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DEVELOPMENT OF A THERMALLY
DRIVEN DIRECT ACTING ORGANIC
RANKINE PUMP

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FORWARD

Foster-Miller Associates has contracted with the Department of Energy to perform the following work in the development of a thermally driven direct acting organic rankine pump:

- Establish design specifications for the pump
- Prepare a preliminary design of the acting organic rankine pump
- Analyze the pump design for thermodynamic operating parameters, lift, speed, capacity, thermal efficiency, cost, and reliability
- Fabricate and perform development testing of a laboratory prototype pump
- Prepare a design and thermodynamic analysis of a prototype pump.

The overall objective of this program is to demonstrate the feasibility of the direct acting organic rankine pump concept for use as a low temperature thermally driven irrigation pump. The program was originally contracted to be performed in the time period between June 1, 1978 and May 31, 1979.

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W.M. Toscano was the Program Manager and John Dieckemann, Scott Jameson, and Charles Burrill were co-researchers. Technical and management support was provided by Mr. Robert Alvis, the technical monitor for the program.

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1. INTRODUCTION

Development of devices to use solar energy to pump irrigation water has been pursued by various individuals since the 1600's. Pumping irrigation water is an appealing use of solar energy because the demand for irrigation water and the maximum availability of solar radiation generally occur simultaneously. This concurrence of supply and demand is important because of the high capital cost of solar energy conversion and storage equipment. If close to the maximum available output of the solar device can be used productively *when* it is available, the need for energy storage is greatly reduced or eliminated, while maximum savings in marginal operating costs are generated to repay the cost of the equipment.

The latest work on solar powered irrigation pumping has centered on systems employing concentrating solar collectors providing heat to a conventional Rankine cycle driving a conventional pump. Although conversion efficiencies have been high, the costs of demonstration installations also have been quite high. Even allowing for "*learning curve*" cost reductions, systems of this type show little promise of being economically competitive with electric or diesel driven pumps. So, while a Rankine cycle engine based pumping system will perform at conversion efficiencies as high or higher than other types of systems for a given temperature solar heat input, the costs of such a system must be reduced in order for the system to be an economically attractive means of pumping irrigation water.

A significant portion of the high costs of the current solar Rankine system is attributable to the following factors:

- Complexity of the system
- Complexity of individual components in the system
- High cost of tracking solar collectors
- Power transmission losses between the Rankine engine and the pump.

The demonstration systems have been complex in terms of the number of components and the complexity of their interconnections. A typical system includes concentrating collectors with a heat transfer fluid circulating system, thermal storage, a boiler, a condenser, a regenerator, the turbine, and the pump. Many of the components are quite complex; for example, diurnal tracking concentrating solar collectors and the turbine with gearbox.

Three fundamentally different approaches can be followed to reduce the impact of these factors on solar powered pumping costs.

- Design for absolute simplicity in the heat-to-water pumping device
- Design systems that operate on extremely low temperature heat inputs, perhaps integrating the solar collector and pumping device
- Design a Rankine cycle system that is specifically tailored to the requirements (including the economic requirements) of irrigation pumping.

In the first category are devices such as the open cycle pulse jet pump. These devices can be built for a negligible portion of the total cost of the solar powered irrigation system. However, the pump has an inherently low (2 to 3 percent) thermal efficiency and requires a high temperature (220°F to 230°F) heat source to operate. The reduced cost of this pump is not likely to offset the cost of the additional high temperature solar collector area required to compensate for the inherently low efficiency of the pump.

The second category of pumping systems includes various thermal expansion engines such as the Banks Nitonol Engine, engines based on the expansion of melting waxes, and air-volume change pumps. The major advantage of these systems is their ability to operate with a very low temperature heat source (50°F above ambient). At these low temperatures, the potential exists for greatly reducing costs per unit collector area or even directly integrating the collector and pumping device into a single unit. However, the inherently low efficiencies of these systems requires the collection, handling, and rejection of large quantities of low grade heat and a large (if inexpensive per unit area) solar collector surface to achieve a given power output. Based on our preliminary assessment of potential concepts of this type, no cost relief is likely with this approach either.

The Rankine cycle, with its inherently high thermal efficiency, can be implemented so that it is specifically tailored for the conversion of solar energy to power output in the form of pumped water. To accomplish this, the pumping system should incorporate several important features.

- The vapor expander and pump should be a closely integrated, single component
- The complete system should be as simple as possible, that is, the value of components such as thermal storage and a regenerator should be assessed in economic terms and the extraneous components should be eliminated if the economics prove marginal

- The use of less expensive flat plate collectors (and operating at an input temperature appropriate to flat plate collectors) should be considered instead of concentrating collectors.

The pumping device described in the next subsections incorporates these features and will lead to a substantial improvement in the economics of solar powered irrigation pumping. A comparison between several options for solar powered irrigation pumps is presented in Table 1.

1.1 Description of the Direct Acting Organic Rankine Cycle Pump

A schematic representation of a solar powered pumping system incorporating the Foster-Miller Associates, Inc. (FMA) pump is presented in Figure 1. The operation of the system is described as follows. Condensate is pumped to the hot end working pressure, whether by a separate electrically driven pump or one integrated with the expander-pump package. The condensate enters the boiler and is converted to saturated vapor at the system operating temperature. The boiler may be a separate heat exchanger heated by a heat transfer fluid circulated from the solar collectors or boiling can be accomplished directly in the solar collector. Boiling in the solar collector eliminates the separate boiler from the system and the separate fluid circulation loop. The saturated vapor is expanded in the expander-pump package, simultaneously powering a stroke of the water pump. The spent vapor from the expander is condensed in the condenser and the condensate is then returned to the inlet of the condensate pump. The condenser is cooled directly with the pumped water.

Operation of the expander-pump package is equally simple. In its simplest version (Figure 2) the device consists of a thick piston attached to a coil spring. The piston-spring system oscillates inside the cylinder. Intake and exhaust valves are both actuated directly by motion of the piston. The intake valve is opened briefly by the striker on the piston when the piston is at the expander head end of its travel (the bottom position in Figure 2). The exhaust valve is pulled open until it is pushed closed by the piston at the bottom end of its travel. This valving technique is used successfully in reciprocating steam engines. In a typical expansion-exhaust cycle, the following sequence of events occurs:

- a. The intake valve is opened, admitting a metered amount of hot vapor. The exact amount admitted depends on the height of the valve striker and the response of the valve-spring system.
- b. The vapor begins to expand and the spring begins to elongate, transferring work to the piston. A portion of this work is transferred to the water, lifting it to its delivery height. The remainder is stored as kinetic energy of the piston.

Table 1. Comparison of Solar Powered Irrigation Pumping Techniques

Type of System	Required Input Temperature	Estimated Collector Area Per Delivered kW Pumping Power	Estimated Pumping Cost, Dollars Per Delivered kW Pumping Power	
			Prototype Demonstration	Commercial Production
Organic Rankine systems such as the Gila River Ranch demonstration	300°F up	145 ft ²	0.41 ⁽¹⁾	0.30 ⁽¹⁾
Open cycle pulse jet	220°F to 250°F	400 to 500 ft ²	1.00 ⁽²⁾	0.50 to 0.60 ⁽²⁾
Low temperature differential such as Nitinol Engine	130°F to 150°F	500 to 600 ft ²	0.75 ⁽²⁾	0.40 ⁽²⁾
Foster-Miller system	170°F to 200°F, Can use higher if the economics are favorable	250 ft ² of flat plate with temperatures between 170°F to 200°F	0.31 ⁽³⁾	0.19 ⁽³⁾

1. Source: J.T. Pytlinski and N.D. Eckhoff, "Basic Technical and Economic Aspects of the Use of Solar Energy for Pumping Irrigation Water"
2. Estimated
3. Estimated costs

FIGURE 1. Direct Acting Organic Rankine Cycle Pumping System

- c. At some point in the stroke, the vapor pressure and spring force have decreased so that they are less than the opposing force of the water head. The kinetic energy stored in the motion of the piston provides the work to displace the remaining water and stretch the spring.
- d. Near the end of the piston travel, the exhaust valve is pulled open.
- e. The stored energy in the spring accelerates the piston through the exhaust stroke, providing the needed suction lift.
- f. Near the end of the exhaust stroke, the piston strikes the exhaust valve stem causing it to close, then strikes the inlet valve, opening it and beginning the next stroke.

This pumping concept has a great deal of flexibility in meeting combinations of pump suction lift, and discharge head and flow rate while being driven by different working fluids at different top and bottom temperatures.

Many other configurations of the basic direct action concept are possible. The configuration sketched in Figure 3 has a separate pump diaphragm (or piston) and expander piston. This allows the pump and expander to have different displacements while preserving the integrity of the single pump-expander package. The configurations shown in Figures 3 and 4, both having separate expander and pump pistons, have the advantage of completely separating the working fluid and the pump water, eliminating the possibility of either one contaminating the other.

Losses in the expander-pump arise from the same sources as those in most reciprocating machinery. They include:

- Aerodynamic losses, primarily the pressure loss through the intake and exhaust valves
- Heat transfer loss from the hot vapor to the cylinder wall
- Piston ring friction
- Main spring dissipation due to other vibrational modes
- Guide rod friction (see Figures 2 through 4) although it may be possible to eliminate this part from the design
- "Mixing" loss during vapor intake

FIGURE 3. Direct Acting Rankine Cycle Pump with Oil Coupled Diaphragm Pumping Section

FIGURE 4. Direct Acting Rankine Cycle Pump with Rod Coupled Diaphragm Pumping Section

- Dissipation of slight extra work (<5 percent) used to ensure stable operation
- Valve actuation loss
- Pressure loss across the pump side intake and exhaust check valves.

In a well designed reciprocating machine, these losses can be maintained below 40 percent of the available work so that an overall conversion efficiency in excess of 60 percent of the ideal Carnot cycle efficiency can be expected.

The major advantages of this approach to irrigation water pumping are:

- Simplicity and compactness
- High conversion efficiency, equal to and probably exceeding the overall efficiency of the separate turbine and pump systems, given equal collector temperatures
- Low cost
- Great flexibility in accommodating different flow, suction lift, and discharge head requirements
- Great flexibility in utilizing available collector and condenser temperature
- Concept is based on existing state-of-the-art technology.

In the next subsection, the results of the Phase I development of a thermally driven direct acting organic Rankine pump are presented.

1.2 Development Program Description and Results

The objective of this program is to demonstrate the feasibility of the "Direct Acting Organic Rankine Pump" concept for use as a low temperature thermally driven irrigation pump. The program consists of four major tasks:

- Establish design specifications for the proposed pump concept
- Design and analysis of an experimental breadboard pump
- Perform a laboratory test program to verify analyses
- Design and analysis of a prototype pump.

Prior to establishing the design specification, an agricultural pump market analysis was performed to provide a basis for determining the solar pump economic potential and design requirements. It was determined that there exist two broad categories of agricultural irrigation, sprinkler and gravity. Sprinkler irrigation comprises approximately 40 percent of the market and requires both high pressure and flow rate pump capabilities (125 hp, 220 ft of head at 1500 gal/min). Gravity irrigation requires less pumping capability (20 hp, 50 ft of head, 1000 gal/min). The solar pump can be designed to meet the requirements of gravity irrigation. The projected economic market (that is, current replacement rate of pumps used for gravity irrigation) for the solar pump is only 7 percent of the nation's total agricultural irrigation pump market. However, the world market potential is three to four times that of the United States market.

The design and analysis of an experimental breadboard pump consisted of system optimization, critical components investigation, and preliminary pump design preparation. A computer model characterizing the thermodynamics and mechanical behavior of the Rankine cycle expander-pump system was developed as a design. With the aid of the computer program and marketing analysis, pump parameters such as size, speed, stroke, valve arrangement, and working fluid were determined. The critical components were expander piston seal, hermetic shaft seal, and expander intake and exhaust valves. A small laboratory model of the direct acting Rankine cycle pump was designed. The laboratory pump was designed for an output power of 1/2 hp at 50 to 70 ft discharge head, and a flow rate of 30 to 40 gal/min. A schematic representation of the laboratory pump is presented in Figure 5. The key differences between the laboratory pump and the original concept as presented in Figure 2 are the following:

- The laboratory pump uses solenoid actuated intake and exhaust valves on the Freon expander rather than mechanically actuated valves
- The laboratory pump uses a weight to power the exhaust stroke rather than the spring.

Table 2 lists the nominal design specifications for the laboratory pump.

The objectives of the laboratory test program were to demonstrate the direct acting organic Rankine pump concept, verify preliminary design and analysis, and to provide a basis for prototype design. The laboratory test program resulted in the following:

- Concept of a thermally driven direct acting organic Rankine cycle pump was successfully demonstrated

FIGURE 5. Direct Acting Rankine Cycle Experimental Pump

Table 2. Laboratory Pump Design Specification

Single cylinder, single acting, free piston
 Poppet valves, solenoid actuated
 Carbon impregnated vespel expander piston ring
 High strength polymer water piston cup seal

	<u>Design</u>	<u>Off Design Limit</u>
Bore, in.	3.5	-
Stroke, in.	5.2	7
Displacement, in. ³	50.0	67
Speed, cpm	158.0	240
Inlet temperature, °F	180.0	-
Inlet pressure, psia	80.0	-
Cutoff, percent	38.0	-
Flow rate, lbm/hr	115.3	300
Water discharge head, ft	50.0	-
Water flow rate, gal/min	33.6	-
Power output, hp	0.47	1
Efficiency		
Rankine (percent)	83.0	-
Carnot (percent)	68.0	-
Overall (percent)	11.6	-

- Overall pump efficiency is 2.9 percent which is significantly less than the design efficiency of 11.6 percent
- Evaluation of laboratory pump off-design conditions substantiated the validity and usefulness of the dynamic model as a design tool and diagnostic tool
- It is projected that with the incorporation of laboratory pump modifications, the overall pump efficiency will increase from 2 to 3 percent to 6 to 8 percent.

