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ORGANIC RANKINE CYCLE TECHNOLOGY
PROGRAM

COVERING THE PERIOD 1 JULY, 1966 to
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FOREWARD

This quarterly report was prepared by Sundstrand Aviation, a Division of Sundstrand Corporation, in accordance with the requirements of Contract AT (04-3)-651. Mr. Gerald S. Leighton, Division of Space Nuclear Systems, USAEC, Washington, D. C., administers the program, while Mr. A. Warren Adam serves as Project Engineer at Sundstrand. Mr. M. R. Schilling is the Contract Program Manager for Sundstrand.

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1.0

INTRODUCTION AND SUMMARY

The Organic Rankine Cycle Technology Program was initiated February 15, 1966, under United States Atomic Energy Commission, Contract No. AT(04-3)-651.

The purpose of this program is to investigate the technology necessary to successfully develop an Organic Rankine Cycle power conversion system capable of long duration operation in a space environment. The specific working fluids to be evaluated are biphenyl and the eutectic mixture of biphenyl and biphenyl ether.

The program is divided into four principal tasks.

They are:

a. The Working Fluid Investigation

The purpose of this task is to investigate the pyrolytic degradation of biphenyl and the eutectic mixture of biphenyl and biphenyl ether.

b. The Bearing Investigation

The purpose of this task is to investigate performance and stability of non-rolling contact and rolling contact bearings operating in biphenyl.

c. The Condensate Pump Investigation

The purpose of this task is to evaluate small high speed pumps operating at high suction specific speeds with biphenyl.

d. The Boiler Investigation

The purpose of this task is to investigate heat transfer, stability and performance characteristics of organic working fluid boilers.

In addition, a detailed preliminary design of a complete power conversion system will be performed, as well as some miscellaneous analyses to be defined by the USAEC Project Manager.

During this reporting period, work has been proceeding on the working fluid investigation. In addition, the boiler investigation and bearing investigation have been started. Major activities and accomplishments were as follows:

a. Working Fluid Investigation

Fabrication of the test loops continued. The loop regenerator and condenser-subcooler assemblies are being cleaned prior to final assembly. Fabrication and assembly of the control consoles and heat rejection system were started. Preparation of the test facility

continued. Construction of the test cell itself is near completion and it is now being plumbed and wired. The test loop design specification was completed. All major components are on order or presently being fabricated. The delivery date of the system pumps has been slipped to January 25, however, and unless this date can be changed, this task will incur a significant schedule slippage. Alternate approaches are presently being studied.

b. Bearing Investigation

Activities in this area were principally concerned with determination of speed and rotating weights requirements for the bearing rig and selection of the four bearing configurations to be tested. Determination of the fluid properties of the working fluids is in progress.

c. Boiler Investigation

Preliminary test loop and test section configurations were selected. Instrumentation requirements were established. Task requirements were reviewed and test objectives were outlined.

2.0 TECHNICAL DISCUSSION

The following is a discussion of the work accomplished during this reporting period.

2.1 Working Fluid Investigation

The objective of this activity is to investigate the pyrolytic degradation of biphenyl and the eutectic mixture of biphenyl and biphenyl ether. To this end, four hermetically sealed test loops are being fabricated. Figure 2.1-1 is an isometric view of a typical loop assembly. It shows the infrared spectrophotometer in position to take a reading through the gas side cell. Small deviations in this component arrangement and plumbing may be necessary, but in general, the loop will be assembled in this configuration. Status of the loop components is given in Section 2.1.2.

DEGRADATION LOOP INSTALLATION

FIGURE 2.1-1

2.1.1 HYDROGEN EVOLUTION AND DIFFUSION IN THE LOW
PRESSURE SIDE OF THE TEST LOOPS

A potential problem area resulting from the pyrolysis of biphenyl or the biphenyl-biphenyl ether eutectic is the evolution of non condensible and possibly insoluble gasses. From previous degradation studies it is predicted that hydrogen and benzene will be evolved in the high temperature area of the loop. Due to the bulk flow in the system, the majority of these noncondensibles will be forced into the terminal point in the loop which is the condenser. The benzene will be soluble in the working fluid and will recycle without excessive difficulty. However, the hydrogen is practically insoluble and will, after some time, attain a sufficient partial pressure to hamper the heat transfer in the condenser.

