical Technology, Inc.* This study indicates that over most of the operating range the journal bearings are operating in the laminar regime. It is only when the design condition of 8400 rpm is exceeded that the bearing will be in the turbulent regime. However, this is not of much importance since the main effect of turbulence is to increase the friction losses in the bearing. The minimum film thickness in the bearing was between 0.0001 in. and 0.0004 in., being about 0.0003 in. at the design point. As a result of the analysis it was decided to increase the bearing length from the 0.5 in. of the Mark I design to 0.75 in. in order to increase the minimum film thickness.

The completely open impeller was used to reduce its contribution to the thrust bearing load at all operating conditions. Thus, axial thrust is relatively independent of the turbine pump speed and primarily dependent on the differential pressure maintained between the turbine cavity and the pump suction. It is believed best to keep this differential pressure slightly biased so that the turbine end thrust bearing will be loaded to prevent an excessive flow of liquid potassium into the turbine cavity.

The thrust bearings are of a design similar to that used in the Mark I unit as is the flow path of the lubricating potassium. The base materials used in the Mark II units are the same as those used in the Mark I units except that the turbine pump housing is type 304 stainless steel.

Thermal Sleeve

To meet the requirements of the precepts set forth for the design of the Mark II unit, namely that the piping to the turbine pump would not have to be cut for removal of the rotary assembly of the unit, the turbine inlet line was brought through the body of the turbine pump housing adjacent to the bearing and pump portion of the unit. At the design conditions of the unit this meant that there would be a thermal gradient from the 1540°F vapor to the 1000°F liquid. A thermal sleeve was installed to provide a temperature gradient that would not result in excessive stresses.

*J. E. Lund, "Analysis of the Bearing Characteristics and the Rotor Dynamics of the Mark II Turbopump," MTI-65TR38, July 14, 1965.

Water Tests

Some test results have been obtained from operating a water mockup of the Mark II turbine pump while driving the turbine with nitrogen. Head-flow test data at various speeds are shown in Fig. 12. The overall efficiency of the turbine pump versus pump flow is shown in Fig. 13. Pressure data were also taken to determine the pressure distribution around the collector from which the resultant radial thrust on the impeller and its direction was calculated for use in the bearing analyses.

Status

At present the components of the Aeronutronic Mark II turbine pumps are being fabricated. A prototype, having the same basic configuration as the Mark II potassium unit, has been designed for testing the turbine pump with water. The radial clearance of the journal bearings has been increased to about twice that used in the Mark II potassium unit to provide equal Sommerfeld numbers and equal Taylor numbers for operation with water at 120°F and potassium at 1050°F. Induction type proximity probes are located adjacent to each journal bearing to measure bearing eccentricities, attitude angles, and the onset of instability. Another proximity probe is to be installed for measuring axial displacement of the rotor, in particular, for determining the pressure conditions for zero axial load on the thrust bearings. The bearing surfaces are chrome plate against bearing bronze. All the other components are stainless steel except for the turbine wheel which is a Mark II TZM molybdenum turbine wheel. Pressure taps are located in the collector of the pump to measure the pressure distribution around the impeller. This is being done to determine the effect of cavitation on the radial thrust on the impeller. A water-steam test stand has been designed for testing the prototype. The necessary design for installing a Mark II potassium unit in a small (35 kw) boiling potassium system has been completed.

Conclusions

1. We have found that it is feasible to design potassium vapor turbine driven centrifugal pumps for use in potassium Rankine cycle systems and to obtain successful operation. A wide range of operating conditions has not been investigated.

2. The bearing material combination of tungsten carbide against itself (hard on hard) has proven to be operable in potassium up to 1000°F.
3. The performance characteristics of both the turbine and the pump (Aeronutronic Mark I turbine pump) have shown no signs of deterioration after 1000 hr of operation in potassium. During most of this time the pump has operated with cavitation coefficients varying from 0 to 33% but normally at about 15%.
4. Steam-water tests provided a good insight into most of the mechanical problems and provided adequate measures of the thermodynamic and hydraulic characteristics of the turbine pumps.

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"Materials Compatibility Report," compiled by Bearing and Oil Seal Branch of the Bureau of Ships, USN.