A conceptual prototype design similar to the laboratory pump except that the valves are integrated into the pump was prepared. The projected pump isentropic expander efficiency is 64 percent and the estimated overall thermal efficiency is 11.2 percent. A schematic illustrating the proposed pump is shown in Figure 6. The system cost of the 1 hp prototype pump is \$5,460.

Finally, a proposed Phase II program plan which includes five tasks designed for the successful design, fabricating, and testing of a 1 hp prototype pump system is presented.

FIGURE 6. Direct Acting Rankine Cycle Prototype Pump

- Materials study
- System dynamic study and system performance optimization
- Prototype detailed design, fabrication and testing
- Market assessment.

2. AGRICULTURAL PUMP MARKET ANALYSIS

2.1 Introduction

Despite all the attention given the sun over the past few years, serious efforts to harness the sun's energy for irrigation have been few. Only recently has there been a major attempt by private firms and government agencies to find a way of using sunlight to power the nation's irrigation systems.

Three major irrigation projects have been started thus far; two in Arizona and one in New Mexico. Each employs concentrating solar collectors providing heat upwards of 500°F to a conventional Rankine cycle turbine driving a conventional pump. Power ratings have increased from 10 to 50 to 200 hp in an attempt to meet the requirements of deep well turbines which account for the bulk of the nation's \$1 billion energy bill for irrigation. Although conversion efficiencies have been high, the costs of demonstration installations also have been quite high. Even allowing for learning curve cost reductions, systems of this type show little promise of being economically competitive with electric or diesel driven pumps.

A direct action solar driven organic Rankine pump has been proposed that operates in the lower temperature range of flat plate solar collectors in an effort to more economically meet the nation's irrigation needs. The pump is simple in design and has a maximum power rating of 25 hp.

2.2 Purpose

The objective of the market analysis is to investigate the nation's agricultural irrigation market to provide a basis for determining the solar pump's economic potential and design requirements. Specifically, the market was analyzed to determine the following:

- General categories of agricultural irrigation
- The general category(s) where the solar pump could be used
- The economic potential of the pump in its intended use
- The specific design requirements that optimize the pump's potential in the market
- Possible alternative uses not readily foreseeable.

2.3 Summary

There are two broad categories of agricultural irrigation, sprinkler and gravity. Sprinkler irrigation comprises approximately 40 percent of the market and requires both high pressure and flow rate pump capabilities (125 hp, 220 ft of head, 1500 gpm). Gravity irrigation requires less pumping capabilities (20 hp, 50 ft of head, 1000 gpm).

The solar pump can be designed to meet the requirements of gravity irrigation.

The projected economic market (that is, current replacement rate of pumps used for gravity irrigation) for the solar pump is only 7 percent (dollar volume approximately \$14 million annually) of the nation's total agricultural irrigation pump market.

World market potential is 3 to 4 times that of the United States market.

2.4 Detailed Analysis

A survey of the agricultural irrigation market was conducted by the Farmer Magazine during the winter of 1976. The survey provided a profile of irrigation, farms, ownership of general farm and irrigation equipment, and buying intentions in a number of selected product areas. Tables 3 through 8 were excerpted from this survey and are discussed in the following sections.

2.4.1 Land Use

Fifty percent of all the cultivated acreage in the United States is presently under irrigation. An additional 25 percent is suitable for future irrigation. The majority of irrigated acreage lies within the Midwest farm region east of the Rocky Mountains and west of the Mississippi River. In eight states (Texas, Oklahoma, Kansas, Colorado, Nebraska, Wyoming, North and South Dakota), 24 million acres were irrigated in 1977 or about 50 percent of the irrigated acreage in the United States.

The use of irrigation in this region has grown considerably during recent years. In 1956, the same eight states reported 13.8 million acres irrigated, so irrigated acreage in this region has doubled during the past 21 years.

2.4.2 Irrigation Methods

There are two major categories of agricultural irrigation; sprinkler irrigation and gravity irrigation. Tables 3 and 4 list the equipment types and acreage irrigated by these two methods.

Table 3. Sprinkler Irrigation Equipment Ownership Survey Result

	<u>Percent Reporting</u>	<u>Number of Owners</u>	<u>Average Owned</u>	<u>Total Units Owned</u>	<u>Average Acres Per Unit</u>	<u>Total Acres</u>
Center Pivot						
Electric	18.6	22,320	2.09	46,649	146	6,810,725
Hydraulic	10.9	<u>13,164</u>	2.16	<u>28,434</u>	151	<u>4,293,570</u>
Total Center Pivot		35,484		75,083		11,104,295
Travelling gun	3.8	4,560	1.40	6,384	124	791,616
Boom Type	1.4	1,680	3.20	5,376	19	102,144
Side Roll	9.6	11,520			195	2,246,400
Later Roll with Trail Tubes	2.5	3,000			52	156,000
Tractor Tow Line	5.8	6,960			125	870,000
Hand Move	15.9	19,080			158	3,014,640
Solid Set	2.3	2,760			95	262,200
Drip	0.4	480			134	<u>64,320</u>
TOTAL SPRINKLER ACREAGE						18,611,615

Table 4. U.S. 1976 Acreage

	<u>Percent Reporting</u>	<u>Average Acres</u>	<u>Number Owners/ Operators</u>	<u>Total Acres</u>
Total acres owned or operated	98.3	2,019	117,960	238,161,240
Acres under cultivation	95.8	839	114,960	96,451,440
Acres under irrigation	89.4	449	107,280	48,168,720
Acres suitable for future irrigation	60.3	330	72,360	23,878,800

Sprinkler irrigation accounts for approximately 43 percent of the total agricultural irrigation market. The most widely used sprinkler irrigation method is center pivot irrigation, accounting for about 60 percent of the sprinkler market. An estimated 88.9 percent of the increased irrigated acreage in Nebraska between 1972 and 1975 is attributable to the growth of center pivot irrigation.

Center pivot irrigation is designed around a central water/pressure source (usually a deep well turbine). A radial arm of high pressure pipe fitted with special nozzles is mounted on a wheel assembly about 6 to 8 ft tall. This arm and wheel assembly extends outward up to a half mile from the center pivot. Water is distributed as the arm sweeps out a circular path around the center pivot. High pressures and flow rates are required. In general, if a grower must go to the expense of installing a deep well turbine (100 to 500 ft lifts), a center pivot irrigation system provides a more efficient, uniform methods of irrigation.

Gravity irrigation, Table 5, is subdivided into furrow and flood irrigation. Flood irrigation usually does not require pumps or piping systems. Water from a common reservoir gravity drains through ditches or trenches covering the field. Furrow irrigation is similar, but a piping system carries the water to the field. Pumping systems are required to lift water from shallow wells (50 to 100 ft) or nearby streams, and pump it to a common reservoir. Gravity irrigation accounts for 57 percent of the nation's irrigated acreage.

Table 5a Gravity Irrigation (Furrow)

	<u>Percent Reporting</u>	<u>Average Acres</u>	<u>Total Operators</u>	<u>Total Acres</u>
Gated Pipe				
Aluminum	44.7	278	53,640	14,911,920
Plastic	1.6	40	1,920	76,800
Automated Gated Pipe				
Aluminum	1.1	880	1,380	1,214,400
Plastic	1.0	354	1,200	424,800
Siphon Tube from Open Ditch	16.4	188	19,680	<u>3,699,840</u>
Total Gravity Irrigation (Furrow)				19,902,960

Table 5b Gravity Irrigation (Flood)

	<u>Percent Reporting</u>	<u>Average Acres</u>	<u>Total Operators</u>	<u>Total Acres</u>
Border or Strip Flood	13.1	210	15,720	3,301,200
Dead Level	2.2	212	2,640	599,680
Terraced or Contour	3.6	267	4,320	<u>1,153,440</u>
Total Gravity Irrigation (Flood)				5,054,320
Total Sprinkler Irrigation				18,611,615
Total Gravity Irrigation				<u>24,957,280</u>
Total Irrigation				43,568,895

2.4.3 Source of Water

Table 6 provides a breakdown, by distribution method, of the ownership and the acreage of ground water irrigated farms. Surface water sources account for 27 percent of agricultural irrigation. The majority of surface source irrigation water is distributed by gravity.

Tables 7 and 8 provide estimates of the total number of pumps presently owned, under the general categories of deep well turbines, submersibles, and surface source pumps. A percentage breakdown is as follows:

Deep Well Turbines	67 percent
Surface Source Pumps	23 percent
Submersible Pumps	10 percent

Table 6. Surface Source

	<u>Percent Reporting</u>	<u>Average Acres</u>	<u>Number of Users</u>	<u>Total Acres</u>
<u>Not Controlled by District</u>				
Flood Only	10.2	277	12,300	3,407,100
Stationary Pump	6.6	160	7,920	1,267,000
Portable Pump	6.8	161	8,160	1,313,760
Floating Pump	0.6	98	720	70,560
				<u>6,058,420</u>
<u>Controlled by District</u>				
Flood Only	14.3	223	17,160	3,826,680
Stationary Pump	6.8	201	8,160	1,640,160
Portable Pump	3.3	85	3,960	336,300
Floating Pump	0.3	38	360	13,680
				<u>5,819,820</u>
Total Acres Irrigated From Surface Source				11,875,240

Table 7. Ground Water Source Pumps Ownership Statistics

	<u>Percent Reporting</u>	<u>Average Owned</u>	<u>Total Owners</u>	<u>Total Owned</u>
Deep Well Turbine				
Gas Engine	2.7	1.47	3,288	4,833
Diesel Engine	15.4	1.77	18,528	32,906
LP Gas Engine	13.2	1.80	15,936	28,684
Natural Gas	22.6	2.97	27,192	80,760
Electric Motor	31.7	3.22	38,100	<u>122,682</u>
Total Deep Well Turbine				269,944
Submersible Pump				
Gas Engine	0.4	1.00	516	516
Diesel Engine	1.1	1.25	1,344	1,680
LP Gas Engine	1.1	1.25	1,368	3,146
Natural Gas	0.6	1.75	720	1,260
Electric Motor	8.8	3.13	10,560	<u>33,053</u>
Total Submersible Pumps				40,743
Total Ground Water Source, Pumps			116,552	310,687
Tailwater Recovery Systems				
Own and Use Now	18.8	1.89	22,560	42,638
Plan to Build 1977	9.5	1.38	11,400	15,732

Table 8. Pumps and Irrigation Pumping Plants
Ownership Statistics

	<u>Percent Reporting</u>	<u>Average</u> <u>Pumps</u>	<u>Number Who Own or Plan to Buy</u>	<u>Total</u>
Gear Heads				
Own	40.3	3.85	48,360	186,186
Plan to Buy	3.9	1.59	4,680	7,441
Bowls				
Own	45.5	5.51	54,600	300,846
Plan to Buy	4.9	2.24	5,880	13,171
Stationary Centrifugal				
Own	15.6	2.78	18,720	52,042
Plan to Buy	2.3	1.19	2,760	3,284
Portable Centrifugal				
Own	15.3	1.61	18,360	29,559
Plan to Buy	2.2	1.60	2,640	4,224
Floating Pump				
Own	4.8	1.84	5,760	10,598
Plan to Buy	1.0	1.30	1,200	2,280

Irrigation Pumping Plants

Combustion Engine				
Own	42.1	2.89	50,556	146,106
Plan to Buy	4.8	1.58	5,760	9,100
Electric Motor				
Own	35.6	3.29	42,768	140,706
Plan to Buy	4.8	1.27	5,760	7,315

Existing dams and reservoirs presently irrigate 7.4 million acres or 17 percent of the nation's irrigated acreage.

2.4.4 Irrigation Pumping Equipment

2.4.4.1 Size

Agricultural irrigation pumping methods can be roughly divided into two groups, deep well turbines and surface pumps.

According to Irrigation Age Magazine, the typical deep well turbine is approximately 125 hp, provides 220 ft of head, and a flow rate of 1500 gpm. Turbines of 400 hp, providing 500 ft heads at 5000 gpm are not uncommon. Conversely, surface source pumps range from 15 to 25 hp, provide 30 to 50 ft heads, and flow rates of 800 to 1500 gpm.

2.4.4.2 Cost

The larger horsepower turbine pumps cost approximately \$50 per horsepower. The cost of a typical 125 hp pump is \$6250. The smaller surface water source pumps cost approximately \$70 per horsepower. For example, a 20 hp pump would cost \$1400.

2.4.4.3 Replacement

The projected annual replacement rate of deep well turbines is approximately 16 percent. This rate includes drilling new deep well turbines, reconditioning old pumps, and purchasing new pumps. The purchase of new pumps (which includes drilling new deep well turbines) is approximately 12 percent.

The purchase of new surface source pumps is estimated at 11 percent. No data were available on reconditioning. Because of the comparatively low cost, reconditioning may not be economically viable. A summary of deep well turbine versus surface source pump statistics including projected costs is contained in Table 9.

2.4.5 Estimated Value of Market

Based on the survey conducted by Farmer Magazine, the market for a solar powered pump under 25 hp is limited compared to deep well turbines. The market potential, however, is not as bleak as the survey suggests.