The hydrogen will diffuse through the stainless steel walls of the condenser and the entire low pressure side of the loop toward areas of lower hydrogen partial pressure. For the sake of conservatism it was assumed that the entire amount of hydrogen evolved will be present in the condenser and any diffusion will be from this area. Hydrogen will diffuse through

the outside walls of the condenser and, more importantly through the tube walls containing the coolant, since both the atmosphere and the coolant will be relatively free from hydrogen. In order to increase the diffusion rate a diffusion disk of pure palladium having an area of 4 square inches (25.6 cm²) and a thickness of .030 inches (0.76 mm) will be gold brazed over a cut out in the outside condenser wall. The diffusion rate of hydrogen through a metal is calculated by the following equation:

$$q = \frac{k a (\sqrt{P_1} - \sqrt{P_0})}{t} \left(\frac{\text{cc (STP)}}{\text{hr}} \right) \quad (1)$$

where

k is the permeability of the metal in cc(STP)mm

$$\frac{\text{atm}^{1/2} \text{ cm}^2}{\text{hr}}$$

a is the surface area in square centimeters

P₁ is the higher H₂ partial pressure in atmospheres

P₀ is the lower partial (in this case assumed 0)

t is the metal thickness in millimeters

q is the flow rate per hour of standard (STP) cc of hydrogen

1 cc of hydrogen at 1 atmosphere and 77°F standard pressure and temperature (STP) weights 1.8 x 10⁻⁷ pounds

Thus, rewriting the diffusion equation in terms of mass flow rate gives the following:

$$\dot{m} = 1.8 \times 10^{-7} \frac{k a \sqrt{P_1}}{t} \quad \text{pounds per hour (2)}$$

Table 2.1.1-1 shows the amount of important diffusion areas and the metallic thickness associated with the condenser components. Thus, the area of 316 stainless steel available for diffusion is equivalent to a piece 9890 square centimeters in area and 1 millimeter thick.

The permeability value used in equation (2) was taken from Figure 2.1.1-1. A value of 1.0×10^{-5} cc(STP)mm

$$\frac{\text{cm}^2 \text{ atm}^{1/2}}{\text{hr}}$$

was chosen. This is in actuality the value shown for 347 stainless steel at 300°F (149°C). However, this was used in preference to extrapolating the 316 stainless steel curve from the meager data available. This will provide a more conservative solution, which is desirable in light of the distinct possibility that the rate of hydrogen evolution given in Figure 2.1.1-2 is not accurate. The permeability of palladium is taken from Figure 2.1.1-1 as 6.0×10^{-3} cc(STP)mm

$$\frac{\text{atm}^{1/2} \text{ cm}^2}{\text{hr.}}$$

TABLE 2.1.1-1

LOCATION	AREA (in ²)	THICKNESS (in)	GEOMETRY FACTOR, $\frac{A}{t}$ (cm ²) mm
Condenser Shell	120	.125	224
Available Reservoir Shell	126	.090	355
Condenser Tube Bundle	1275	.035	9260
Tube Bundle End Cap	18	.090	51

DIFFUSION OF HYDROGEN
THROUGH METALS
FIGURE 2.1.1-2

EVOLUTION RATE OF LOW BOILERS
FIGURE 2.1.1-2

Evolution of hydrogen in the boiler will be much greater than in the rest of the loop due to the exponential temperature dependence of the decomposition process. The boiler fluid temperature is 650°F and the maximum regenerator temperature is 550° (thus more than an order of magnitude difference in the rates.) From past experience with a similar boiler, it is expected that the boiler inventory will be about 2 pounds at all times. At 650°F two pounds of biphenyl will evolve approximately 8×10^{-9} pounds of hydrogen per hour. With this amount of hydrogen evolving, the equilibrium can be calculated from equation 2.

$$\dot{m} = 8.0 \times 10^{-9} \frac{\text{lb}}{\text{hr}} \text{ evolved} = 1.8 \times 10^{-7} \sqrt{P_1}$$

$$(1.0 \times 10^{-5} \times 9890 + 6.0 \times 10^{-3} \times \frac{25.6}{0.76})$$

$$P_1 = \left[\frac{8.0 \times 10^{-9}}{1.8 \times 10^{-7} (.0989 + .201)} \right]^2 = .0219 \text{ atm} = .324 \text{ psia}$$

$$P_1 \text{ without palladium} = \left[\frac{8.0 \times 10^{-9}}{1.8 \times 10^{-7} (.0989)} \right]^2 = .202 \text{ atm} = 2.97 \text{ psia}$$

While a 2.97 psi increase in the condenser pressure would not be intolerable in the degradation loops, it would cut the turbine pressure ratio to about one fourth of the design value in an actual Rankine cycle dynamic system. In addition, the diffusion data and the gas

evolution data are not certain. It was therefore considered advisable to add the palladium plug to the condenser.

