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Report No. TE4254-10-80

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MASTER

FINAL REPORT

**DIRECTLY COUPLED SOLAR THERMAL WATER
PUMPING CONCEPTS FOR AGRICULTURE**

October 1978 - May 1979

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**CONTRACT ET-78-C-03-1571
August 19, 1978 - May 31, 1979**

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1. INTRODUCTION

1.1 CONTRACT OBJECTIVE

Thermo Electron Corporation was to analyze, design, test, and evaluate concepts that utilize solar thermal energy to pump water without intermediate production of mechanical or electric power. Low-cost and reliable means for utilizing the expansion of a working fluid with thermal energy were to be explored. Working laboratory models were to be constructed and tested to check the validity of the analyses. Designs were to be generated and cost comparisons made.

1.2 SUMMARY OF PROGRAM ACTIVITIES AND ACCOMPLISHMENTS

Several alternative direct pumping cycles and systems were analyzed for their potential as directly coupled solar water pumps. The most promising of these was predicted to be capable of delivering water at a sunlight-to-pumped-water conversion efficiency of approximately 9½ per cent. This is achievable over a variety of pumping conditions with an insolation of 1 kW/m². The system power cost, based upon average solar insolation and an estimated selling price, is \$4 per delivered hydraulic watt for a 1 kW average power system. Present value analyses of alternative water pumping methods, in remote locations, indicate that the directly coupled solar thermal water pump is the lowest-cost pumping method after four years of operation. Details of the financial analysis can be found in Section 10.3.

Six different concepts were investigated, to varying degrees, depending mainly upon their viability. The general analysis of Section 2.1, performed for a single-cycle-per-day system, indicated that improved economics could be obtained with a multiple-cycle-per-day system. Heat loss calculations, in Section 2.2, for a nonspecific, integral collector pump, using a constant pressure cycle, directed the design effort to separating the collector and the pump as described in Section 4.

Results of the preliminary analysis on specific constant-pressure cycles utilizing air, water, and organic working fluids are presented in Sections 3.1, 3.2, and 3.3, respectively. The attractiveness of organic compounds dictated their use as the working fluid. Thermodynamics of the working fluids are provided in Section 7.1. The substantial efficiency improvement, achieved by utilizing the expansion work of a gas, is delineated in Section 5.1. Four of the expansion cycle concepts considered are described in Sections 5.2 - 5.5 and include a "fluid flywheel" cycle, a jet pump, mechanical linkages, and a regenerator.

The most promising water-pumping concept is based upon an expansion thermodynamic cycle, which operates over similar cycle conditions, independent of delivery requirements. It attains high efficiency by operating at higher temperatures and pressures, even in low pumping head applications. The elevated pressure is utilized efficiently by storing excess work during the pumping stroke as kinetic energy in the pumped water column. The stored energy is returned when the working fluid pressure falls below the total pumping head and, as a result, the working fluid is expanded to a pressure below the delivery head. Because of its method of storing energy, this cycle is described as a "fluid flywheel." Section 5.2 provides a detailed analytic description of the cycle. The test results, available in Section 8.2, have shown that the analytic model, used to generate the system simulation results of Section 10.1, has been validated by experiment. When pressurized air was used as a test working fluid, the experimental results agreed with the simulation data within the experimental error. Results with n-Pentane, the organic working fluid, were within 10 percent of simulation values in spite of some "nonideal" factors. The fabrication cost estimates in Section 9 were based upon the designs of Section 6. Many of the costs were derived from commercially available hardware, and several areas for cost reduction are evident. These costs, extrapolated to a selling price, were in turn used in the financial comparison in Section 10.3. That comparison included small pumps powered by gasoline and diesel engines, wind, and photovoltaic

modules. The engine-powered pumps had lower first costs, but their higher operating costs resulted in higher present value costs after four years of operation. The directly coupled system is somewhat limited in size, primarily because of its periodic flow characteristics. Additionally, larger systems can incorporate more complex and efficient hardware without a severe penalty.

The simpler nonexpansion constant-pressure pumping cycle, which would utilize hardware similar to that of the "fluid flywheel" cycle had a predicted efficiency which was, at best, only 30 percent of the preferred cycle. Furthermore, its performance degrades at low pumping heads unless mechanically matched to the load. Detailed performance predictions were made for a jet pump augmented cycle but it failed to show significant advantages over the constant-pressure cycle. Two nonlinear mechanical linkages were considered because of their ability to match the working fluid source impedance to that of the pumping load, thereby achieving an expansion cycle. Their analysis shows that further investigation is warranted, but was not possible because of the scope of this program. Such linkages permit a reduction in the magnitude of the intracycle energy storage. The indirect use of expansion work by regenerative heat recovery, did not demonstrate sufficient efficiency improvement over the constant-pressure cycle to justify detailed investigation.