According to Paul Fischback, Extension Irrigationist at the University of Nebraska, gravity irrigation will supply a greater percentage of the nation's future irrigation needs than it has in the past. Because of the increased cost of energy and money, the rate of construction of new center pivot irrigation systems will decline.

Table 9. Summary of Irrigation Pump Costs

	No. of Pumps	Annual Replacement Rate (%)	Average HP	Cost Per Unit (\$)	Annual* Energy Cost (\$)
Deep Well Turbine	269,000	12	125	6250	5600
Surface Source Pump	91,000	11	20	1400	900

* Based on 100 days per year, 24 hours per day at \$0.025 per kW-hr

An additional U.S. market that shows promise but requires further investigation is supplying drinking water for the cattle industry. Pumping requirements are similar to surface irrigation but more important, the water supply is required in areas that are isolated from conventional energy sources.

Little detailed information is available on global irrigation. General observations from Ron Larson, Associate Editor of Irrigation Age, indicate that the world market would be very receptive to low pressure, low head solar powered pumps for the following reasons.

1. Mr. Larson estimates that the world market greatly favors surface irrigation systems as opposed to center pivot irrigation systems.
2. Underdeveloped nations lack the capital required to construct and maintain a large energy consuming center pivot system.
3. World irrigation is labor intensive. Labor is cheap compared to our standards. Energy sources are extremely expensive or non-existent.

2.4.6 Prototype Pump Requirements

Based on the results of the agricultural irrigation market survey, the solar pump will be designed to operate from surface water sources. From discussions with Ron Ross, editor of Irrigation Age Magazine and with Paul Fischback, Extension Irrigator at the University of Nebraska, the majority of surface source

irrigation needs can be met with 15 to 25 hp pumps. Lift requirements are 30 to 50 ft with enough reserve head capability to overcome pipe friction. Required flow rates range from 800 to 1500 gpm.

3. SYSTEM OPTIMIZATION

3.1 Introduction

Several Rankine cycle working fluids were evaluated over a 100°F to 300°F temperature range. The study was performed to compare the ideal expander/pump efficiency expected from different working fluids, to determine if any fluid offered superior efficiency, and to provide a basis for estimating ideal system cost.

The 100°F to 300°F temperature range was chosen because less expensive flat plate solar collectors can be used instead of focusing collectors. Solar collector cost is by far the dominant factor in total system cost, consequently solar collector area must be minimized to reduce this cost. The thermal performance of flat plate, evacuated tube, and focusing collectors are evaluated over a 100°F to 300°F temperature range. These collectors represent the present state-of-the-art in cost and efficiency of commercially available collectors.

Ideal Rankine cycle fluid efficiency increases with increased operating temperature. Solar collector performance, however, decreases with increased operating temperature. Optimum operating temperature balances increased expander/pump efficiency with decreased collector efficiency, maximizing system efficiency and reducing overall collector cost.

System optimization applies only to those systems where heat generation is an added cost. Waste (or "free") heat sources should be utilized at the maximum temperature possible, maximizing expander/pump efficiency.

3.2 Expander Performance

The ideal thermal efficiency of several operating fluids is determined over a temperature range of 100°F to 300°F. Saturated vapor at collector outlet temperature is expanded isentropically. Expander outlet temperature is assumed equal to condenser temperature and maintained constant at 70°F. Representative curves are presented in Figure 7.

The actual thermal efficiency of the expander pump package was not estimated. Any change in engine efficiency is assumed to cause a corresponding change in overall system efficiency regardless of solar collector type.

FIGURE 7. Ideal Engine Efficiency Versus Inlet Temperature

3.3 Collector Performance

Several types of solar collectors are presently available, including flat plate, evacuated tube, and focusing. The thermal performances of various commercially available collectors representative of each type are shown in Figure 8 as a function of output temperature for constant ambient conditions. The collector efficiencies are based on the thermal performance curves in the product specification brochures for these collectors.

Flat plate and focusing collectors are capable of using expander operating fluid as the solar collector circulating fluid. Evacuated tube collectors cannot withstand the pressure requirements and require an additional heat transfer loop. Using expander fluid as collector fluid reduces collector efficiency by approximately 10 percent compared to water as a circulating fluid. Water is therefore used as the circulating fluid for each collector for comparison purposes. The economics of an additional heat transfer loop are discussed in later sections.

The ambient conditions (insolation, ambient temperature, inlet temperature) listed in Figure 8 are typical of those encountered in the south western and mid-western states where the pump would be applied. The inlet temperature to the collector (70°F) is based on 65°F cooling water supplied to the condenser by the pump.

The flat plate collector provided by Solar Energy Products, Inc. (SEP) of Gainesville, Florida is typical of the majority of flat plate collectors and available in market. The collector has been designed for heating domestic hot water. Thermal efficiencies are relatively high (above 70 percent) in low temperature applications (120°F to 160°F). Costs vary from $\$10/\text{ft}^2$ to $\$15/\text{ft}^2$. The cost of the SEP solar collector panel, $\$13/\text{ft}^2$ is used for comparison.

The evacuated tube solar collector produced by Owens-Illinois (SUNPAK) is non-tracking like flat plate collectors. Higher thermal efficiency (above 50 percent) than flat plate collectors can be achieved at temperatures near 300°F . The cost of the unit is approximately $\$20/\text{ft}^2$.

Two focusing collectors were evaluated. The Solar Kinetics Corp. collector is marketed for domestic hot water heating. ACUREX Corp. collector is designed for industrial heating and cooling applications. The Kinetics collector has not been used in applications requiring temperatures in excess of 240°F , while the ACUREX collector has been used in applications in excess of 500°F . The cost for the Kinetics collector including tracking device is $\$21.50/\text{ft}^2$. ACUREX system costs vary depending on installation and system size. It is estimated that the ACUREX collector costs are about the same as the Kinetics collector. The Kinetics collector was eliminated from further evaluation since its performance was less than that of SUNPAK.

FIGURE 8. Solar Collector Efficiency Versus Output Temperature at Constant Insolation

3.4 Overall System Performance

Optimum collector operating temperature balances resulted in increased pump-expander efficiency and decreased collector efficiency with increasing temperature. Freon -11 was used as the operating fluid to develop the overall system performance curves. These curves are presented in Figure 9.

The flat plate collector system (SEP) exhibits optimum efficiency at 200°F. Reference 1 corroborates this finding and estimates optimum efficiency of the evacuated tube and focusing collectors at 300°F and 600°F, respectively.

Despite the increased efficiency of the evacuated tube (SUNPAK) and focusing collectors (ACUREX) at 200°F, the costs of these collectors are 44 percent and 58 percent greater than the flat plate collector, respectively. Ideal system costs for a 1 horsepower engine are presented in Figure 10.

FIGURE 9. Overall Ideal Solar Collector - Expander Efficiency Versus Outlet Temperature at Constant Insolation

FIGURE 10. Solar Collector Cost Versus Outlet Temperature

4. PUMP DESIGN STUDIES

4.1 Introduction

Results of the agricultural pump market analysis and system optimization study dictated the prototype solar pump's overall design requirements. Significant flexibility, however, still existed in the detailed pump design. Size, speed, stroke, valve arrangement, and working fluid are major design parameters that were still undefined. These options needed to be investigated before any major detailed development effort could be initiated.

Several sizing estimates were performed for a 1 hp pump using Freon-11 and Freon-114 at various displacements and speeds in order to obtain a preliminary design. In addition, intake and exhaust valve pressure drop calculations were performed to estimate valve areas required. This information along with representative design sketches fixed the size of the expander cylinder.

Refined estimates were performed based on procedures outlined in Reference 2. This procedure estimates power, mass flow, and efficiency based on valve characteristics, clearance volume, pressure ratio and speed. These estimates in conjunction with the previous discussion fixed the major operating parameters as follows:

Working fluid	- Freon -11
Supply temperature	- 220°F
Supply pressure	- 135 psia
Condenser temperature	- 70°F
Condenser pressure	- 13.4 psia
Expander cylinder diameter	- 3.5 in. I.D.

An important design objective is to minimize the pump speed which will enhance the pump reliability. However, sufficient energy must be stored in the reciprocating mass to force the piston to travel through a fuel expansion stroke. The dynamics of reciprocating engines using flywheel energy storage has been previously investigated; however, the dynamics of a free piston, using a compressible gas as the driving force, acting on an incompressible liquid has not been investigated. The documenting and analoging of this behavior were of primary importance in the development of the direct acting organic Rankine cycle pump.

4.2 Objective

The objective of the pump design studies is to prepare a preliminary design of the direct acting organic Rankine cycle pump. Critical components of the unit are identified as well as potential mechanical and material problems. Preliminary design is directed primarily toward the requirements of the laboratory pump; however, the requirements of the larger capacity prototype pump were also taken into consideration.

4.3 Summary

A computer model characterizing the thermodynamic and mechanical behavior of the Rankine cycle expander-pump system was developed as a design aid. The model is capable of predicting a complete time history of expander and water cylinder pressure, expander cylinder temperature, piston velocity and displacement, and Freon vapor mass flow. The model has not been extensively verified.

Potential energy storage and pump suction are accomplished using a reciprocating mass. Use of the mass is considered a key design improvement because it provides a constant restoring force, can easily be adjustable, and eliminates a potential pump failure mode.

A simple, effective seal was developed to separate hermetically, the Freon working fluid from its surrounding environment. The seal is comprised of a pressurized oil bath trapped between two polyurethane shaft seals. An oil with low Freon solubility and low vapor pressure at room temperature is used.

A brief discussion of the stability and effects of Freon compounds is presented. Most of the commonly used construction metals can be used with Freon compounds at moderate temperatures. Considerable variation is found in the effects of Freon compounds on various non-metals. Compatibility should be determined for specific applications.

4.4 Detailed Analysis

4.4.1 Dynamic Computer Model

In order to understand the thermodynamic processes occurring in the Rankine cycle engine water pump system, a detailed analysis was performed. The equations resulting from the analysis were programmed for solution on a digital computer.

Figure 11 shows schematically the "free piston" engine-pump system. High pressure gas enters the gas cylinder and displaces both the gas piston and the water piston upward, thus forcing

water into the discharge line and raising the weight, W . At the upper end of the stroke the gas inlet valve is closed, the gas outlet valve is opened, and the weight forces the pistons to move downwards exhausting the gas to the condenser and drawing water into the water cylinder. This water will, in turn, be discharged during the next upward stroke.

The equations used in the analysis are:

- Newton's law of motion for the two pistons and the weight, W
- Gas flow equations for the gas inlet and outlet ports
- Waterflow equations for flow into and out of the water cylinder
- First Law of Thermodynamics for the changes in gas properties in the gas cylinder
- Equation of State for the working fluids.

Figure 11. Schematic of the "Free Piston" Engine-Pump System

During the power stroke, energy from the gas accelerates the pistons and weight, W, raises the weight W, compresses the spring k (if present), and displaces the water into the discharge line. The return stroke is powered by the potential energy which was stored in W during the power stroke.

Calculations have been performed to determine appropriate values of piston diameter, weight W, valve areas, valve timing, etc., for specified pump parameters such as, suction lift, discharge pressure and water flow rate.

Required data for a calculation are shown in Table 10.

Sample calculated results are shown in Figures 12 and 13.

A second analysis and associated computer program were developed for a novel version of the pump in which the pump exit was eliminated and the water was discharged into a long straight pipe. Because of the inertia of the water in the discharge line, when the water piston reversed direction, the direction of the flow of water in the discharge line remained the same but its velocity was reduced. The low pressure in the water cylinder caused water to flow in through the inlet check valve.

During operation, the model predicts a complete time history of vapor and water cylinder pressure, vapor cylinder temperature, piston velocity and displacement, and vapor mass flow. Graphic representation of the output available from the computer program is presented in Figures 12 and 13. Figure 12 shows a typical pressure-volume (P-V) diagram obtained from the program. Features of concern are intake valve throttling, amount of expansion, and exhaust valve throttling. Figure 13 shows typical pressure time (P-T) diagrams, which again indicate throttling losses and expansion, frequency, time required for the power stroke and return strokes, and water valve dynamics.

Initial results were not encouraging. The model required excessive reciprocating mass (200 lbm) and return spring force (400 lbf) to control speed (600 cpm) which was still too fast for the laboratory pump. Additional results reduced the speed to approximately 300 cpm but still required an excessive reciprocating mass and return spring force.

It was decided at this time to expand the scope of the dynamic model to include alternative methods of potential energy storage. Variations that were investigated were:

- Conventional flywheel
- Discharge pipe line inertance, and
- Vertical spring/mass arrangement.

Table 10. Pump Performance Model Input Data

Variable	Required Information
Working Fluid	Supply pressure and temperature, discharge pressure, thermophysical properties
Reciprocating Mass	Mass, return spring force and spring constant
Vapor system parameters	Piston diameter, clearance volume, intake and exhaust valve areas, intake and exhaust valve timings
Water system parameters	Piston diameter, suction lift, discharge head, inlet and outlet valve areas

The use of a conventional flywheel arrangement eliminated the need for a return spring, set the stroke, and provided a valve actuating mechanism. The flywheel also provided a standard from which alternatives could be evaluated.

The discharge pipe line inertance concept employed the water momentum built up in the water discharge pipe to provide part of the suction lift, and also eliminated the need for a large return spring. The disadvantage to this concept was the length of discharge pipe required.