The equilibrium pressures are attainable only after an infinite time. However, the pressure build-up with time is a function of the volume available for the non-condensibles, the evolution rate and the diffusion rate. Equating the rate of storage to the difference between the evolution rate (8.0×10^{-9} lb/hour) and the diffusion rate (equation 2) the pressure build-up rate can be written as follows:

$$\dot{m}_{\text{storage}} = 8.0 \times 10^{-9} - 1.8 \times 10^{-7} \sqrt{P_1} \sum \frac{k_1 a_1}{t_1} \quad (3)$$

$$m_{\text{stored}} = \frac{P_1 \times V \times 2060}{RT} \text{ from perfect gas laws} \quad (4)$$

$$P_1 = \frac{\dot{m}_{\text{storage}} (RT)}{2060 V} \quad (5)$$

Where:

$\dot{m}$ = rate of storage lb/hour

R = universal gas constant ft lb/lb mole °R

T = absolute temperature °R

V = total volume of non-condensibles ft³

P_1 = hydrogen partial pressure in atmospheres

$\dot{P}_1$ = time rate of change in P_1 atmospheres/hour

Combining equations (3) and (5) and solving the resulting differential equation the pressure is determined as a function of time as shown in Figure 2.1.3-3. The solution to this equation takes the form:

$$\text{time} = \frac{(B/A - B/a \ln (B/A - P_1^{1/2}) - P_1^{1/2})}{A/2} \Big|_0^{P_1} \quad (6)$$

where:

$$A = 1.8 \times 10^{-7} \sum_{t_1} \frac{k_1 a_1}{t_1} \frac{RT}{2060V}$$

$$B/a = .0219^{1/2} = P_1(\text{maximum})^{1/2}$$

The curve in Figure 2.1.1-3 can be flattened to any extent desirable by the addition of volume to the condenser or the addition of more palladium for diffusion to accomodate any mission length desired.

BUILD-UP OF HYDROGEN PARTIAL PRESSURE IN CONDENSER

FIGURE 2.1.1-3

2.1.2 Test Loop Status

The following is a status of the major components to be used in the test loops.

ED 16305 - Regenerator - all parts have been completed and await cleaning prior to final assembly. Figure 2.1.2-1 shows the regenerator with the side plate removed. Shown are the rows of tubes and the milled channels that separate the flow passages. Figure 2.1.2-2 has the side plate installed with the stiffening fasteners. These shorten the span between the welded ends and minimize the deflection of the plate when the liquid side of the regenerator is pressurized.

ED 16315 - Condenser - all parts have been completed except for the palladium diffusion section. A sample strip of material has been ordered to develop a welding technique to join palladium to 316 stainless steel. The palladium required for the condenser assembly will be ordered at a later date. Delivery time is only 10 days. Figure 2.1.2-3 is an exploded view of the condenser components, tube bundle, outside shell and subcooler. Figure 2.1.2-4 shows all parts, except the front

FIGURE 2.1.2-1

ED-16305 REGENERATOR

FIGURE 2.1.2-2
ED-16305 REGENERATOR
WITH SIDE PLATES

FIGURE 2.1.2-4

ED-16315 CONDENSER, ASSEMBLED

flange assembly, in an assembled but unwelded arrangement. All parts are awaiting cleaning prior to final assembly.

ED 16310 - Boiler - The boiler has been completed except for the boiler tubing. The tube assembly is due to be shipped from the vendor October 10. When received, the tube will be leak checked and cleaned for installation into the loop.

ED 16307 - Jet Pump - The design and drafting has been completed and parts are now on order.

ED 16302 - Silicon Windows - The windows have been ordered and are due to be shipped shortly.