Models for both the boiling and condensing of n-Pentane were made during the analytic phase of the program. Testing during the experimental phase showed that the overall condensing film coefficient was 37 percent below the estimated value when 1 psi of noncondensable vapor was present. The condensing film coefficient degraded by an additional 50 percent when the pressure of noncondensable vapor increased. Experimental work on a simple system to purge the condenser of noncondensable vapor is required. The correlation for the recommended, natural convection collector/boiler was excellent. It conservatively predicted collector tube temperature within 1°C at quasi-steady-state operating conditions.

The potential for a simple, efficient, and cost-effective solar thermal pumping system was demonstrated during the program. The construction and testing of prototype pumps is recommended as the next phase in demonstrating the viability of the design concepts. A parallel cost reduction effort can improve the already attractive economics for remote applications.

2. SYSTEM ANALYSIS OF THE COMBINED COLLECTOR/PUMP CONSTANT PRESSURE SYSTEM

2.1 SINGLE CYCLE PER DAY

The basic cycle for the combined collector pump shown in Figure 2.1 was analyzed to determine the viability of the single-cycle-per-day concept (see Figure 2.2). The solar input power is given by $P_{in} = I_o A$, where I_o is the insolation and A is the area of the collector/pump. The outpower is given by $P_{out} = \Delta P A L/T$, where ΔP is the difference between delivery and inlet pressures, L is the pump/collector length or depth of water, and T is the cycle time period. The system efficiency is then given by $\eta = P_{out}/P_{in}$ or $\eta = \Delta P L/T I_o$.

If we estimate η at 2.5 percent for the basic nonexpanding system, and I_o at 0.8 kW/m² or 0.381 ft-lbf/sec-in², then the time period is $T = 105 \text{ sec/psi-ft} \cdot \Delta P L$. A low head system might have a pressure requirement of 5 psi, whereas a high head system might have a 50-psi pressure requirement. The ratio of time period to pump length is $T/L = 8.75 \text{ min/ft}$ at 5 psi, and $T/L = 87.5 \text{ min/ft}$ at 50 psi.

For the low head system, the single-cycle-per-day concept does not appear viable since the tank length would be excessive. Even for the high-pressure system, the tank length would have to be at least 5 ft to contain sufficient water for a diurnal cycle.

Thermal losses in the basic system can be partially estimated from the thermal mass that must be heated. Similarly, system cost can be estimated from the mass of material required. If a cylinder with hemispherical ends is assumed, then the axial stress in the cylinder walls is given by $\sigma_z = pr/zt$, where p is the pressure difference, r the cylinder radius, and t the material thickness. The circumferential stress is $\sigma_\theta = pr/t$.

Yielding occurs when the yield stress, Y , is equal to $0.87 pr/t$ according to Mises criterion or $Y = pr/t$ according to the maximum shear stress criterion. The minimum cylinder wall thickness is then $t_c = pr/Y$.

Figure 2.1 Nonexpanding Vapor Cycle Engine Concept

$$\eta = \frac{W}{Q_{in}}$$

$$W = {}_2W_3 - {}_4W_1$$

$$W = (P_2 - P_0)(\nu_3 - \nu_2) - (P_4 - P_0)(\nu_4 - \nu_1)$$

$$Q_{in} = {}_1Q_2 + {}_2Q_3$$

$${}_1Q_2 = E_2 - E_1 + {}_1W_2$$

$${}_1Q_2 = u_2 - u_1 + {}_1W_2$$

$${}_1Q_2 = h_2 - h_1$$

$${}_2Q_3 = E_3 - E_2 + {}_2W_3$$

$${}_2Q_3 = u_3 - u_2 + P_2(\nu_3 - \nu_2)$$

$${}_2Q_3 = h_3 - h_2$$

$$Q_{in} = h_3 - h_1$$

$$\eta = \frac{P_2(\nu_3 - \nu_2) - P_4(\nu_4 - \nu_1)}{h_3 - h_1}$$

Figure 2.2 Nonexpanding Engine Thermodynamics

for maximum shear strength. In a hemispherical head, the minimum head thickness can be similarly shown to be $t_h = pr/2Y$. The mass of the cylinder is then given by $M_c = 2 \pi r^2 \rho L p / Y$, where ρ is the density, and the mass of the head is $M_h = \pi r^2 \rho L p / Y$.

The heated thermal mass per unit area with specific heat, cp , is then $M_T/A = cp(\rho p/Y)(r + 2L)$ with an uninsulated cylinder, or $M_{Ti}/A = (cp \rho pr)/Y$ if the cylinder is insulated from the working fluid. The cost per unit area is $C/A = S (2\rho p L/Y)(1 + r/L)$, where S is the cost of material and processing per unit of mass.