The original spring/mass arrangement was re-oriented to the vertical. This concept used the weight of the reciprocating mass and gravity to provide the return stroke. A vertical orientation also reduced friction and side loading on support bushings.

Extensive studies were performed on the inertance concept and vertical pump concept. Both arrangements present equally viable operating characteristics. Table 11 depicts their specifications for the laboratory pump design. Studies of the flywheel arrangement were discontinued when both the line inertance and vertical mass systems yielded viable results.

Figure 12. Diagram of Cylinder Pressure Versus Volume
Predicted from Computer Model

Figure 13. Diagram of Pressure Versus Time Predected from Computer Model

Table 11. Laboratory Model Dynamics

<u>Parameter</u>	<u>Vertical Mass</u>	<u>Line Inertance</u>
Bore, in.	3.5	3.5
Stroke, in.	8	6
Displacement, in. ³	76	58
Speed, cpm	150	143
Reciprocating Mass, lbm	100	10
Discharge Pipe Length	N/A	200
Initial Return Spring Force lbf	N/A	25
Return Spring Constant lbf/in.	N/A	0
Water Discharge Head, ft	50	50
Water Flow Rate, gpm	50	35
Power Output, hp	0.7	0.5
Efficiency:		
Percent Rankine	88.5	89.1
Percent Carnot	66.4	66.8
Percent Overall	14.6	14.7

A comparison was made of the advantage and disadvantage of the vertical mass and line inertance concepts. The vertical mass concept was selected for continued detailed design and fabrication of the laboratory pump. The inertance concept was eliminated because of the length of discharge pipe required, necessity for a return spring, and increased sensitivity to operating parameter variations. The inertance system was evaluated further for consideration in prototype design.

The dynamic model was an extremely important design tool in the detail design of the laboratory pump. The model allowed precise "fine tuning" of the intake and exhaust vapor valve dynamics, inlet discharge water valve characteristics, vapor pressure and temperature, and reciprocating mass weight.

Results of the preliminary investigation fixed the laboratory pump expander cylinder size at a diameter of 3.5 in., speed at 15 to 200 cpm, Freon-11 as the working fluid, and maximum power at 1/2 hp. The reasons for setting these parameters were discussed in the preceding sections. Parameters that still needed to be resolved were:

- Reciprocating mass weight
- Intake and exhaust water and vapor valve areas
- Intake and exhaust vapor valve actuation setpoints, and
- Water discharge head.

Pump speed is affected by supply pressure, intake supply pressure cutoff, reciprocating mass weight, and discharge head. Once a reasonable set of parameters has been selected, intake supply pressure cutoff is the controlling factor in determining pump speed. Figure 14 shows the change in frequency compared to supply pressure cutoff with all other controlling parameters maintained constant for a cutoff at 0.4 in. and 2.0 in. Figure 15 shows the change in frequency compared to a change in reciprocating mass. Similarly, changes in discharge head result in little frequency variation.

In addition to speed, cutoff controls vapor mass flow rate, stroke and corresponding water flow, and power. Assuming the vapor exhaust valve timing is actuated, a 25 percent decrease in supply pressure cutoff will result in approximately a 10 percent increase in frequency, a 20 percent decrease in vapor mass flow and stroke, and a 10 percent decrease in power.

Small changes in reciprocating mass or water discharge head affect pump operation very little. A 25 percent change in either of these parameters would have less than a 10 percent change on any operating characteristics. A 100 percent increase in reciprocating mass decreases power by 20 percent.

System optimization determined the system pressure and temperature ranges for the most cost effective operation. Variation in supply pressure is very important in determining overall system performance. Although it has very little effect on speed, supply pressure has a significant effect on all other operating characteristics. A 25 percent increase in supply pressure will result in a 25 percent increase in stroke, water flow, power, 13 percent increase in vapor mass flow, and 9 percent increase in efficiency. Sufficient flexibility in mechanical restrictions must be designed into the pump to allow for variations in supply pressure. Figure 16 shows the variation in stroke that results from a 25 percent increase in supply pressure.

Table 12 summarizes the effects that supply pressure, supply pressure cutoff, reciprocating mass weight, and water discharge head variations have on other operating characteristics.

The detailed design of the laboratory pump was accomplished using information from the parameter study and from the results of materials and mechanical studies which are presented in later sections. Table 13 lists the detail design specifications for the laboratory pump.

4.4.2 Mechanical Design Studies

During the mechanical design studies, emphasis was placed on the effect of the design features on the efficiency of the pump-expander package. The following aspects of the expander-pump system were addressed.

FIGURE 14. Frequency as a Function of Supply Pressure and Cutoff

FIGURE 15. Frequency as a Function of Mass

FIGURE 16. Displacement as a Function of Supply Pressure

Table 12. Independent Parameter Effects on System Dynamics

Supply Pressure Increase:	Increases stroke, water and power and to a lesser extent mass flow and efficiency. No change in frequency.
Supply Pressure Cutoff Increase:	Increases stroke, power, and mass flow. Decreases frequency and efficiency.
Reciprocating Mass Increase:	Increases mass flow. Decreases stroke, power, efficiency and to a lesser extent frequency.
Water Discharge Head Increase:	Increases mass flow and power. Decreases stroke and to a lesser extent frequency and efficiency.

- Design of the positive displacement cavities
- Expander valve actuation
- Water valve actuation
- Hermetically sealing the Freon system
- Potential energy storage
- Mounting support.

The design of the displacement cavities allows for considerable flexibility. Separate pistons, diaphragms, or intermediate fluids are possibilities. Rigidly connected pistons were selected as the best method for the developmental pump. Separate displacements only created additional variables that would have to be accounted for in analysis of the pump's operating characteristics. This selection should not be interpreted to mean that rigidly connected expander/water pistons is the most efficient method available. On the contrary, using a "buffer" fluid between the expander piston and water piston could have a dramatic effect on pump performance. Additional displacement cavity design considerations are discussed in the material section.

Table 13. Laboratory Pump Design Specification

Single cylinder, single acting, free piston
 Poppet valves, solenoid actuated
 Carbon impregnated vespel expander piston ring
 High strength polymer water piston cup seal

<u>Performance</u>	<u>Design</u>	<u>Off Design Limit</u>
Bore, in.	3.5	-
Stroke, in.	5.2	7.0
Displacement, in. ³	50	67
Speed, cpm	158	400
Inlet temperature, °F	180	250
Inlet pressure, psia	80	-
Cutoff, percent	38	28
Flow rate, lbm/hr	115.3	300
Water discharge head, ft	50	100
Water flow rate, gpm	33.6	-
Power output, hp	0.47	1.0
Efficiency:		
Percent Rankine	83	-
Percent Carnot	68	-
Percent Overall	11.6	1.32

Three mechanically actuated expander valve assemblies were designed and evaluated; axial spool, axial spool-poppet, and offset poppet. Figures 17 through 19 illustrate each concept, respectively. Each design has its own advantages and disadvantages. The key disadvantage to all three concepts is their "uniqueness" compared to commercially available valves. However, a new valve concept requires an extensive development program. The program workscope included a pump concept development program and not a valve development program. The mechanically actuated valves were shelved for the moment, in favor of commercially available solenoid valves.

Microswitch actuated solenoid valves offer the advantage of "infinite" expander valve timing, simple adjustment, proven reliability, and low cost. The valves are mounted external to the pump and as close as possible to the expander in order to reduce the clearance volume. Valve actuation was performed external to the pump by means of the extension of the piston connecting shaft. As a result, valve adjustment is possible during pump operation.

Maximum response time of the solenoid valves was approximately 10 msec. In order not to severely restrict vapor flow during the intake stroke, intake valve open time was required to be approximately 50 msec; 10 msec opening ramp, 30 msec dwell, and 10 msec closing ramp. Since intake accounts for 10 to 20 percent of total cycle time, pump cycle time was now restricted to 250 to 500 msec, 120 to 240 cpm. The ultimate effect of this slow cycle rate was an increase in the required amount of reciprocating mass. A secondary effect of the slower cycle rate is increased heat transfer to the cylinder wall, and a decrease in pump efficiency. The amount of heat transfer is unknown and not readily analyzed.

For similar reasons, externally mounted, commercially available check valves were selected for the water side valves. Optimizing water side valves was not the objective of the pump development program.

The dynamic response of swing flap check and ball check were evaluated. They are slow acting and require a substantial pressure differential, but are adequate for the laboratory test assembly. Ball check valves were selected because of lower cost.

It is important that an effective hermetic seal be employed in the pump. Leakage of air or water vapor into the system, or Freon out of the system results in decreased pump performance. Maintenance costs increase and reliability decreases. An effective hermetic seal must be developed.

FIGURE 17. Axial Spool Valve Assembly

FIGURE 18. Axial Spool-Poppet Valve Assembly

FIGURE 19. Off-Set Poppet Valve Assembly

Many seal designs were considered. Two were extensively evaluated;

- Bellofram rolling diaphragm
- Oil-filled shaft seals.

The Bellofram rolling diaphragm shown in Figure 20 does not have a mechanical spring gradient, break-out friction effects, possesses low hysteresis qualities, and is capable of a long stroke.

Bellofram Corporation performed a design analysis of a diaphragm for the laboratory pump using a 3.5 in. piston, 2.0 in. stroke, 150 cpm, and Freon -11 as the working fluid. Fatigue life of the diaphragm was estimated at 20,000 cycles. At 100 cpm the lifetime is only three hours. Reducing the stroke to 1.0 in. increased the lifetime to nearly 10 hrs which is still inadequate for the laboratory pump.

Figure 21 is a schematic representation of an oil-filled shaft seal. Oil pressure is maintained slightly higher than Freon condenser pressure, so that oil can leak in to the system, but Freon cannot leak out of the system. This system performs well in theory, but the practical aspect is unproven. Friction effects, required oil pressures, and leakage rates can only be estimated, and could vary significantly.

The oil-filled shaft seals concept was selected because of simplicity, cost, and feasibility. It was considered a critical component for laboratory evaluation.

A number of energy storage systems were evaluated;

- Linear inertia (large piston mass)
- Rotational inertia (conventional flywheel)
- Hydraulic accumulator
- Spring mechanisms
- Discharge line inertance.

An energy storage system is required to store high pressure vapor during the initial portion of the stroke so that it can be used during the latter portion of the stroke when the vapor pressure is less than the water head pressure. The energy storage system smooths the power stroke and increases efficiency.

Potential energy storage is accomplished in the laboratory pump using linear inertia in the form of an attached 100 lbm reciprocating mass. This concept was selected because it is:

- Simple
- Reliable
- Compatible with original concept, and
- Meets the required laboratory design parameters.

FIGURE 20. Bellofram Rolling Diaphragm Seal

FIGURE 21. Design Sketch of Hermetic Shaft Oil Seal

Potential energy storage in the prototype pump is accomplished in a similar manner, except the mass is incorporated into the water and vapor pistons. This was made possible by increasing pump speed, and reducing required mass. A more detailed discussion is contained in Section 6.4.

Mounting system dynamic imbalance forces are not considered of great importance to the laboratory pump because of the low cycle rate involved. Design of the support fixture was accomplished primarily with ease of access to the pump and test assembly components in mind.

The pump support stand was constructed in the form of an elevated table. The top was made of $\frac{1}{2}$ in. steel plate with a 10 in. diameter hole in the middle in which the pump was bolted. The legs were made of 3 in. by $\frac{3}{8}$ in. steel angle which were welded to the top and to another $\frac{1}{2}$ in. steel base plate. The base plate was in turn bolted to the shop floor. Shock pads of $\frac{1}{2}$ in. rubber were used between pump and table, and base plate to dampen any vibration. The table stood four feet tall.

4.4.3 Material Studies

The materials of construction can have a dramatic effect on cost, thermal performance, mechanical performance, and life-time of the expander-pump system. Material selection was carefully evaluated.

Studies were conducted in four specific areas of expander-pump design:

- Seal materials
- Expander system materials
- Pump system materials
- Support structure.

The selection of materials for the expander piston seal and hermetic shaft seal was the most important aspect of the Materials Study task. The selection of expander system materials needed to be evaluated with respect to the effects of Freons. Pump system and support structure materials selection were straightforward.

There were two methods available for designing the expander piston seals, lubricated or unlubricated. Lubricated systems exhibit less sliding friction and generally longer wear, but require an oil sump, oil pump, and distribution system. There are also a variety of materials available for unlubricated sliding service that offer low friction and long wear. An unlubricated system was preferred for the laboratory pump to reduce system complexity.

The optimum expander piston ring material must have a low coefficient of friction on steel, must tolerate temperatures of 250°F, and must tolerate exposure to Freon environments.

Most of the materials suitable for unlubricated service are plastics such as glass filled Teflon, Delrin, or Rulon. Freons attack these materials with varying degrees of severity, including swelling, softening, and embrittlement.

It is worthwhile to discuss Freon compounds and their effect on various materials.

The Freon products are organic compounds containing one or more carbon atoms and fluorine. Chlorine, bromine, and hydrogen atoms also may be present. General characteristics include non-flammability, low toxicity, thermal and chemical stability, high density, low boiling point, and low viscosity and surface tension. A brief discussion of their stability and effect on other materials is presented below. The DuPont Products Information Center has available more specific information regarding Freons.

Most of the commonly used construction metals such as steel, cast iron, brass, copper, tin, lead, aluminum - can be used satisfactorily with the Freon compounds. At elevated temperatures, some of the metals may enhance thermal decomposition of the compound. As general examples, Table 14 lists thermal stability data for some Freon compounds. Steels generally promote less decomposition than aluminum or copper alloys.