Chempumps - The delivery date has been slipped 14 weeks. The vendor's reasons for the delay are the unavailability of raw materials and the long lead time on the pump castings.

Data Logging System - System is in transit to Rockford from the vendor.

Transducers - Some of the pressure transducers have been received. However, delivery of the differential pressure transducers has been slipped to the later part of November.

Heater Controls - West controller has been received.

Power Modulators - Silicone control rectifiers have been received.

Control Console - Consoles are partially completed and are in transit to Rockford, where the assembly will be completed.

Working Fluid - The biphenyl and the eutectic biphenyl-biphenyl ether have been ordered and delivery is expected within two weeks.

The remaining parts, those not specifically listed in the status report, are either completed or are due to be received with no delays anticipated.

2.2

Bearing Investigation

A bearing investigation program has been initiated with the objective of defining bearings suitable for space power combined rotating units which produce one(1) to ten (10) kilowatts of usable electrical power. The bearings are to be lubricated with the cycle process fluid which will be biphenyl. Design life is to be 10,000 hours at rated speed. Bearing stability is important since the system must ultimately operate in a zero g environment, where loads are minimal and stability problems are most likely to occur. Test bearings will be designed to

support a test rig shaft with weight equal to the system CRU weight, and will be capable of operating in any attitude.

2.2.1 Test Rig Requirements

Since bearing power consumption is not insignificant in small power systems, a single bearing system size would not be suitable to cover the order-of-magnitude power range under study. Likewise, critical speed is closely related to shaft stiffness and will vary over the power range and be yet different in the test rig. This will not invalidate bearing development effort, rather it simply means that once a mission is fully defined and actual system capacity is established the bearings must be suitably tailored for that system.

A system capacity about in the middle of the power range under study has been selected as representative of that range and will be used as the basis upon which test hardware will be built and evaluated. The system selected to define the test bearings has a capacity of 5 kw and a rotating speed of 24,000 rpm. Following are listed other parameters of this system which influence bearing design:

- (a) Lubricant: biphenyl
- (b) Lubricant temperature: 297°F
- (c) Rotor weight: 12.6 pounds. Design bearings for one g capacity in any axis.
- (d) Bearing span: 7.25 inches.
- (e) Wheel pitch diameter: 7.00 inches.
- (f) Wheel nozzle and tip clearance: .005 inch.
- (g) Turbine output: 8.37 horsepower.

The test rig shaft will be kept geometrically simple so that its stiffness can be easily calculated. Shaft stiffness and computed bearing stiffnesses will be used to make predictions of shaft critical speeds. As previously stated, orbital bearing operation involves the combination of low load and high speed which maximizes the need for bearing stability. A zero g environment cannot, of course, be created for any substantial period in laboratory surroundings. Since stability problems are associated with radial bearings, a reasonable simulation of the zero g condition is obtained by operating the rig with the shaft oriented vertically. This will be done to verify the zero g characteristics of the radial bearings.

2.2.2 Bearing Configuration Selection

Four bearing types are to be built and tested in this program; these include two fluid film configurations and two rolling contact bearing designs. Type selection and preliminary design work has progressed to the point where the following bearing configurations may be put forth as the most attractive candidates. Included with the bearing descriptions are general comments and indications of the directions likely to be taken in the final detail designs.

- a. Tilting Pad - This bearing will be fully self aligning and will be basically similar to successfull predecesor bearings built by Sundstrand. The new bearings will utilize a modular concept in lieu of the piece assembly used in prior designs. Soft vs hard materials formerly used will be replaced by hard vs hard materials to improve resistance to scoring during many start-stop cycles when boundary lubrication conditions exist. This sacrifices the feature of embedability but this is felt to be of insignificant importance in a closed loop system. The ultimate material selection will probably be carbide or ceramic.