For example, if steel is used with an estimated cost of \$0.80/lbm, and $Y = 20 \times 10^3$ psi, $\rho = 487$ lbm/ft³, and $p = 100$ psi (50 psi $\times 2$ for a safety factor), then the cost per unit area is \$3.90 L/ft³. Even if r/L is small, which would increase the surface area, and therefore heat losses, the cost for a one-cycle-per-day system would be a minimum of \$19.50/ft². The cost per output power can be estimated directly if we note that $L = T\eta I_o / \Delta P$ and $C/A = (S2\rho p Y)(1 + r/L)$; neglecting r/L and assuming $p = 2\Delta P$, then $C/A = (S4\rho T\eta I_o)/Y$. The estimated cost per watt can be projected by

$$\frac{C/A}{\eta I_o} = \frac{S4\rho T}{Y};$$

so

$$\frac{C/A}{\eta I_o} = \frac{\$1.44}{\text{watt}} \frac{T}{\text{hr}}.$$

For a 7-hr day, the cost for a single-cycle-per-day system is then a minimum of \$10/W. These cost figures are for a minimum material usage and do not reflect the true cost of a complete system, but provide a handle for the approximate cost magnitude. As is evident from the above analysis, the cost per output of the system can be reduced by decreasing the length, and therefore the cycle time, of the pump.

2.2 MULTIPLE CYCLES PER DAY

The energy lost through cyclic heating of the collector pump can be estimated by comparing the energy delivered to the energy lost. The

energy delivered is given by $E_d = \Delta P A L$, and the energy lost by $E_l/E_d = (M/A)/\Delta P L$, with $2\Delta P = p$, then $(E_l/E_d)_a = (2C_p \Delta T/Y)(r/L)$ for adiabatic walls, or $(E_l/E_d)_e = (2C_p \rho \Delta T)/Y(2+r/L)$ for walls in thermal equilibrium with the working fluid. Utilizing the figures for steel with $c_p = 0.113 \text{ Btu/lbm-}^\circ\text{F}$, $\Delta T = 100^\circ\text{F}$, and $r/L = 0.5$ gives $(E_l/E_d)_a = 1.49$ and $(E_l/E_d)_e = 7.43$.

In addition to this loss, the loss to the environment and other cyclic losses must be included. An insulated pump cylinder sitting in the open will lose heat at a rate given by

$$q_e = \frac{\Delta T}{\frac{1}{2\pi L r_o h_o} + \frac{\ln r_o/r_i}{2\pi L K} + \frac{1}{2\pi L r_i h_i}}$$

where ΔT is the temperature difference, r_o the outer diameter, r_i the inner diameter, K the thermal conductivity, and h_i and h_o the film coefficients for the inner and outer surfaces, respectively. The ratio of heat loss to heat input is

$$\frac{q_l}{q_{in}} = \frac{q_l}{\eta_c I_o \pi r_i^2} = \frac{\Delta T}{\frac{r_i}{r_o h_o} + r_i \ln \left(\frac{r_o/r_i}{k} \right) + \frac{1}{h_i}} \cdot \frac{2L}{\eta_c I_o r_i}$$

where η_c is the collection efficiency to the upper surface. Neglecting the thermal impedance on the surfaces gives

$$\frac{q_l}{q_{in}} = \frac{2KL\Delta T}{\eta_c I_o r_i^2 \ln \left(\frac{r_o}{r_i} \right)}$$

If the following parameter values are assumed:

$$L/r_i = 2,$$

$$I_o = 254 \text{ Btu/hr-ft}^2,$$

$$\eta_c = 0.5,$$

$$\Delta T = 100^\circ\text{F},$$

$$r_o/r_i = 1.17,$$

$$k = 0.022 \text{ Btu/hr-ft-}^\circ\text{F}$$

then $q_1/q_{in} = 0.45/r_i$. This is not an inconsequential loss and, since the walls are not truly adiabatic and condenser losses have not been included, the losses can readily consume a large fraction of the potential efficiency. These losses can be reduced by placing the pressure vessel on the outside and the insulation on the inside. This tends to increase cost because the mass of the vessel increases as the radius is squared; furthermore, with the insulation on the inside, the effective collection and pumping areas are relatively smaller.

3. WORKING FLUIDS

3.1 AIR

In addition to the several organic working fluids considered, two inorganic fluids, water and air, were also given consideration. Due to the relatively low pressures obtainable with these fluids at flat-plate collector temperatures, the range of practical working heads is quite limited. Within this range, however, they do offer certain attractive features: both are readily and universally available; automatic makeup can be inherent in the pump design; and there are no problems with toxicity or contamination of the pumped water.