A brief summary of the effects of the Freon compounds on various non-metals is given in Table 15. Compatibility should be tested for specific applications.

Considerable variation is found in the effects of the Freon compounds on elastomers, depending on the particular compound and elastomer type. Comparison of the swelling of elastomers provides a relative indication of their suitability. Such a comparison is given in Table 16.

Information available from manufacturers (i.e., DuPont, Dow Corning) was generally for low temperature applications. There was no definitive information available for the particular application proposed. The information available was speculative at best.

The best materials identified are acetal resin thermal plastics impregnated with small amounts of TFE, such as Delrin AF, suitable for temperatures to 230°F, and polyimide resin based plastics, such as Vespel, impregnated with small amounts of molybdenum disulfide to reduce the friction coefficient.

Table 14. Thermal Stability of Freon Compounds

<u>Compound</u>	<u>Formula</u>	<u>Maximum Temperature for Continuous Exposure in the Presence of Oil, Steel and Copper (°F)</u>	<u>Decomposition Rate at 400°F in Steel (%/Year)</u>	<u>Temperature for First Trace of Decomposition in Quartz (°F)</u>
Freon 11	CCl_3F	225	2	840
Freon 113	$\text{CCl}_3\text{F}-\text{CClF}_2$	225	6	570
Freon 12	CCl_2F_2	250	1	1000
Freon 114	$\text{CClF}_2-\text{CClF}_2$	250	1	*
Freon 22	CHClF_2	300	*	550
Freon 502		300	*	*
Freon 113	CClF_3	300	*	*

* Not Measured

Table 15. Effects of Freon Compounds on Various Materials

<u>Material</u>	<u>Effect of Freon Compounds</u>
Teflon	No swelling observed when submerged in Freon liquids but some diffusion found with Freon 12 and Freon 22.
Polychlorotrifluoroethylene	Slight swelling but generally suitable for use with the Freon compounds.
Polyvinyl alcohol	Not affected by the Freon compounds but very sensitive to water. Used especially in tubing with an outer protective covering.
Vinyl	Resistance to the Freon compounds depends on vinyl type and plasticizer and considerable variation is found. Samples should be tested before use.
Orlon acrylic fiber	Generally suitable for use with the Freon compounds.
Delrin acetal resin	Suitable for use with the Freon compounds under most conditions.
Vespel polyimide resin	Suitable for use with the Freon compounds under most conditions.
Silicon Elastomers	Generally not suited for use with the Freon compounds.
Viton fluoroelastomer	May be suitable for use with some Freon compounds.

Table 16. Swelling of Elastomers by Freon and Other Compounds

Per Cent Increase in Length at Room Temperature

<u>Product</u>	<u>Neoprene GN</u>	<u>Buna N (butadiene/ acrylonitrile)</u>	<u>Buna S (butadiene/ styrene)</u>	<u>Butyl (isoprene/ isobutylene)</u>	<u>Polysulfide Type</u>	<u>Natural Rubber</u>
Freon 11	17	6	21	41	2	23
Freon 12	0	2	3	6	1	6
Freon 13	0	1	1	0	0	1
Freon 21	28	48	49	24	28	34
Freon 22	2	26	4	1	4	6
Freon 113	3	1	9	21	1	17
Freon 114	0	0	2	2	0	2
Freon 115	0	0	0	0	0.2	0
Freon 502	1	7	3	1.6	1.6	4
Freon 13B1	2	1	1	2	0	1
Freon 114B2	7	7	15	22	1	26
Freon C-318	0	0	0	0	0	0
Methyl Chloride	22	35	20	16	11	26
Methylene Chloride	37	52	26	23	59	34

The long lead time required for ordering piston rings fabricated from high performance plastics influenced the final decision for the laboratory pump. Graphite impregnated Vespel rings were available from Koppers with an approximately six to eight week lead time. The coefficient of friction (0.24) is higher than desired, but the thermal and mechanical properties are suitable for the 200°F to 250°F Freon environment.

It was discovered during another development program*, using Freon -11 as a working fluid, that Delrin AF is subject to swelling at 140°F. Delrin was used as the bushing and gear materials in a small gear pump used to pump Freon -11 condensate back to a Freon boiler. The pump bound up on a number of occasions using Rulon, Teflon, and finally Delrin AF. The degree of swelling was not measured.

A hermetic shaft seal was developed to isolate and contain the Freon working fluid from its surrounding environment. The seal was comprised of a pressurized oil bath trapped between two polyurethane shaft seals.

Polyurethane was selected as the seal material because of its toughness and abrasion resistance, particularly under high pressure loading applications and the abrasive environment expected. Materials such as Nitrate or Neoprene cannot withstand the dynamic loading, and Nylons or TFE cannot create as effective a seal.

Polyurethane can be used continuously at 200°F in contact with most fluids, with the exception of acids, ketones, and chlorinated hydrocarbons. There is limited data evaluating the effects of Freon compounds. Severe effects are not anticipated.

A commercially available refrigerant oil, SUNISO 4GS, was used to complete the essentially hermetic seal. SUNISO 4GS has a low Freon solubility and low vapor pressure at room temperature and is commonly used in refrigeration systems with maximum temperatures of 150°F. Shaft temperatures were not expected to exceed 120°F.

The hermetic seal was considered a critical component for laboratory testing.

Solenoid valve seals and discs were the only other critical portions of the expander. Seal material selection was generally limited to Buna "N", Teflon, or Ethylene Propylene from most commercial valve manufacturers. Buna "N" is the most compatible

* DOE Contract No. DE-AC04-77-CS04098; Development of a Passive Heating and Cooling System Using a Pump Heat Pipe

with Freon-11, exhibiting only slight swelling. Ethylene Propylene is unsatisfactory, and Teflon has behaved poorly in another FMA development program*, using Freon -11.

The major concern in the water pump system was corrosion, and the effect corrosion has on the water piston seal. Very few metals can be subjected to a water environment without some corrosion or film depositing. Hard chroming the interior was one alternative, but this is expensive and subject to abrasion from foreign matter in the water. Other alternatives were copper or copper alloys, or stainless steels, which again are very expensive. Non-metals such as fiberglass reinforced plastics were possibilities. These are less costly than the previous examples but are more subject to abrasive wear. It was concluded that cast iron was the most realistic candidate. Cast iron has been widely used in irrigation pumping applications for many years. Simple leather piston cup seals are generally used or high density elastomers.

A polyurethane cup seal impregnated with molybdenum disulfides to reduce sliding friction was selected as the water piston seal for the laboratory pump. This cup seal is a stock item, inexpensive, and readily available. Leather seals could be used just as easily without modifications.

Support structure for the expander-pump system is extremely important. For example, the main shell - expander and pump cylinders - serves as a support member, pressure vessel, and sliding surface on which the piston travels. For this application, low carbon steel is a cost effective material selection. However, using steel throughout the shell could contribute to large heat conduction losses from the hot expander and to the cold water pump end.

The main support structure, specifically the expander-pump cylinder head, was evaluated with respect to the following:

- Strength
- Thermal expansion
- Thermal conductivity
- Machining qualities
- Cost.

The most important considerations for the laboratory pump were machining qualities and thermal properties.

* Op. Cit.

Although its thermal conductivity is high, low carbon steel was selected for the expander-pump cylinders and heads because of

- Machining equipment
- Low thermal expansion coefficient
- Availability
- Inexpensive.

However, thermal insulation can be added to reduce heat transfer.

Fiberglass reinforced plastic is also a logical choice for the main support structure, for it would thermally isolate the expander from the water pump without sacrificing mechanical strength. Machining qualities, however, are questionable and thermal expansion is an order of magnitude greater than steel. Fiberglass can experience a 0.005 to 0.025 in. diameter increase from ambient to 200°F. Also plastic base piston seals exhibit a higher coefficient of friction on plastic than on steel. Fiberglass or similar materials are still a possibility for the prototype pump.

4.4.4 Laboratory Pump Design

A small laboratory model of the direct acting Rankine cycle pump was designed and tested. The objective of the design was that it aided in the development of the prototype pump in the following areas:

- Dynamic analysis
- Mechanical design
- Material selection.

Sufficient flexibility was designed into the pump and associated test loop to allow for minor changes in pump design or operating conditions if necessary.

There was no attempt to design the laboratory pump as a miniature version of the prototype pump. Rather, the laboratory pump was designed as a breadboard model representative of the concept. Stock items were used wherever possible to reduce cost and improve reliability. The most important criteria for the laboratory pump was that it be readily accessible for instrumentation and modification.

One piece of laboratory test equipment that limited the performance of the laboratory pump was the Freon boiler. A 10 kW electrically heated refrigerant vaporizer was purchased to simulate the solar collectors. At 200°F the maximum thermal efficiency achievable using Freon -11 was approximately 15 percent as is shown in Figure 7. Consequently, for a 50 percent conversion efficiency, the overall efficiency is 7.5 percent. This limits the power of the laboratory pump to 0.75 kW, or approximately 1 hp.

The laboratory pump was conservatively designed for an output power of 1/2 hp, at 50 to 70 ft discharge head, and 30 to 40 gpm.

A schematic representation of the laboratory pump is presented in Figure 22. The key differences between the laboratory pump and the original concept, as presented in Figure 2 are as follows:

- The laboratory pump uses solenoid actuated intake and exhaust valves on the Freon expander rather than mechanically actuated valves as shown in Figure 2
- The laboratory pump uses a weight to power the exhaust stroke rather than the spring shown in Figure 2.

The use of solenoid actuated intake and exhaust valves allowed flexibility in valve timing. For a given set of operating conditions, the opening, closing, and open or closed duration for each valve can easily be adjusted to any desired combination. The optimum valve timing for any given set of operating conditions can be determined experimentally. A prototype pump would use mechanically actuated valves, after the optimum valve timing had been determined with the experimental unit and/or computer simulation.

The use of a weight to provide energy for the expander exhaust/water intake stroke is considered a design improvement over the return spring shown in Figure 2 for the following reasons:

- The weight provides a constant restoring force
- The weight is inexpensive, simple, and easily adjustable
- A potential pump failure mode is eliminated (spring breakage).

In the laboratory pump, the weight was provided by a set of "platters" stacked on one another. By varying the weight, pump performance could easily be adjusted to suit existing conditions.

The most difficult task in designing the laboratory pump was keeping the speed down to a rate compatible with the solenoid actuated intake and exhaust valves. The valves limited pump speed to 120 to 240 cpm.

A trial and error method was used, varying expander valve timings, head pressure, and reciprocating mass on the dynamic model to arrive at the laboratory pump's present specifications. Table 17 lists the nominal design specifications for the laboratory pump.

Figure 22. Direct-Acting Rankine Cycle Experimental Pump

Table 17. Laboratory Pump Design Specification

Single cylinder, single acting, free piston
 Poppet valves, solenoid actuated
 Carbon impregnated vespel expander piston ring
 High strength polymer water piston cup seal

	<u>Design</u>	<u>Off Design Limit</u>
Bore, in.	3.5	-
Stroke, in.	5.2	7.0
Displacement, in. ³	50.0	67.0
Speed, cpm	158.0	240.0
Inlet temperature, °F	180.0	-
Inlet pressure, psia	80.0	-
Cutoff, percent	38.0	-
Flow rate, lbm/hr	115.3	300.0
Water discharge head, ft	50.0	-
Water flow rate, gal/min	33.6	-
Power output, hp	0.47	1.0
Efficiency,		
Rankine (percent)	83.0	-
Carnot (percent)	68.0	-
Overall (percent)	11.6	-

There were 3 items in the overall pump design that were considered critical components.

- Expander piston seal
- Hermetic shaft seal
- Expander intake and exhaust valves.

Although not considered part of the actual Rankine cycle pump design, erratic behavior of the solenoid actuated expander intake and exhaust valve could hamper or even cripple laboratory pump performance. The laboratory test program depended on stable solenoid valve performance.

Performance of the piston and shaft seals are critical to the continued development of the direct acting organic Rankine cycle pump concept. Performance of the laboratory pump is not expected to be greatly affected by seal performance. If the seals fail in the laboratory pump they can be replaced. Reliability in the prototype pump, however, is of utmost importance.

Another performance aspect that is important to the prototype pump but of little concern in the laboratory is heat transfer. Thermal losses are expected to be high in the laboratory pump because of the steel expander cylinder, and the low operating speed. Preliminary calculations revealed that losses on the order of 20 percent can be expected. This will in no way detract from the operating characteristics of the laboratory pump with the exception of decreasing thermal efficiency. The extent of the thermal losses will be determined by comparing predicted performance with actual performance.

5. LABORATORY TESTING

5.1 Introduction

A laboratory test program was prepared and a test facility was designed to evaluate the pump. A test loop had to be established that not only simulated solar collector and heater system dynamics, but also allowed control over each system and recorded each system's performance independent of each other.

5.2 Objective

The objectives of the laboratory test program were as follows:

- Demonstrate direct acting organic Rankine pump concept
- Verify preliminary design and analysis
- Provide a basis for prototype design.

It was important to achieve reasonable thermodynamic performance and stable operation for both design and off design conditions.

5.3 Summary

All of the laboratory test program objectives were achieved with the possible exception of thermal efficiency.

The pump was operated for 10 hrs during the three week test program. Stable operation was easily achieved under a variety of conditions. A summary of the completed test results is presented in Table 18.