- b. Floating Pad - This bearing type utilizes orbiting pads which operate on fluid films on both sides of the pads. The pads are shaped so that each side acts as though it were a simple pad with optimum pivot location. Orbital speed of the pads will be from about $1/3$ to $1/2$ of shaft speed depending upon detailed design geometry. Limited data published on these pads indicates a highly successful history. The thrust bearing accompanying the floating journal pads will be either arcuate floating pads or spiral groove thrust pads. Since pad centrifugal forces are substantial, low density materials will be selected for the pads. Under consideration are hard anodized aluminum and polyimide, a plastic material with favorable bearing characteristics and outstanding thermal properties. It may be desirable to fabricate interchangeable pad elements from both of these substances.
- c. Cageless Roller - This is a concept which was originated at NASA and upon which limited development was done by the MRC Bearing Company. The concept provides pure rolling and thus

eliminates cage rubbing, a feature that is extremely attractive with marginal lubrication conditions. The bearing worked out for the present evaluation tests utilizes dumbbell shaped separator rollers between each of the neighboring main rollers. The small separator diameter experiences pure rolling contact with the rollers it separates and the larger separator diameters roll on auxiliary races which rotate at shaft speed. Thrust capacity is provided by shoulders with a height equal to 20% of the roller diameter. Skuffing is minimized by the use of conical shoulders which mate against conical roller end surfaces. One bearing with thrust capability will be used in actual shaft support and the other end will be designed to float axially to allow for thermal differential expansion. The available history on this bearing type is brief but spectacular in a test where the bearing was run absolutely dry. With reasonable expectations from the lubricants associated with this program, optimum rolling contact bearing performance can be expected from this candidate.

d. Angular Contact - This is a relatively conventional ball bearing type. Since environmental temperatures are high and this type bearing does not experience pure rolling of balls against races it should be made of M-50 tool steel. The polyimide plastic mentioned in connection with the floating pad bearing will be used for cage construction. This material alone has provided the sole lubricating mechanism for experimental high temperature bearings by gradual wearing away in the ball pockets and smearing the balls and races with itself. Again, with reasonable expectations from the lubricants in this program, expectation of good life characteristics are not unreasonable. This concept is felt to be the weakest of the four basic designs presented here, but to at least be worthy of test evaluation.

2.2.3 Rotating Assembly Weight and RPM Determination

In order to size the bearings and the bearing test rig, it was necessary to determine the weight of the rotating assembly and the anticipated design speed. The rotating assembly is comprised of the turbine wheel, the shaft, the pump impeller, and

the alternator rotor. A study was conducted to optimize turbine efficiency for the cycle selected for this program as a function of wheel pitch diameter and rpm. The pitch diameter must be determined in order to establish the turbine wheel weight. Determination of rpm, in addition to being necessary for the test program itself, is also required for determining the alternator weight. The significant parameters which were held constant throughout the study were:

Electrical Conversion Efficiency	80%
Turbine Inlet Temperature	1110°R
Turbine Inlet Pressure	70 psia
Turbine Pressure Ratio	70:1
Number of Blades	115.0
Blade Angle	28°
Chord	0.3 in.
Nozzle Flow Coefficient	.94
Blade Height	0.4 in.
Axial Clearance	.005 in.
Tip Clearance	.005 in.
Exhaust Head Loss Factor	1.00

Pitch diameters from 5.0 to 9.0 inches and wheel speeds of from 14000 to 28000 rpm were considered.

The turbine efficiency was calculated using an IBM computer program developed by Sundstrand which computes turbine performance based on specified turbine geometry and speed.

The calculation procedure is basically an aerodynamic analysis with each aspect of the turbine separately evaluated. The effects of Reynolds number, Mach number, rotor blade choking and rotor blade aspect ratios are evaluated. Losses associated with disk friction, rotor blade turning, incidence angle, flow leakage, blade pumping and blade scavenging are also included in the calculations. The losses not considered in the program are the bearing and alternator rotor windage losses. Variation in turbine efficiency as a function of rpm and pitch diameter is shown in Figure 2.2.3-1.

Predictions of bearing and alternator losses were obtained by an analysis of factors affecting these losses. It was established that if the bearing geometry were optimized for each speed, the bearing power absorption would vary directly as the rotating assembly weight to the first power and the speed squared.

$$\frac{HP_2}{HP_1} = \frac{(W_2) (U_2)^2}{(W_1) (U_1)^2}$$

TURBINE AND CRU PERFORMANCE

FIGURE 2.2.3-1

The bearing and windage losses were extrapolated from data obtained from the 1.5 kwe CRU operating at 24,000 rpm. (HP_1). By knowing the weight of the 1.5 kw hardware and estimating weights for a 5 kwe rotating assembly as a function of rpm and wheel diameter, bearing and windage losses were calculated for the full range of speeds and pitch diameters investigated. These losses were factored into the turbine performance using CRU efficiency, which is defined as follows:

$$\eta_{CRU} = \frac{\text{Turbine HP} - \text{Bearing and Windage Losses}}{\text{Turbine HP}}$$

The effect of the bearing and windage losses is not insignificant in a machine of this size, as illustrated in Figure 2.2.3-1. It is interesting to note, however, that if the turbine design is optimized for wheel diameter and bearing configuration, wheel speed is not very significant except as it affects alternator design. As a result of this study, the following parameters were determined, and will be used throughout this study.