The air cycle, which involves no phase change, is the most limited in pumping range. It is possible, however, by interacting with the atmosphere and using it as an infinite source and sink, to increase this range somewhat. The direct pumping air cycle studied is as follows. Air, at ambient pressure and temperature, is introduced into a vessel over a quantity of water. The air is then heated. As the temperature of the gas is increased, the product of its pressure and volume also increases to maintain the perfect gas relationship $T_2/T_1 = P_2V_2/P_1V_1$. If T_2 is large enough, P_2 reaches the discharge head, and V_2 increases displacing water. At the end of the water ejection cycle, the air, now at temperature T_2 and P_2 , is vented to atmosphere. The temperature and pressure of the air in the tank will drop due to this expansion, which, for the air remaining in the tank, is essentially isentropic, until atmospheric pressure is reached at temperature T_2' . The air is then cooled to temperature T_1 , either by letting the tank sit overnight, or by actively cooling with some of the pumped water. As the air is sufficiently cooled, water is again drawn into the tank until the starting volume is reached. Ambient air is then readmitted and the cycle is restarted.

As the system interacts with ambient air, cycle operation is sensitive not only to maximum operating temperature and cooling water temperature, but also to ambient temperature and pressure. This effect is offset slightly by increasing collector efficiency for higher ambient temperatures.

To gain perspective on the practical limits to pumping head, the upper limit is considered to be the head at which the pump stalls; that is, the point at which no water is displaced and the efficiency goes to zero. Assume the following conditions:

atmospheric pressure $P_1 = 10.3$ meters-water column

ambient temperature $T_1 = 30^\circ\text{C} = 303^\circ\text{K}$

final temperature $T_2 = 95^\circ\text{C} = 368^\circ\text{K}$

From the statement of the problem $V_1 = V_2$; therefore,

$$\frac{P_1}{P_2} = \frac{T_1}{T_2}$$

$$P_2 = 10.3 \text{ meters-water column } \frac{368^\circ\text{K}}{303^\circ\text{K}} = 12.5 \text{ meters-water column}$$

$$\Delta P = P_2 - P_1 = 2.2 \text{ meters-water column}$$

The heated air is then expanded isentropically to atmospheric pressure P_1 . The temperature T_2' at the end of this expansion can be calculated using both the relative pressure relationship $(P_a/P_b)_{s=\text{constant}} = P_{ra}/P_{rb}$, and a table that gives P_r as a function of T . Using the Keenan & Kaye "Gas Tables",

$$P_r = 2.846 \text{ at } 368^\circ\text{K}$$

$$\frac{12.5}{10.3} = \frac{2.846}{P_{r2}}$$

$$P_{r2}' = 2.345$$

$$\text{at } P_r = 2.345, T \approx 348^\circ\text{K}$$

If the air is then cooled to 30°C (303°K), again stipulating that V remains unchanged, then

$$P_3 = 10.3 \text{ meters-water column } \frac{303^\circ\text{K}}{348^\circ\text{K}}$$

$$P_3 = 9.0 \text{ meters-water column}$$

$$\Delta P = 10.3 - 9 = 1.3 \text{ meters-water column suction}$$

The total $\Delta P = 1.3 + 2.2 = 3.5$ meters-water column. Therefore, for the system to do any useful work, the suction head must be less than 1.3 m and the pressure head less than 2.2 m. The range of applicability of a pump such as this is quite limited, with perhaps its best application being to pump water from a holding pond into an irrigated field. The primary attraction of the air system is its use of the atmosphere as an infinite source and sink of working fluid.

3.2 WATER

Water is of course the most common working fluid in use for vapor engines. Its primary disadvantage in solar pumping applications is its relatively high boiling temperature at atmospheric pressure. At even modest delivery heads, losses of heat to the surroundings become very large, and overall efficiency very poor, relative to that obtained with an organic working fluid. To its advantage, however, water vapor reaches quite low absolute pressures at fairly high temperatures. At 50°C, for instance, the absolute pressure of the vapor is only 1.27 meters-water column. Thus, although a direct-acting pump using water vapor is quite limited in delivery head, it can quite easily deliver several meters of suction, and could be quite useful in situations where these suction heads are required. It, of course, offers the same advantages as air; that is, nontoxicity and a steady makeup supply.

Using a portion of the pumped water primarily in a suction mode does introduce certain problems. During each cycle, dirt and dissolved salts will be left behind on the heat transfer surfaces, and air or other dissolved gases will be introduced into the vapor space. In the multiple-cycle scheme, in which some of the pumped water is used to cool the vapor, these problems could possibly be overcome.