Pump speed and power were low, and vapor consumption was significantly higher than projected, resulting in lower performance than predicted. This reduction in performance was traced to inefficiencies in the test equipment. The inefficiencies have been simulated on the computer model with close agreement to actual performance. Elimination of the inefficiencies will result in a 200 to 300 percent increase in pump performance. Modification to the test equipment and additional testing should be performed to quantify these projections.

Performance of critical components was more than satisfactory. No problems were encountered with either the vessel, expander piston rings or the oil-filled hermetic shaft seal. Oil usage in the shaft seal was virtually non-existent. Operation of the expander intake and exhaust solenoid-actuated valves was consistent and reliable, although not as efficient as expected.

Table 18. Summary of the Laboratory Pump Test Results

Run #	Ps (PSIA)	Ts (°F)	Cut-off (% of Stroke)	stroke (In)	Freq (cpm)	Mass flow (lb/hr)	Head (Ft)	Water flow (gpm)	Power (hp)	Mass (lbs)	Efficiency (percent)
NPC #1	90	190	47.2	4.5	90-100	193.4	33	15	0.125	110	1.7
RSEV-2	95	190	50	5	88	167.9	60	12	0.183	110	2.9
RSEV-3	95-100	190	32	3.5	116	162.8	42	11.8	0.125	130	2.1
RSEV-3-6	120	207	26.5	4.25	113	130	33	15.85	0.132	130	2.64
RSEV-4	95	190	34.6	3.25	115	130	42	10.8	0.114	130	2.3
RSEV-5	85	180	42.8	3.0	104	130	32	10.8	0.07	130	1.0

Water piston friction was greater than projected. The U-cup water piston seal was oversized for its intended application. No leakage was noted past the seal during the entire test program. The sealing capabilities of the U-cup can be decreased to improve performance.

5.4 Detailed Testing

5.4.1 Test Loop and Instrumentation

A schematic representation of the laboratory test equipment used to evaluate the experimental pump is presented in Figure 23. Photographs of the actual test equipment are shown in Figure 24 and 25. Key elements in the test set up include:

- A 10 kW freon boiler which simulates the solar collectors
- A condenser which condenses pump exhaust vapor back to the liquid state
- A condensate return pump which returns the condensed freon to the boiler
- A water discharge head control vessel which simulates a fixed discharge head.

The 10 kW electrically-heated freon vaporizer is ASME code approved for 600 psig working pressure, and can be used with Freon-11, -12, -22, or -114. Working pressure is temperature/pressure controlled with a low watt density electric (440/3/60) immersion heater. Saturation temperatures up to 300°F can be simulated with the vaporizer. Figure 26 is an engineering drawing of the Freon boiler.

The condenser is a commercially available, water-cooled shell and tube refrigerant condenser. The unit is rated at five tons, and can be used with any of the previously mentioned Freon compounds. Maximum (shell side) vapor temperature and pressure are 150°F and 350 psig, respectively. Maximum (tube side) water pressure is 150 psig.

The condensate return pump is a positive displacement pump consisting of a separate motor, pump, and V-belt drive. A positive displacement pump was selected because it was possible to measure Freon flow rate based on pump speed. This pump with a Viton shaft seal has previously been used for Freon applications. Problems with Freon (swelling etc.) are not anticipated in this application.

Figure 23. Solar Agricultural Pump Laboratory Test Schematic

Figure 24. Laboratory Pump Test Assembly

Figure 25. Photograph of the Solar Agricultural Pump

FIGURE 26. Schematic of Freon Boiler

The water discharge head control vessel is a one cubic ft pressurized expansion tank. By adjusting the water relief valve immediately adjacent the control vessel to the desired discharge head, and pressurizing the vessel with air until water just begins to leak past the relief valve, a constant discharge head can be maintained on the pump. Discharge heads to 100 ft can easily be simulated with the control vessel.

Instrumentation is provided throughout the test loop to monitor system performance. Pressure and temperature gages are located at the vaporizer outlet, expander inlet, expander exhaust, and condenser shell. Temperature gages are located at the condenser liquid Freon outlet, and condenser cooling water pump head and the discharge head control vessel. Detailed instrumentation and piping schematics of the Freon and water test loops are presented in Figures 27 and 28, respectively.

Laboratory pump water flow rate was measured with a stop watch and bucket. Condenser cooling water flow was measured in a similar manner.

Vapor flow rate could not be measured directly, and was measured indirectly by either monitoring Freon vaporizer "on" time and power consumption, condensate pump and "on" time, or condenser cooling water flow and temperature rise. Condenser cooling water flow rate was used in most cases because vaporizer and condensate pump "on" times were difficult to monitor.

5.4.2. Laboratory Test Plan

Overall objectives of the laboratory test program have been stated in Section 5.2. Specifically, the test program has been established to achieve the following:

- Reasonable overall performance
- Critical component performance
- Stable operation
- Start up characteristics
- Ability to handle off-design conditions
- Thermal to actual hydraulic conversion efficiency.

In order to evaluate pump performance in a timely and efficient manner, a detailed test plan was developed. The test plan ensured that all aspects of pump performance including critical component testing would be evaluated.

Laboratory testing followed the test plan and included initial checkout of procured or fabricated parts through final pump demonstration. A complete listing of the Detailed Test Plan follows.

FIGURE 27. Freon Test Loop

FIGURE 28. Water Test Loop

5.4.2.1 Laboratory Test Plan

A. Testing Objectives

1. Demonstrate concept
2. Verify preliminary design
3. Provide basis for prototype design.

B. Prerequisites

1. The pump has been operated manually to ensure freedom from binding
2. Intake and exhaust valve microswitches have been adjusted
3. Intake and exhaust solenoid valves have been checked for proper operation
4. The Freon portion of the test loop has been satisfactorily "hydroed" to 100 psig
5. The water portion of the test loop has been satisfactorily "hydroed" to 60 psig
6. The oscilloscope is within current calibration
7. The oscilloscope has been calibrated with necessary pressure and timing inputs
8. Freon R-11 available (90 lbs)
9. Vacuum pump available (0 to 30 in. Hg) for evacuating Freon system prior to fill
10. Boiler has been checked out per O/I-21.

C. Testing Procedure

1. Evacuation and Fill of Freon System
 - a. Connect a vacuum pump to valve V-7A
 - b. Open vent and fill valve V-7A and start the vacuum pump
 - c. Operate the pump until a vacuum of 28 in. Hg is obtained, or until a vacuum of at least 22 in. Hg is obtained and no significant increase in vacuum occurs within a 30 min period

- d. Secure the pump and shut valve V-7A
- e. Connect a Freon bottle to the system upstream of the condensate pump at valve V-6A. Open valve V-7A and fill the system with Freon R-11 until the boiler level switch is deactivated (approximately 65 lbs of R-11).

2. Water Relief Valve Setpoint

- a. Fill the Buffer vessel approximately half full of water, using a convenient water fill connection
- b. Connect a pressurized air supply to the Buffer vessel
- c. Pressurize the Buffer vessel to 10 $\pm$ 1 psig
- d. Adjust the setpoint adjustment screw on relief valve RV-1 until the valve opens.
NOTE: If valve doesn't open (or stays open), depressurize, drain if necessary, and replace the spring. Repeat steps a through d.
- e. Depressurize and repressurize the Buffer vessel a number of times to ensure the relief valve is set properly
- f. Depressurize the Buffer vessel if the water pump is not to be started up immediately.

3. Initial Pump Start-Up

- a. Ensure Buffer vessel relief valve RV-1 is set to the required position (normally 21 psig). If unknown set RV-1 IAW paragraph 2
- b. Ensure solar pump is fully primed with water
- c. Energize intake and exhaust solenoids by placing solenoid switches in "Warm Up" position
- d. Energize boiler and bring Freon system to N.O.P.T. (65 psig, 180°F)
- e. Place intake and exhaust solenoid switches in the "Start Up" position. NOTE: The reciprocating mass should rise to maximum vertical position

f. Place intake solenoid switch in the "Run" position (The pump should start and run)

- If pump slowly stops operating, reduce discharge pressure by turning Buffer vessel relief valve, RV-1, set point adjusting screw in a counter-clockwise direction. Continue reducing relief valve set point as necessary to achieve continuous operation.
- If pump continues to stop, increase intake valve cutoff by positioning the cutoff microswitch further from the solar pump, in $\frac{1}{2}$ in. increments, and repeat start-up.
- If pump becomes unstable (i.e., it's speed increases) increase discharge pressure by turning Buffer vessel relief valve, RV-1, setpoint adjusting screw in a clockwise direction.
- If pump is still unstable, place intake and exhaust solenoid switches in "Warm Up" position. Decrease intake valve cutoff in $\frac{1}{2}$ in. increments, and repeat start-up.

h. Continue with Steady-State Operation or Shut-Down procedure, as applicable.

4. Steady-State Operation

- a. Steady-state operation is achieved when the pump operates continuously without adjustment for a minimum of ten (10) min, with no significant change in operating characteristics (i.e., temperature, pressure, or speed)
- b. Record all information required by Table 19.

5. Off-Design Conditions

- a. Increase vaporizer temperature and pressure to:

$$(1) \quad T_1 = 190^{\circ}\text{F} \quad P_1 \approx 75 \text{ psig}$$

$$(2) \quad T_1 = 200^{\circ}\text{F} \quad P_1 \approx 90 \text{ psig}$$

Table 19. Data Table

Test No.: _____ Date: _____

Vaporizer:

Voltage: _____ volts
 Current: _____ amps
 Pressure (P_1): _____ psig + 14.7 = _____ psia
 Temperature (T_1): _____ °F + .460 = _____ °R
 Cycle Time: _____ seconds

Solar Pump:

Expander:

Pressure: Intake (P_4) _____ psig
 Exhasut (P_5) _____ psig
 Temperature: Intake (T_4) _____ psig
 Exhaust (T_5) _____ psig

Microswitch Setpoint: Intake Open (M_1) _____
 Intake Shut (M_2) _____
 Exhaust Open (M_3) _____
 Exhaust Shut (M_4) _____

Reciprocating Mass Weight: _____ lbs
 Stroke: _____ in.
 Speed: _____ cpm

Water Pump:

Discharge Head _____ psig x 2.307 = _____ ft
 Suction Lift _____ ft
 Flow Rate _____ gpm

Condenser:

Freon Pressure (P_7) _____ psig
 Water Inlet Temp (T_7) _____ °F
 Water Outlet Temp (T_8) _____ °F
 Flow Rate _____ gpm

Condensate Pump:

Inlet Temp (T_6): _____ °F
 Speed: _____ rpm
 Cycle on Time: _____ sec
 Time Between Cycles: _____ sec

Expander Power:

Oscilloscope P-V Diagram

$$(3) \quad T_1 = 210^{\circ}\text{F} \quad P_1 = 105 \text{ psig}$$

Record required data in Table 19.

b. Increase Buffer vessel discharge pressure to:

$$(1) \quad 15 \text{ psig @ Vapor } T_1 = 180^{\circ}\text{F} \\ P_1 = 65 \text{ psig}$$

$$(2) \quad 20 \text{ psig @ Vapor } T_1 = 180^{\circ}\text{F} \\ P_1 = 65 \text{ psig}$$

$$(3) \quad 25 \text{ psig @ Vapor } T_1 = 180^{\circ}\text{F} \\ P_1 = 65 \text{ psig}$$

Record required data in Table 19.

c. Determine stall conditions at normal operation

(1) Reduce boiler P_1 and T_1 until pump stalls

6. Shut-down

a. Position intake and exhaust solenoid switch to "Warm Up" position

b. Shut V-1, and secure power to boiler

c. Allow system to cooldown, then secure cooling water to condenser by shutting V-4

d. Secure power to condensate pump.

5.4.3 Test Results

Laboratory pump testing was performed over a three week period. During this time, major component and system checks were performed; the pump was operated using compressed air and Freon-11; minor test system changes were incorporated; a minor pump component change was made; and the necessary laboratory pump testing was performed. A discussion of these major items follows.

The hermetic shaft seal was tested when the Freon test loop was pressure tested and subsequently evacuated and filled. Initially, the entire Freon system, condenser included, was subjected to 100 psig compressed air for approximately two hours during which time system integrity was evaluated and leakage

points repaired. During this time, there was no perceptible change in oil reservoir level. Similarly, 28 in. Hg vacuum was maintained on the Freon system for 30 min prior to filling the system with Freon. Again there was no noticeable change in oil reservoir level.

Piston seal friction was much higher than anticipated, primarily attributable to the water piston seal. Referring to Figure 27, it was discovered that the water piston rider ring formed an interference fit with the water cylinder. Approximately 0.010 in. was taken off the diameter of the ring. As a result, the friction forces were decreased, but piston seal friction remained high. Approximately 20 psig of pressure was required to lift the piston and mass to the top during initial startup. Allowing for the weight of the mass, shaft, pistons, etc., starting friction was almost 50 lbs.

Reevaluation of the water piston seal clearly showed that it was overdesigned for its intended purpose. In fact, the suction seal was not used because the power seal was more than adequate. The U-cup seals used are general purpose hydraulic seals noted more for their sealing capabilities than their sliding friction. A schematic of the U-cup seal used is presented in Figure 28 along with a proposed correction to the high friction problem. This correction and conventional leather cup seals will be tested at a later date.

The Freon test loop was evacuated, filled with Freon-11, and placed in stable operation with only minor adjustments to the microswitch setpoints. The pump was operated for approximately one hour before being secured. It was discovered during this initial checkout that the determination of vapor mass flow would be extremely difficult because of the rapid cycle rates of both the boiler and the condensate pump.