Pitch Diameter	7.0 in.
Wheel Speed	24000 rpm
Wheel Weight	2.8 lbs.
Alternator Rotor Weight Shaft Weight and Impeller Weight	9.8 lbs.
Total Rotating Assembly Wt.	12.60 lbs.
Turbine	74.1%
CRU	69.4%

The maximum load that will be used to size the bearings will be the total weight of the rotating assembly (12.60 lbs.). This weight is felt to be reasonably conservative.

2.3 Boiler Investigation

The ultimate goal of the Organic Rankine Cycle Technology Program is to investigate the technology necessary to develop a power conversion system capable of long term operation. A major area of concern for long duration conversion systems using biphenyl is the pyrolytic degradation of the working fluid. One concern is that the non-condensable gaseous decomposition products may result in an appreciable partial pressure in the condenser, thus

reducing the system pressure ratio and power output. Another concern is that the non-gaseous decomposition products may agglomerate on heat transfer surfaces, thereby increasing thermal resistances in heat transfer components.

The available data regarding the pyrolytic degradation of biphenyl indicate that the degradation is very dependent on fluid temperature level, the degradation rate increasing about an order of magnitude for every 70°F increase in temperature. Since the boiler is the hottest component in an organic Rankine cycle, it is thus the critical component as far as pyrolytic degradation is concerned.

Data on the fouling of the heat transfer surface in the boiler will be obtained by examining the boiler tubes used in the Working Fluid Investigation loops.

Every effort should be made to minimize boiler fluid inventory, since the total rate of non-condensable gas evolution is proportional to the

mass of fluid at the high temperature. Also, the boiler must be designed to avoid excessive tube wall temperatures since the fluid adjacent to the tube wall hot spot will degrade at a rate significantly higher than the bulk of the fluid. To this end a boiler tube test program is planned. The heat transfer, stability and performance characteristics of biphenyl boilers will be investigated. This will be done for ordinary boiler tubes as well as for tubes with various fluid inventory control and/or heat transfer augmentation tube inserts. One of the primary objectives of this testing will be to provide enough experimental data to either verify existing boiler design correlations or develop new correlations.

A further objective of the overall program is to analytically investigate the coupling of a biphenyl boiler to an isotope heat source. Obviously, this cannot be undertaken intelligently unless accurate biphenyl side boiler heat transfer data are available.

2.3.1 Boiler Test Program

2.3.1.1 Test Objectives

The broad objective of this test program is to obtain the refined data necessary to eliminate the major sources of uncertainty in biphenyl boiler tube performance.

The specific test objectives and a brief description of their justifications are as follows:

a. Determination of Vapor Quality at which Dry Wall Occurs.

The value of the quality at which dry wall occurs, commonly referred to as the critical quality (X_{crit}), is important since it defines the point where vapor convective heat transfer commences.

b. Measurement of the Axial Tube Wall Temperature Profile.

The wall temperature profile is an important consideration as it determines the maximum fluid temperature possible. It is also necessary in the determination of local heat transfer coefficient, from which X_{crit} is inferred.

c. **Measurement of Fluid Centerline Temperature.**

Centerline fluid temperatures will provide data to help determine the difference between wall and fluid bulk temperatures in the liquid portion of the boiler. In addition, they may show if superheated fluid exists in the core of the flow and may be useful in defining the radial fluid temperature profile in the dry-wall portion of the boiler.

d. **Determination of Heat Transfer Coefficients.**

The heat transfer coefficients, as a function of pertinent system parameters, e.g. Reynolds number, will be useful in predicting performance of other boiler tube configurations and systems.

e. **Measurement of Two-Phase Boiling Pressure Drop.**

There is a high degree of uncertainty associated with available two-phase pressure drop correlations. It is, therefore, desirable to empirically confirm these correlations with experimental data closely related to the system(s) of interest.