In a system using water, or air, as a working fluid, cooling water could be introduced directly into the vapor space. If this cooling water were directed at the evaporation surfaces, a degree of washing would be obtained. This same cooling water, with appropriate design and

valving, could completely flood the tank at the end of the suction cycle, driving off any noncondensable gases. If a scheme such as that given above were combined with an adequate allowance for scale accumulation, the pump would be capable of running for extended periods of time without intervention, and manual cleaning could be limited to regularly scheduled maintenance intervals.

3.3 ORGANICS

Numerous organic fluids were studied for potential use as a working fluid in the direct pumping system. The selection criteria for the working fluids were:

- Possess a boiling point at atmospheric pressure, which is slightly above anticipated ambient temperature or cooling water temperature.
- Preferably have a low solubility in water.
- Preferably be lighter than water.
- Be nontoxic or have low toxicity.
- Be reasonably available and inexpensive.
- Be compatible with other components and stable over the expected working conditions.

Several halogenated hydrocarbons, notably the Freons, were potential candidates. However, their maximum operating temperature should be limited to 300° to 400°F, which is below the potential stagnation temperature of a high-performance thermal collector. When Freons decompose, the products can be fluorides and chlorides, which can be quite hazardous. Pentane, which was the working fluid suggested in the proposal, and ethyl ether meet the stated criteria, but are highly flammable. Thirty-five other organic fluids were investigated, either partially or thoroughly, for possible inclusion. Since pentane meets the selection criteria, and since thermodynamic data for it were available, it was chosen as the principal working fluid for analysis and prototypes. Table 3.1 summarizes some

TABLE 3.1
CHARACTERISTICS OF POTENTIAL WORKING FLUIDS

No.	Refrigerant Name	Chemical Formula	Flammability Explosive Limits (% in air)	Boiling Point (°F)	Safety (ANSI*)	Safety (UL**)	Comments
21	Dichlorofluoromethane	CHCl_2F	Nonflammable	47.8	1	4-5	Decomposes to Phosgene
160	Ethyl Chloride	$\text{CH}_3\text{CH}_2\text{Cl}$	3.7 to 12.0	54.3	2	4a	
11	Trichlorofluoromethane	CCl_3F	Nonflammable	74.9	1	5	
611	Methyl Formate	HCOOCH_3	4.5 to 20.0	89.2	2	3	Explosive
610	Ethyl Ether	$\text{C}_2\text{H}_5\text{OC}_2\text{H}_5$		94.3			
216	Dichlorohexafluoropropane	$\text{C}_3\text{Cl}_2\text{F}_6$		96.2			
30	Methylene Chloride	CH_2Cl_2	Nonflammable	104.4	1	4a	Decomposes to Phosgene
113	Trichlorotrifluoroethane	$\text{CCl}_2\text{FCClF}_2$	Nonflammable	117.6	1	4-5	
1130	Dichloroethylene	$\text{CHCl} = \text{CHCl}$	5.6 to 11.4	118	2	4	

*Group 1 - least hazardous

Group 3 - most hazardous

**See Table 3.2

of the characteristics of several fluids of potential interest. Table 3.2 gives the classification for gases and vapors according to their comparative hazards to life. General information on n-Pentane and Freon 113 is provided in Table 3.3.

The performance potential of a working fluid can be estimated for the basic constant pressure cycle by examining the ratio $N = v_{fg} \Delta P / h_{fg}$, where v_{fg} is the specific volume change, h_{fg} the enthalpy change, and ΔP the pressure change between condensed liquid and hot vapor. The higher the ratio N , the higher the potential efficiency.

The following values were obtained for pentane and Freon 113 between 75°F and 175°F:

$$N_{\text{pentane}} = 0.064$$

$$N_{\text{Freon 113}} = 0.064$$

A graph of N for the two liquids is provided in Figure 3.1. As can be deduced, the efficiency of a constant pressure cycle does not continue to increase with temperature. Both pentane and Freon 113 have similar efficiency potentials for a constant pressure cycle, but different operating pressures.