Table 1. Turbine Pump Design and Operational Criteria

	ORNL	Aeronutronic	
		Mark I	Mark II
Speed, rpm	to 25,000	to 8,000	to 8,400
Turbine, inlet temp., °F	to 1,540	to 1,540	to 1,540
Turbine inlet pressure, psia	30	30	15
Turbine vapor flow, #/hr	130	280	180
Turbine exhaust temp., °F	1,050	1,050	1,140
Turbine exhaust pressure, psia	1.5	1.5	3.0
Pump flow, gpm	0.4	8.0	27.0
Pump suction temp., °F	1,050	1,050	1,050
Pump suction pressure, psia	1.5	1 to 4	1 to 4
Pump discharge pressure, psia	70	50	60

Table 2. ORNL Turbine Pump - Operating Experience on Potassium

Rotary Unit Number	Starts and Stops	Total Operating Time, Hours	Cause of Termination
1	53	90	Unable to attain full speed - Excessive nozzle to turbine wheel clearance and impeller inlet shroud rubbed housing.
2	10	92	Bearing sleeve broke - caused by tight mounting pins and stress concentration in eloxed holes in the WC bearing sleeve.
3	9	10	Low performance - impeller inlet shroud rubbed housing.
4	0	0	Particles clogged thrust faces.
5	3	1/2	Turbine end journal seized.
6	38	327	Impeller end journal seized.
7	15	20	Pump head rise unsatisfactory due to high resistance of filter in pump suction line.
8	1	1/2	Starting difficulty - System was operated 3 weeks on auxiliary pump before attempting to start turbine-pump - found shaft had broken at root of threads flush with out board end of turbine-wheel.
	129	540 hours	

Table 3. Performance of Aeronutronic Mark I Turbine Pump on Potassium

Run No.	Turbine Pump Speed rpm	Turbine				Pump				Cavi- tation %
		Inlet		Exhaust		Suct.	Suct.	Head psi	Flow gpm	
		Temp. °F	Press. psia	Temp. °F	Press. psia	Temp. °F	Press. psia			
A. Non-cavitating performance - Pump operating in closed loop with auxiliary pump supplying boiler feed.										
1	3360	1094	2.2	954	.7	652	1.6	10.3	.7	0
2	4860	1140	3.1	990	1.0	645	2.7	21.2	1.0	0
B. Cavitating performance - Turbine pump driving the system. Auxiliary pump off.										
3	4031	1095	1.6	975	≈0	767	.1	13.3	2.0	9
4	5195	1144	2.9	1029	.5	854	≈0	18.2	2.6	25
5	6116	1198	4.4	1096	1.4	915	.2	29.1	3.5	14
6	6611	1219	5.3	1108	1.8	935	.1	35.8	3.9	9

ORNL Turbine Pump
Fig. 1

ORNL Turbine Pump.

Fig. 2.

ORNL Turbine Pump - Pump Head vs Pump Flow at Various Speeds with Water.

Fig. 3.

ORNL Turbine Pump-Performance Characteristics of ORNL Turbine Pump on Steam and Water.

Fig. 4 .

Aeronutronic Mark I Turbine Pump.

Aeronutronic Mark I Turbine Pump.

Fig. ⁶~~3~~.

Differential Pressure Versus Bearing Leakage During Water Test of Original Aeronutronic Mark I Impeller.

- ORNL TEST WITH IMPELLER VANE-TO-HOUSING AXIAL CLEARANCE OF 0.057 in. TURBINE DRIVEN BY STEAM.
- AERONUTRONICS TEST OF PROTOTYPE PUMP WITH AXIAL CLEARANCE OF 0.026 in. TURBINE DRIVEN BY NITROGEN.

Aeronutronic Mark I Turbine Pump - Pump Head vs Pump Flow at Various Speeds.

Aeronutronic Mark I Turbine Pump - Typical Cavitation Test Results.

Aeronutronic Mark II Turbine Pump.

Fig. ~~12~~.

Aeronutronic Mark II Pump-Overall Turbine Pump Efficiency vs Pump Flow.

Fig. 11.

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Aeronutronic Mark II Turbine Pump - Pump Head vs Pump Flow at Various Speeds with Water.