Pressure data is necessary in determining local fluid saturation temperature and will provide an insight into defining the local hydrodynamic flow regime.

f. Determination of the Factors Influential in Boiling Flow Stability.

Time-varying flow in a single once-through boiler tube has been found to exist under certain conditions by several investigators. Therefore, it is of value to determine to what extent the various system parameters contribute to flow instabilities.

g. Evaluation of Boiler Tube Internal Devices.

Various internal devices have been demonstrated to increase heat transfer coefficients, decrease flow instabilities and reduce fluid inventory and residence time in the high temperature region of the boiler tube.

Hence, measurement of the effectiveness of these devices is necessary in order to optimize the design of boiler tubes for practical applications.

2.3.1.2 Test Loop Requirements

The test section used to investigate the performance of single once-through organic boiler tubes will be designed to simulate the conditions anticipated in a 1.5 kw space power system.

The test section will incorporate design features that provide flexibility in the selection of tube diameters, heated lengths, internal devices, instrumentation and heat fluxes.

The test loop will be designed to provide for continuous control of system flow rate, pressure levels, and boiler inlet fluid temperature. The nominal system flow rate will be 170 lb/hr and the working fluid will be biphenyl. The boiling temperature will be approximately 650°F.

Exclusive of the boiler test section, no attempt will be made to simulate the conditions or components of the test loop with a 1.5 kw space power system.

2.3.1.3 Theoretical Design Considerations

The current analytical techniques used to predict performance of a radiation type boiler tube configuration involves the use of a digital computer program written at Sundstrand for this purpose. To date, a high degree of confidence in the computer results has not been realized because of the required use of numerous empirical coefficients. These empirical coefficients have been established from a very limited amount of experimental data and, therefore,

make the results questionable. The boiler experimental test section design is being based upon computer parametric studies using the best empirical input data available at this time.

2.3.1.3.1 Length of Tubing

The length of tubing required is based upon the maximum allowable fluid temperature. The maximum allowable tubing wall temperature can also be specified as the maximum fluid temperature since the layer of fluid immediately adjacent to the wall will approach the wall temperature. The wall temperature is primarily a function of the internal heat transfer coefficient and fluid temperature. Assuming a .5" O.D. tube, the required length for a 1.5 kw system (165.5 lb/hr) will be approximately 26 feet with a maximum allowable wall temperature of 700°F. See Figure 2.3.1.3.1-1. Based upon the parametric analysis presented in Figure 2.3.1.3.1-1, and taking into consideration the possible range of parameters that may be tested, the maximum length of tubing that can be tested was selected to be 32 feet.

Effect of Surface Emissivity on Boiler Design Parameters

FIGURE 2.3.1.3.1-1

2.3.1.3.2 Tube Diameter

No definitive design criteria are available to determine the optimum tube diameter for a particular system design. A primary consideration in selecting a tube diameter again will be the maximum allowable wall temperature. Figure 2.3.1.3.2-1 depicts the tube diameter as a function of maximum wall temperature for a specific set of system parameters. Other parameters to consider include pressure drop, tube weight, fluid residence time and space limitations. A tube diameter of 0.5" was selected for the initial tests, however, a range of tube diameters will be tested to better define the effects of this parameter.

2.3.1.3.3 X Critical

X critical, the quality where dry-wall vapor binding occurs, is another parameter which cannot accurately be predicted because of the complexity of the mechanism involved and the lack of experimental data. The parametric studies were conducted using an X critical of .8, however, this value was merely a crude estimate based upon a meager amount of test data. The primary effect of an increasing X critical is a decrease in the length of tube in which dry-wall has occurred. The

MAXIMUM WALL TEMPERATURE
vs
TUBE DIAMETER

FIGURE 2.3.1.3.2-1

increased wall temperature in the dry-wall region has the net effect of increasing the amount of fluid at the higher wall temperature, thereby increasing the rate of thermal degradation.

Vapor binding length as a function of X critical is shown in Figure 2.3.1.3.3-1 for a specific set of system parameters.