TABLE 3.2
 UNDERWRITERS' LABORATORIES CLASSIFICATION
 OF COMPARATIVE HAZARD TO LIFE OF GASES AND VAPORS

Group	Definition	Examples
1	Gases or vapors which in concentrations of about $\frac{1}{2}$ to 1 percent for durations of exposure of about 5 minutes are lethal or produce serious injury.	Sulfur Dioxide
2	Gases or vapors which in concentrations of about $\frac{1}{2}$ to 1 percent for durations of exposure of about $\frac{1}{2}$ hour are lethal or produce serious injury.	Ammonia Methyl Bromide
3	Gases or vapors which in concentrations of about 2 to $2\frac{1}{2}$ percent for durations of exposure of about 1 hour are lethal or produce serious injury.	Carbon Tetra- chloride Chloroform Methyl Formate
4	Gases or vapors which in concentration of about 2 to $2\frac{1}{2}$ percent for durations of exposure of about 2 hours are lethal or produce serious injury.	Dichloroethylene Methyl Chloride Ethyl Bromide
Between 4 & 5	Appear to classify as somewhat less toxic than Group 4.	Methylene Chloride Ethyl Chloride
	Much less toxic than Group 4 but somewhat more toxic than Group 5.	Refrigerant 113
5a	Gases or vapors much less toxic than Group 4 but more toxic than Group 6.	Refrigerant 11 Refrigerant 22 Carbon Dioxide
5b	Gases or vapors which available data indicate would classify as either Group 5a or Group 6.	Ethane Propane Butane
6	Gases or vapors which in concentrations up to at least about 20 percent by volume for durations of exposure of about 2 hours do not appear to produce injury.	Refrigerant 12 Refrigerant 114 Refrigerant 13B1

TABLE 3.3
GENERAL CHARACTERISTICS OF n-PENTANE AND FREON 113

n-Pentane*

General Information

Synonym: amyl hydride.

Colorless liquid.

Formula: $\text{CH}_3(\text{CH}_2)_3\text{CH}_3$.

Mol wt: 72.15, bp: 36.1°C , flash p.: -56°F , fp: -129.8°C , d: 0.626 at $20^\circ/4^\circ\text{C}$, autoign. temp.: 588°F , vap. press.: 400 mm at 18.5°C , vap. d.: 2.48, lcl = 1.5%, ucl = 7.8%.

Hazard Analysis

Toxic Hazard Rating:

Acute Local: U.

Acute Systemic: Inhalation 1.

Chronic Local: U.

Chronic Systemic: U.

Toxicology: Narcotic in high concentrations. LC (mice) = 13%.

Fire Hazard: Highly dangerous, when exposed to heat or flame.

Spontaneous Heating: No.

Explosion Hazard: Severe; when exposed to heat or flame.

Explosive Range: 1.4-8.0%.

Disaster Hazard: Highly dangerous; keep away from heat, sparks or open flame; shock can shatter metal containers and release contents.

Countermeasures

Ventilation Control: Section 2.

To Fight Fire: Foam, carbon dioxide, dry chemical or carbon tetrachloride (Section 7).

Storage and Handling: Section 7.

Shipping Regulations: Section 11.

Regulated by CG, DOT, IATA.

Trifluorotrichloroethane*

General Information

Colorless liquid.

Formula: CCl_3CF_3 .

Mol wt: 187.39, mp: 13.2°C , bp: 45.8°C , d: 1.5702, autoign. temp.: 1256°F .

Hazard Analysis

Toxic Hazard Rating:

Acute Local: U.

Acute Systemic: Inhalation 2.

Chronic Local: U.

Chronic Systemic: U.

Fire Hazard: Very slight; when exposed to heat or flame (Section 7).

Disaster Hazard: Dangerous. See chlorides and fluorides.

Countermeasures

Storage and Handling: Section 7.

TABLE 3.3 (Continued)

GENERAL CHARACTERISTICS OF n-PENTANE AND FREON 113

Toxic Hazard Rating Code

- 0 NONE:
 - (a) No harm under any conditions;
 - (b) Harmful only under unusual conditions or overwhelming dosage.
- 1 SLIGHT:
Causes readily reversible changes which disappear after end of exposure.
- 2 MODERATE:
May involve both irreversible and reversible changes not severe enough to cause death or permanent injury.
- 3 HIGH:
May cause death or permanent injury after very short exposure to small quantities.
- U UNKNOWN:
No information on humans considered valid by authors.

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Figure 3.1 Potential Constant Pressure Efficiency

4. COLLECTOR-PUMP CONFIGURATIONS

The collector and pump were originally conceived to be integral, with the pumping chamber acting as its own collector. Incident energy was to enter the pump either through a transparent dome or by heat conduction through an opaque dome that would absorb the incident sunlight.

4.1 TRANSPARENT DOME

The use of a transparent dome on the upper collector-pump surface allows the sunlight to enter the collector and heat the working fluid directly. This has some advantage because, as the water is pumped from the chamber, the location of the working fluid would fall and therefore not be in contact with the upper surface.

Some critical problems arise with the use of a pressurized transparent surface. Transparent plastics, such as acrylic, which have good weatherability, do not have good mechanical properties at elevated temperatures. Glass, which has the ability to survive weathering and is not affected by moderate temperatures, has poor tensile strength and is not as readily formed into a domed shape as are plastics.

One possibility is to use the glass in an inverted position, as shown in Figure 4.1, thereby reversing the tensile load to a compressive load. A second glazing can be placed over the inverted dome to prevent rain and debris from collecting. Even if the mechanical properties of the glazing do not pose a serious problem, the incident light will fall only on the water/working fluid interface either when the sun is nearly overhead, or if the depth of the vapor space is relatively shallow. Otherwise, reflective walls would be required to direct the radiation to the working fluid.