During subsequent startups, the pump behaved erratically and would not operate in a stable manner for more than 10 to 15 min. It was discovered that condensate was forming at the expander exhaust valves and restricting vapor flow. To correct this condition, both intake and exhaust valves were opened simultaneously and vapor was bled through the system for a few minutes. The valves were returned to normal operation and the pump would operate satisfactorily for another 15 minutes.

Two possible causes of the problem were theorized. Since the expansion process was occurring along the Freon-11 saturation curve, condensate could be forming in the cylinder and blocking the undersized exhaust valves. Alternatively, exhaust vapor could be condensing in the exhaust header downstream of the exhaust valves and backing up to the exhaust valves.

A. PRESENT WATER PISTON U-CUP SEAL

B. PROPOSED MODIFICATION TO U-CUP SEAL

FIGURE 28. Water Piston U-Cup Seal

Two half inch ($1/2$) exhaust valves were purchased and installed in the system to replace the single quarter inch ($1/4$) exhaust valves that were presently installed. Little improvement was noticed. The larger exhaust valves have a much longer response time (75 to 100 msec) than the smaller valves (5 to 10 msec) so that any improvement due to the larger valves was offset by the slower response time.

Previously, the condenser had been higher than the exhaust valves with a vapor line teeing off the exhaust header up into the condenser, and a return condensate line back down to the header. It was at this time that the condenser was lowered to the position shown in the test assembly figures, below the exhaust valves. Reinstalling the smaller exhaust valves in addition to lowering the condenser alleviated the instability problem.

Relocation of the condenser also allowed the condensate pump to be controlled by the level switch in the boiler, rather than the level switch off the condenser. This meant that the condensate pump would operate less frequently and for a longer period of time; consequently, it could be accurately monitored. Similarly, the boiler could be monitored. Vapor mass flow could not be more accurately determined.

The laboratory pump did not perform as well as was predicted by the dynamic model. A summary of the laboratory pump test results is presented in Table 18 and the data reduction protocol is presented in Table 20. The pump was designed to produce $1/2$ hp at a supply pressure of 80 psia and discharge head of 50 ft. The maximum power (and efficiency) obtained during testing was 0.183 hp (Run #RSEV-2) at 95 psia and 60 ft. The reduced performance can be traced to inefficiencies in the Water Test Loop, which are discussed in Section 5.4.4.

Trends in pump performance followed the dynamic model very closely. The effects of "off-design" conditions were addressed in Test Runs 13 through 19. Discharge head was varied from 33 to 51 ft, and supply pressure was varied from 75 to 125 psia. Figures 29 and 30 compare these changes with the changes predicted by the dynamic model. Absolute values are quite different because of the difference between actual versus predicted operating characteristics, but the magnitude and direction of changes are similar.

A comparison of the pump design specifications and actual pump performance (Run #RSEV-2) is presented in Table 21. The overall pump efficiency of 2.9 percent is significantly less than the design efficiency of 11.6 percent.

Table 20. Data Reduction Protocol

Mass Flow Rate (M):

Condensate Pump (0.45 in³/Rev):

$$M = \frac{1.45 \times \text{Speed} \times \text{Pump on Time}}{\text{Selected Time Interval}} = \text{_____} \text{ lbm/hr}$$

Condenser:

$$W_{\text{Cond}} = 0.19659 \times \text{Water Flow} \times (T_8 - T_7) = \text{_____} \text{ hp}$$

$$M = \frac{W_{\text{Cond}} \times 2546}{(h_{P_5 T_5} - h_{T_6})} = \text{_____} \text{ lbm/hr}$$

Vaporizer Work (W_{Vap}):

$$W_{\text{Vap}} = \frac{M(h_{P_1 T_1} - h_{T_6})}{2546} = \text{_____} \text{ hp}$$

$$W_{\text{Vap}} = \frac{3 \times \text{Volts} \times \text{Amps} \times \text{On Time}}{746 \times \text{Selected Time Interval}} = \text{_____} \text{ hp}$$

Water Pump Work (W_{pump}):

Hydraulic Work Out:

$$W_{\text{Pump}} = \frac{\text{Water Pump Flow (gpm)} \times \text{Total Head (ft)}}{3960} = \text{_____} \text{ hp}$$

Swept Cylinder Work (W_{SC}):

$$W_{\text{SC}} = \frac{0.0418 \times \text{Stroke (in)} \times \text{Speed (cpm)} \times \text{Head (ft)}}{3960} = \text{_____} \text{ hp}$$

Solar Pump Efficiency (η):

Overall:

$$\eta_o = \frac{W_{\text{Pump}}}{W_{\text{Vap}}}$$

Mechanical:

$$\eta_{\text{SC}} = \frac{W_{\text{SC}}}{W_{\text{Vap}}}$$

FIGURE 29. Percentage Change in Thermodynamic Performance Due to Variation of Discharge Head

FIGURE 30. Percentage Change in Thermodynamic Performance Due to Variation in Supply Pressure

Table 21. Laboratory Pump Design Specifications
and Actual Pump Performance

Single Cylinder, single acting, free piston
Poppet valves, solenoid actuated
Carbon impregnated vespel expander piston ring
High strength polymer water piston cup seal

<u>Thermodynamic Parameter</u>	<u>Design</u>	<u>Actual (Run # RESV-2)</u>
Bore (in.)	3.5	3.5
Stroke (in.)	5.2	5.0
Displacement (in. ³)	50	48
Speed (cpm)	158	88
Inlet Temperature (°F)	180	190
Inlet Pressure (psia)	80	95
Cutoff (%)	38	50
Flow Rate (lbm/hr)	115.3	167.9
Water Discharge Head (ft)	50	60
Water Flow Rate (gpm)	33.6	12
Power Output (hp)	0.47	0.183
Efficiency:		
Rankine (%)	83	16.7
Carnot (%)	68	13.8
Overall (%)	11.6	2.9

5.4.4 Test Equipment Inefficiencies

The major sources of thermodynamic irreversibility causing the reduced system performance are the test equipment inefficiencies. These inefficiencies and the resultant effects are discussed below:

- During operation, the flexible hose that connects the pump to the water test loop was expanding and contracting, resulting in significant restriction of the water flow rate. As a consequence, the return stroke which influences the pump speed was reduced. The actual water flow rate was approximately 35 percent less than the available water flow rate.
- Vapor exhaust valves were restricting exhaust flow which causes higher cylinder pressures for the return mass to work against. As a consequence, the return stroke which influences the pump speed was reduced.
- Restriction in the discharge piping between the pump head and discharge head control vessel were causing higher cylinder head pressure than desired, which reduced pump speed and inhibited proper vapor expansion.

The effects of the inefficiencies have been analytically investigated and verified by means of the dynamic computer model simulation. The analytical results are in near agreement with the experimental results. Physical parameters such as vapor valve area, pump valve area, and cylinder pressure can be individually varied; however, the actual combination is unknown and can be readily determined. Unfortunately, the computer model does not predict cyclic heat transfer losses. Heat transfer losses cause higher vapor flow rate and a less efficient system for the same pumping performance.

A pressure-volume (P-V) diagram of the expander cylinder, which would have aided in evaluating pump performance, could not be obtained with the equipment available. The major obstacle was lack of a reference point, that could be related to cylinder volume, to trigger the oscilloscope. A linear transducer will be used in the near future to provide this input.

A P-V diagram that best illustrates the behavior of the laboratory pump from the data available, is presented in Figure 31. Water pump outlet valve area was reduced to restrict water flow and increase water cylinder head pressure to simulate discharge line characteristics. Water pump inlet valve area and expander outlet valve area were reduced to simulate suction line characteristics. The design P-V diagram is also included in Figure 31.

FIGURE 30. Dynamic Model Estimation of Laboratory Pump Performance Based on Actual Test Results

A summary of the laboratory pump losses is presented in Table 22. The measured inlet valve pressure difference loss is equal to the predicted pressure difference loss; however, the measured exhaust valve pressure difference loss, water pipe dynamics loss and water discharge line losses are significantly larger than the predicted values. Unfortunately, it was not possible to measure the heat transfer losses.

A summary of the possible laboratory pump modifications for improving pump efficiency is presented in Table 23. The three major improvements are the following:

- Hard pipe flexible hose connections - this will increase water flow rate approximately 35 percent
- Relocate discharge head control vessel as close as practical to the pump head - this will reduce the pressure drop between the pump head and control vessel, and increase both pump speed and vapor expansion
- Increase vapor exhaust valve area - this will increase pump speed and increase the pump efficiency 10 to 20 percent.

The incorporation of these changes will increase overall pump efficiency from the 2 to 3 percent, to 6 to 8 percent, an increase in overall efficiency of approximately 300 percent.

5.4.5 Conclusions

1. Results of the Laboratory Test Program were very positive. The concept of a thermally driven direct acting Organic Rankine cycle pump was successfully demonstrated; it exhibited reasonable performance and stable operation.
2. Design performance was low but off-design performance was as good, if not better than expected. The laboratory pump accommodated greater than 25 percent changes in supply pressure and discharge pumping head with no changes in valve timing required.
3. Evaluation of laboratory pump off-design conditions substantiated the validity and usefulness of the dynamic model as both a design tool and diagnostic tool. Further testing is required to quantify actual design predictions.

Table 22. Summary of Laboratory Pump Losses

	<u>Actual</u>	<u>Predicted</u>
Intake Valve Pressure Difference	26%	25%
Heat Transfer to Cylinder Wall	-	20 to 40%
Piston Ring Friction	-	<5%
Water Piston Seal Friction	-	5 to 10%
Exhaust Valve Pressure Difference	35%	10 to 15%
Water Pipe Dynamics	35%	
Water Pump Intake and Exhaust Valves	-	<5%
Water Discharge Line (head to vessel)	30%	

Table 23. Summary of Laboratory Pump Modifications

Hard Pipe Water System	~ 35% Water Flow Rate Increase
Increase Vapor Valve Areas	~ 10 to 20% Efficiency Increase
Increase Water Discharge Pipe Size	~ 25 to 30% Speed Increase
Between Pump Head and Buffer Vessel	~ 25 to 30% Expansion Increase
Insulate Gas Piston	~ 20% Vapor Flow Decrease
Resultant Overall Efficiency	8 to 10%

4. Material performance in the Freon system was excellent. Neither the expander piston seal nor the hermetic shaft seal suffered detrimental effects from the Freon environment.
5. The water piston seal was over designed; consequently, pump performance suffered. The water piston seal is not a critical component in regard to material selection; however, due to the presence of frictional forces, it does affect the thermal efficiency.
6. Test equipment design flaws reduced pump performance and hindered evaluation of pump design parameters. A thorough and complete examination of test equipment requirements and design are required.
7. Modifications to the test equipment and further testing are required to realize the full potential of the laboratory pump.

6. PROTOTYPE PUMP CONCEPTUAL DESIGN

6.1 Introduction

From the results of the marketing analysis described in Section 2, it was shown that there are two broad categories of agricultural irrigation, sprinkler and gravity. The solar pump is more suitable for gravity irrigation which requires pumping capacity less than 20 hp. A prototype conceptual design is proposed for both 1 hp and 20 hp pump size. The economics of the prototype solar irrigation system is presented and a preliminary Phase II program plan is discussed.

6.2 Objective

The objective of the prototype pump conceptual design is to present a viable design which will be compatible with the potential solar agricultural irrigation market and suitable for commercialization.

6.3 Summary

The conceptual prototype pump design is very similar to the laboratory pump except the valves are integrated into the pump. The projected pump isentropic expander efficiency is 64 percent and the estimated overall thermal efficiency is 11.2 percent.

The system cost for the 1 hp prototype pump is \$5,460; however, for a commercial product it is estimated that the system cost would be reduced to \$3,180. Similarly for the 20 hp prototype pump, the system cost is \$91,400; however, for a commercial product it is estimated that the system cost would be reduced to \$56,700.

A proposed Phase II program plan includes five tasks designed for the successful design, fabrication, and testing of a 1 kW prototype pump system.

6.4 Conceptual Design

The direct acting Rankine cycle prototype pump would employ integrated valves. A schematic illustrating the proposed pump is shown in Figure 32. The improvements incorporated in the prototype pump include the following:

- The prototype pump uses mechanically actuated valves in the expander rather than solenoid valves. The mechanical valves are integrated in the pump design and are expected to increase the pump thermodynamic performance and reliability characteristics.

Figure 32. Direct-Acting Rankine Cycle Prototype Pump

- The prototype pump uses integral check valves in the water pump rather than the in-line check valves. Again the proposed valves are expected to increase the pump thermodynamic performance and reliability characteristics.

Aside from the valve design changes, the prototype pump will operate in a manner similar to the laboratory pump as described in Section 4.

The proposed process for the expander is shown in Figure 33. The Freon, in a saturated liquid state, (1) is pumped to the subcooled state (2). Solar energy increases the temperature of the Freon and vaporizes it to the saturated vapor state (3). The Freon is then expanded to the saturated vapor state (4) where the latent heat is removed by means of a water cooled condenser. The expansion work developed by the Freon is used to pump water.