2.3.1.4 Test Loop Design

A schematic drawing of the single-tube organic heat transfer loop is shown in Figure 2.3.1.4-1. The detailed design and specification of test hardware will be initiated in the ensuing quarter. Starting at the exit of the boiler test section and proceeding clockwise around the loop, a description of the major components and their functions is as follows:

A well-insulated, sharp-edged orifice plate is located just downstream of the vacuum chamber with fluid pressure and temperature instrumentation located on both sides. This orifice will provide the pressure drop necessary for determining boiler exit quality.

The vapor side of the regenerator removes heat from the super-heated vapor to reduce the heat load of both the condenser and the preheater. The regenerator will be

FIGURE 2.3.1.3-3-1

identical to those used in the degradation test loops.

The condenser, subcooler and reservoir is an integral unit and will be identical to those used in the degradation test loops.

A fixed-displacement pump will be used to provide system flow. A pressure regulator is located just downstream of the pump and monitors the pressure at the inlet of the boiler section. The system flow rate is therefore controlled by this regulator setting and the size of the orifice at the exit of the test section. A turbine type flow meter will measure the system flowrate.

A bypass circuit allows a controllable amount of fluid to be passed through the liquid side of the regenerator to reduce the load on the preheater when a minimum amount of subcooling is desired.

The preheater consists of a series of electrical clamshell heaters and will provide the final control for the test section fluid inlet temperature. A throttling valve at the entrance to the test section will provide the necessary pressure drop to maintain hydrodynamic stability throughout the test section.

The boiler test section will consist of 32 feet of heated stainless steel tubing enclosed in a vacuum chamber.

The vacuum will insure minimum heat leakage and eliminate convective heat transfer from the radiating surface to the boiler tube. The boiler tube will be broken up into two straight 16 foot long sections joined by an insulated U-section of tubing. Sixteen 2 foot long clamshell heaters will provide the radiation heat flux and each will be spaced a short distance apart to allow the instrumentation to be routed from the boiler tube. The outside of the clamshell heaters will be wrapped with from 1 1/2" to 2" of foil-wrapped super insulation to further minimize heat leakage to the vacuum chamber walls.

2.3.1.5 Instrumentation

2.3.1.5.1 Wall Temperature

Small diameter sheathed chromel-alumel thermocouples will be embedded into the tube wall at one foot intervals. The thermocouple leads will be routed axially to the separation in clamshell heaters and from there pass directly to the vacuum chamber instrumentation port.

2.3.1.5.2 Fluid Temperature

Small diameter sheathed chromel-alumel thermocouples will

be inserted through the tube to the center of the fluid stream. These fluid temperature thermocouples will be located at two foot intervals. The thermocouple leads will be wired into the adjacent wall temperature thermocouple leads in a way which will yield a differential temperature between the wall and the fluid. Provisions will also be incorporated which will provide a readout of each individual temperature.

2.3.1.5.3 Fluid Pressure

Fluid static pressure will be continuously recorded at the beginning and end of the test section by pressure transducers. Intermediate pressures will be measured by using mercury differential manometers located at the separation of each 2 foot long clamshell heater. Each manometer will be referenced to the inlet pressure to provide a measurement for determining the pressure drop as well as the local static pressure (see Figure 2.3.1.5.3-1). The entire pressure recording system will have to be maintained above 160°F to prevent freezing of the biphenyl. No contamination of the biphenyl is anticipated to result from the mercury-biphenyl interface.

TRANSDUCER AND MANOMETER ARRANGEMENT

FIGURE 2.3.1.5.3-1

2.3.1.5.4 Radiation Source Temperature

Radiation source temperatures will be measured for each clamshell heater. A thin axially slotted stainless steel tube will be placed on the inner diameter of each clam shell heater to provide a uniform radiation source. The surface temperature of this tube will be measured by spot-welding a chromel-alumel thermocouple to the surface.

2.4 Test Facility

Work is proceeding on the construction of the test cell, fabrication of the heat rejection system and assembly of the test loop controls and data acquisition system.

Construction of the test cell is near completion. Work remaining to be done involves mostly the installation of plumbing, electrical, and instrumentation facilities. The test cell is 24 by 28 feet. The walls are painted with white epoxy paint and the floor will be of white terrazo. It will be divided into two separate areas. One will be limited access and will contain the endurance test loops. The other, larger area will be utilized for final assembly, checkout, and performance testing.

The test facility will be completed and checked out during the next reporting period.