4.2 OPAQUE ABSORBER DOME

The use of a metallic dome eliminates the structural problems of a transparent dome but creates the need for a means of maintaining a thermal path to the working fluid. One means is to utilize wicking or screen, as

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Figure 4.1 Inverted Glass Dome on Combined Collector Pump

in a heat pipe, to draw the working fluid by capillary action from the water interface to the heated dome. Although this is feasible, such wicking action is strongly dependent upon fluid properties and can easily be fouled. One solution is to form the upper surface, as shown in Figure 4.2. In this design, the outer metal surface absorbs the incident sunlight and conducts heat to the lower surface by interconnecting fins. The working fluid is trapped between the two surfaces during condensation when this entire upper unit is cooled by water passages. This unit could be made as a casting, but is relatively complicated and suitable mainly to a larger diameter, which requires significantly more material.

4.3 SEPARATE COLLECTOR AND PUMP

A further development of the above design would be to extend the upper absorbing surface beyond the pump chamber, thereby reducing the structural requirement. If tubes are used to hold the working fluid, and fins to conduct heat to the tubes, the upper surface becomes a horizontal flat-plate collector. The pressure vessel requirements for the entire collecting surface are reduced to the tubular flow passages, and the pumping chamber can be relatively small, improving its utilization. The collector, as shown in Figure 4.3, is constrained to lie in the horizontal position to ensure uniform fluid distribution. In nonequatorial regions, the requirement for a horizontal collector introduces a severe performance penalty. The collector can be tilted and maintained in a flooded condition if valving is added between the collector vapor chamber and the pumping chamber, as shown in Figure 4.4. Now the cyclic heating of the collector mass can be eliminated, thereby reducing heat loss. The condenser can be similarly isolated so that it too is not thermally cycled. If only one pumping chamber is used, the collector serves to store thermal energy during the condensing phase of the cycle; if two pumping chambers are used, however, the collector and condenser are continually utilized.

The major remaining thermal loss mechanism is the cyclic heating and cooling of the pumping chamber walls. As previously described, this can

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Figure 4.2 Metal Cover Configuration on Combined Collector Pump

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Figure 4.3 Horizontal Collector

Figure 4.4 Tilted Collector

be reduced by using an inner insulating cylinder to thermally isolate the pressure vessel walls from the working fluid. The previous disadvantage of increased wall mass is diminished because now the pumping chamber does not have to match the collector area in size. By using a collector area significantly larger than the pumping chamber area and allowing a slight overpressure to be created before connecting the two, a relatively rapid water displacement cycle is achieved. This rapid cycle reduces the time during which heat transfer to the walls can occur.

The advantages of separating the collector and pump chamber are summarized below:

- Reduction of structural problems associated with a large pressure vessel.
- Elimination of the problem of heat transfer to the working fluid.
- Enhanced flexibility in collector orientation.
- Elimination of the cyclic heating and cooling of major system components.
- Design flexibility by the decoupling of the collector and pumping chamber areas.
- Introduction of a more rapid water displacement, thereby reducing thermal losses to the pump chamber walls.

5. ALTERNATIVE THERMODYNAMIC CYCLES

5.1 COMPARISON OF EXPANSION AND NONEXPANSION CYCLES

The additional work that can be obtained by the use of the expansion work of a gas increases as the operating temperature of the system increases. Figure 5.1 shows this relationship for pentane. As can be seen in this figure and in Figure 3.1, the potential thermodynamic efficiency peaks at just over 6 percent for a nonexpansion cycle, whereas the efficiency for an expansion cycle continues to increase with increasing temperature. This relationship can be quantified by the following analysis in which the work of an isobaric-isentropic cycle is determined for various amounts of expansion.

The isentropic pressure-volume relationship of saturated pentane vapor can be approximated by $Pv = \text{constant } K$. The value of the constant K is dependent upon the starting point of the expansion, as shown in Table 5.1. The work extracted from an isobaric-isentropic cycle is given by

$$W = \int_{v_o}^{v_f} (P - P_{\min}) dv + (P_o - P_{\min}) v_o$$

where $P = K/V$ and $P_{\min}$ is the back pressure, or

$$W = (P_o - P_{\min}) v_o + K \ln \frac{v_f}{v_o} - P_{\min} (v_f - v_o).$$

A maximum in the work is obtained when $P = P_{\min}$ or $V_f/V_o = P_o/P_{\min}$. The above expression can be normalized by the nonexpansion work $(P_o - P_{\min}) v_o$ to obtain

$$\frac{W}{W_{NE}} = 1 + \frac{\ln \frac{v_f}{v_o}}{1 - \frac{P_{\min}}{P_o}} - \frac{\frac{v_f}{v_o} - 1}{\frac{P_o}{P_{\min}} - 1}.$$