The projected thermodynamic losses developed by the expander are summarized in Table 24. The total losses are estimated to be 36 percent of the reversible adiabatic work; as a result, the projected expander isentropic efficiency is 64 percent. The overall thermal efficiency is estimated to be 11.2 percent. The required collector area is 270 ft²/kW. The value was determined from the following assumptions:

FIGURE 33. T-S Diagram of the Freon Cycle

Table 24. Summary of Thermodynamic Losses Occurring in the Freon Expander/Pump

<u>Loss Source</u>	<u>Percentage of Isentropic Work</u>
Expander Valve Pressure Difference Losses	2
Heat Transfer to the Cylinder Wall	8
Heat Generation Due to Piston Ring Friction	4
Irreversible Mixing Loss	8
Valve Mechanical Actuation Loss	3
Freon Pump Valve Pressure Difference Losses	8
Miscellaneous	<u>3</u>
Total Losses	36 Percent

- The average insolation from 9 AM to 3 PM during the growing season in the Southwest and Great Plains regions of the United States is approximately 280 btu/hr-ft²
- The average collector efficiency is 40 percent
- Based on an overall thermal efficiency of 11.2 percent, the required solar energy input is 8.9 kW/expander kW.

Table 25 is a summary of system specifications for the 1 hp pump and a 20 hp pump. Comparing these two prototype systems, one will notice that even though the expander work is increased by a factor of 20, the required expander displacement and weight are increased by factors of 33.3 and 4, respectively.

6.5 Economics

A summary of the estimated costs of both the 1 hp and 20 hp pumps is presented in Table 26. For each pump size the costs for a prototype demonstration system and a commercial system are included. As to be expected, the pump value per horsepower decreases with increased pumping capacity.

Table 25. Summary of Proposed System Specifications for both 1 hp and 20 hp Pumps

<u>Component</u>	<u>1 hp Pump</u>	<u>20 hp Pump</u>
Condenser	<p>Surface Area = 8 ft² Tube Shell Hx 36 in. long 5 in. diameter</p> <p>Speed = 300 rpm Expander Displacement = 60 in.³ Bore = 4 in. Stroke = 5 in. Overall Dimensions = 6 in. diameter by 18 in. long Weight = 150 lb</p>	<p>Surface Area = 160 ft² Tube and Shell Hx 108 in. long 12 in. diameter</p> <p>Speed = 300 rpm Expander Displacement = 2000 in.³ Bore = 14 in. Stroke = 13 in. Overall Dimensions = 20 in. diameter by 60 in. long Weight = 600 lb</p>
Condensate Pump	3/4 gal/min gear pump	15 gal/min gear pump

Table 26. Cost Summary for Direct-Acting Rankine Cycle Pump

Case	Solar Collectors	Condenser ²	Pump- ³ Expander Packages	Condensate ⁴ Pump	Pipe ⁵ Tubing Fittings	Total System Cost	System Cost/kW	Pumping Cost, \$/Delivered kWh
1 kW Prototype Demonstration	\$ 3,720	\$ 150	\$ 1,100	\$ 240	\$ 250	\$ 5,460	\$5,460	\$0.38
1 kW Produced Commercially	2,480	150	220	80	250	3,180	3,180	0.21
20 kW Prototype Demonstration	74,400	2,500	12,000	500	2,000	91,400	4,570	0.31
20 kW Produced Commercially	49,600	2,500	2,400	200	2,000	56,700	2,835	0.19
¹ Allowance for installation cost included in individual component costs. Costs of well drilling and water distribution not included. ² This is a commercially available piece of hardware. Cost estimate based on purchases of similar equipment. No "learning curve" cost reduction would be expected. ³ Pump-expander package cost estimate based on \$20/lb prototype fabrication cost and \$4/lb production hardware cost for reciprocating machinery. ⁴ Condensate pump available off-the-shelf for the demonstration in the indicated price range. However, a condensate pump tailored to this application could be manufactured for the indicated pricing for the commercial hardware. ⁵ Estimated								

The kilowatt-hour pumping costs are estimated on the basis of 9 percent interest, 4 percent depreciation, and 1 percent maintenance and operating costs for a useful operating time of 2,000 hr/yr.

The total system cost for the 1 hp prototype pump is \$5,460 and for the 20 hp prototype pump is \$91,400.

6.6 Proposed Phase II Development Program

The advantage of the proposed Phase II development program is that the prototype pump design will incorporate the experience and knowledge obtained in the Phase I effort. The proposed Phase II development program will consist of the following tasks:

- Mechanical design study
- Materials study
- System dynamic study and system performance optimization
- Prototype detailed design, fabrication and testing
- Market assessment.

The tasks are discussed briefly in the following sections.

6.6.1 Task 1 - Mechanical Design Study

The proposed 1 hp pump design will be further evaluated. Design areas to be investigated will include the following:

- Optimal expander valve configuration and actuation
- Pump valve configuration and actuation
- Integration of the condensate pump into the overall pump design
- Reduction of the expander clearance volume to minimize mixing loss
- Optimal selection of the cylinder wall dimensions, load carrying members, small peripheral components
- Incorporation of favorable maintenance and reliability characteristics.

The design study will emphasize the effect of mechanical design alternatives on the thermodynamic efficiency and reliability of the pump system. The viable design options are not expected to influence substantially the manufacturing costs.

6.6.2 Task 2 - Materials Study

The materials of construction can have a significant effect on the system cost, thermodynamic performance, and reliability of the pump system. The materials selected for each component will be carefully investigated with respect to function and to interaction of the component with remaining parts of the pump.

Material life and reliability will be given preference wherever thermodynamic performance is not sacrificed. For example, Teflon is a prime candidate for water piston seals because of its extremely low coefficient of friction; however, Teflon may not be able to withstand the extreme environmental conditions that it may be subjected to in remote water pumping applications. Leather cup seals may be a preferable material for this application because of its demonstrated reliability and performance.

6.6.3 Task 3 - System Dynamic Study and System Optimization

Although the pump system is conceptually simple, the dynamics are complex. The system can be designed to use the dynamics to its advantage; whereas, for a poorly designed system, the dynamic effects can result in mechanical dissipation, poor valve timing, excessive wear, and high cyclical loads which cause premature failure of components. The major system dynamic areas to be analyzed include the following:

- Spring-mass systems of the intake and exhaust valves
- Transfer of work from the expanding Freon to the water via the piston mass
- Transfer of heat into the boiler
- Identification and quantification of the major sources of irreversibility and their effect on the dynamic behavior of the system
- Potential vibrational modes
- Effect of design conditions on the system operation.

The computer model developed during Phase I will be used as the design load to study the dynamic behavior of the system and to determine the optimum design configuration.

6.6.4 Task 4 - Prototype Detailed Design, Fabrication and Testing

The purpose of this task is to evaluate the design improvement, system thermodynamic performance, and reliability.

Detailed manufacturing drawings of a 1 hp prototype pump will be prepared. The development testing will be accomplished in two steps. For the first step the proposed mechanical valve actuation configurations and integrated condensate pump design will be evaluated on the existing 10 kW test facility. Depending on the outcome of these test results, design refinements may be incorporated. For the second step, additional testing will be performed using the existing FMA Freon collector/boiler system. In this manner the prototype pump performance will be demonstrated for actual field conditions. Daily performance data will be obtained as a function of the insolation rate. Fluctuations in insolation rates will be analyzed to determine the requirements for off-design conditions and thermal storage.

In addition, reliability data will be accumulated during this task.

6.6.5 Task 5 - Market Assessment

For this task the thermodynamic performance and economic projections obtained in Phase I will be updated. Conclusions and recommendations regarding the potential for commercial development of the direct acting organic Rankine pump will be prepared.

7. CONCLUSIONS AND RECOMMENDATIONS

The first phase of a development program to demonstrate the feasibility of the "Direct Acting Organic Rankine Pump" concept for use as a low temperature thermally driven irrigation pump has been successfully completed. The program consisted of four major tasks:

- Establish design specifications for the proposed pump concept
- Design and analysis of an experimental breadboard pump
- Perform a laboratory test program to verify analysis
- Design and analysis of a prototype pump.

The conclusions resulting from the program include the following:

- There exists two broad categories of agricultural irrigation, sprinkler and gravity. The solar pump was designed to satisfy the requirements of gravity irrigation (20 hp, 50 ft of head, 1000 gpm).
- The projected economic market for the solar pump is 7 percent of the nation's total agricultural irrigation pump market. However, the world market potential is 3 to 4 times that of the United States.
- A computer model characterizing the thermodynamics and mechanical behavior of the Rankine cycle expander/pump system was developed.
- A laboratory pump was designed for an output power of 0.47 hp at 50 to 70 ft discharge head, and a flow rate of 30 to 40 gpm. The actual power output was 0.18 hp at 60 ft discharge head and a flow rate of 12 gpm.
- The overall thermodynamic performance is 2.8 percent which is significantly less than the predicted efficiency of 11.6 percent.
- It is projected that with the incorporation of laboratory pump modifications, the overall pump efficiency will increase from 2 to 3 percent to 6 to 8 percent.
- A conceptual prototype design similar to the laboratory pump except that the valves are integrated into the pump was prepared. The estimated overall efficiency is 11.2 percent and the system cost of the 1 hp prototype pump is \$5,460.

- A proposed Phase II program plan which includes five tasks designed for the successful design, fabrication, and testing of the 1 hp prototype pump system was prepared.

The recommendations for future development work include the following:

- The existing laboratory pump and test equipment be modified in order to improve the system overall thermodynamic performance
- A 1 hp prototype pump system be designed, fabricated, and tested. The prototype system will be specifically designed for the agricultural irrigation market
- A commercialization plan for the solar driven agricultural pump be prepared.

OPERATING INSTRUCTIONS 0/+ - 21

Thermax Directly Heated Vaporizer For Compressed/Liquefied Gas

Pressure/Temperature Actuated Side Arm Style

Description

This style vaporizer is suitable for service on carbon dioxide and related compressed/liquefied gases. The complete vaporizer consists of:

1. Heater and vaporizer assembly
2. Control panel containing electrical controls and pressure switch.

Flow through the vaporizer is of the "side-arm" or pressure maintenance arrangement. The vaporizer is normally flooded with product and vapor is returned to the top of the storage tank.

The heater is immersed in the vaporizer shell assembly. The thermostat in the heater maintains the heater below 230°F (adjustable), whenever tank pressure is below the pressure switch set point. Pressure is sensed from a tap in the vaporizer shell.

3. Refer to Thermax drawing for temperature, operating pressure, and pressure switch ratings.

Operation

When the liquefied gas storage tank pressure falls below the pressure switch setting, the switch closes the heater circuit. The heater warms, which in turn vaporizes the liquified gas as it passes through the shell. The gas bubbles back into the customer storage tank. If the liquid supply is low or interrupted, the heater will heat to about 230°F in 1/2 to 2 min, which is normal, and then cycle ON-OFF via the adjustable heater thermostat. This style vaporizer delivers saturated vapor at the temperature corresponding to the tank pressure.

CAUTION: Carbon steels are normally ASME limited to -20°F operation. Do not operate liquid CO₂ systems below 215 psia without proper consideration of equipment.

Installation

1. Unit should be installed to conform to local codes.
2. Inspect the unit for obvious shipping damage.
3. Check nameplate data with site process conditions: fluid, pressure, and pressure switch settings.
4. Secure unit to foundation.
5. Check service power for voltage. Units are shipped either 230 or 460 volts. Some models may be field converted as required. To convert, it is necessary to reconnect control transformer and heater terminals and check fusing.
- 6.* Connect power to the contractor inside the cabinet and ground unit as required.
7. Close door and turn unit "ON". Pilot should light, since pressure switch is open to air. Switch should close and HEATER "ON" light should come on, showing heater operation. Turn unit "OFF".
8. Connect process liquid and vapor lines and install drain valve. Check for leaks.
9. Check tank relief valve capacity and increase as required. If vaporizer block valves are installed, add relief valve to vaporizer shell side of valves. A plugged port is provided for this purpose.
10. Note tank pressure and turn unit "ON". If tank pressure is above setting**, power "ON" light will (only) light. If tank pressure is below setting, both pilot lights will come "ON". The vaporizer should then operate until tank reaches set pressure, at which time the heat "ON" light should go out.
11. As tank cools or customer draws product, heat will come back "ON", if pressure falls below setting.

* Or to disconnect device if furnished.

** Units with disconnect device do not have "POWER ON" pilot.

Routine Inspection/Service

1. After initial installation, unit should be checked during several pressure cycles to make sure thermostat and controls are operating properly. Observe tank pressure during cycles as evicence of proper settings.
2. After start-up, check system periodically, as you would any control system.
3. Blow-down product side of vaporizer periodically to remove impurity buildup resulting from continuous boiloff of higher boiling point fluids (water/oils)

Trouble-Shooting

1. If unit does not heat, check:
 - a. Fuse on control circuit and power to unit.
 - b. Tank pressure... unit will not heat unless tank pressure is below normal set point.
 - c. Thermostat... unit will not heat unless thermostat is set above gas temperature in vaporizer. Units have thermostats installed under the heater cap. Do not set above 300°F, without review of materials.
 - d. If above controls are operating properly, and tank is below settings, (both) lights should light and contactor should close, sending power to the heater. If heating still does not occur, check heater element for burn-out, and replace as required. To replace heater, turn power off at disconnect; isolate and blow-down vaporizer; unhood conduit and remove heater from well. Put in new element and set thermostat at 230°F.
2. If unit heats, but tank pressure remains low:
 - a. Check tank storage level and consumption rate.
 - b. Check pressure switch setting. With no gas consumption, tank should gradually increase to the set point.
 - c. Check pressure switch for proper actuation. Check for frozen pressure tap line.

- d. Check thermostat setting.
- e. Check power draw. All three heater phases should draw listed amperes. A variance indicates element failure or damage, or possible power problem.

NOTE: 10 percent lower voltage means 20 percent less heat.

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