Figure 5.1 Thermodynamic Cycle Efficiency Versus Evaporation Temperature

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TABLE 5.1
CONSTANTS FOR ISENTROPIC EXPANSIONS
FOR VARIOUS PRESSURES

Temperature (°F)	Pressure (psia)	K psi (ft ³ /lbm)
125	24.42	82.54
150	36.68	84.46
175	53.11	85.90
200	74.5	86.81

Figure 5.2 shows the results obtained for the various initial pressures of Table 5.1 and a back pressure of 10 psia, which corresponds to a condensing temperature of 77.5°F or 25°C. The maximum work obtainable for each pressure is indicated with a star above the last data point. The three auxiliary lines that are shown indicate where 80 and 90 percent of maximum work is obtained, and where an expansion to atmospheric pressure (14.7 psia) ends. The results for the fraction of maximum work are tabulated in Table 5.2.

It is evident from the table that large expansion ratios are not required to obtain high fractions of the total available work. Note that the final pressures are significantly above atmospheric pressure.

The "fluid flywheel" concept, as described in Section 5.2, utilizes the energy storage of the water column not only to extract expansion work, but to allow the use of a higher initial pressure, and therefore collector temperature, to achieve improved thermodynamic efficiency. Because of this difference, a higher expansion ratio is required in order to reduce the pressure to a level below the pumping head and to recover the stored energy.

5.2 "FLUID FLYWHEEL" CYCLE

The concept of leveling the output of a power source or the input requirements of a load, by means of an energy storage device, has been applied in almost all applications where power is generated or consumed. This storage can also be considered as a form of impedance matching. The most common form of storage for rapid cycles of power equipment is a flywheel, where excess mechanical energy is stored during part of a cycle, then retrieved when there is a power deficit. The directly coupled solar water pump concept is, in its purest form, one that has no conversion to mechanical energy. Because of this lack of mechanical energy, a fluid analog was sought that would allow the expansion work of the working fluid to be extracted. This process, because of the working fluid properties, is one in which there is a great variation in the output impedance of the pump. Similarly, the combination of the collector and the working fluid

Figure 5.2 Total Work Relative to Nonexpansion Work as a Function of Expansion Ratio for Various Initial Pressures

TABLE 5.2
EXPANSION RATIOS AND FINAL PRESSURES
FOR VARIOUS INITIAL PRESSURES

Initial Pressure (psia)	90% of W_{max}		80% of W_{max}	
	v_f/v_o (psia)	P_f (psia)	v_f/v_o (psia)	P_f (psia)
24.42	1.55	15.8	1.25	19.5
36.68	2.1	17.5	1.62	22.6
53.11	2.8	19.0	2.06	25.8
74.5	3.7	20.1	2.6	28.7

has a maximum theoretical efficiency at a particular operating temperature and pressure, which will not, in general, correspond to the particular pumping characteristics encountered. Therefore, a means of matching the output impedance of the pump to the required pumping task is also sought.

The cycle developed during the program uniquely solves both problems without the need for a mechanical interface stage. This is possible because the inertia of the pumped water column can be utilized as the energy storage medium. A working fluid is selected whose pressure at the desired operating temperature exceeds the pumping pressure requirement. When this excess pressure is applied to the directly coupled water pump, the water column accelerates until either the pressure is removed or the sum of the flow friction pressure drop and the elevation head equals or exceeds the pumping pressure. Beyond this point, the flow will decelerate; in doing so, however, it allows additional water to be pumped by pressure that is below the pumping head requirement. If the friction pressure drop is not allowed to become excessive, then the efficiency of the pump is improved.

Figure 5.3 shows an idealized cycle where the directly coupled pump of Figure 5.4 is fed working fluid vapor at constant pressure during the first part of the cycle. In the second part of the cycle, the volume of working fluid vapor within the pumping chamber is isolated from the source and allowed to expand as the water continues to flow from the pump. During the final phase, after the water flow has ceased, the working fluid is condensed in an external condenser and fresh water is drawn into the pumping chamber. Because of the short duration of the cycle, heat transfer to the pumping chamber is minimized and the expansion approaches isentropic conditions.

The thermodynamic process for this cycle is shown in Figure 5.5. The real collector and pump, shown in Figure 5.6, would have a temporal response, as shown in Figure 5.7, and a process representation, as shown

Figure 5.3 Idealized Pumping Cycle

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Figure 5.4 "Fluid Flywheel" Pump

P-V RELATIONSHIP

T-S RELATIONSHIP

Figure 5.5 Idealized Thermodynamic Process Representations

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Figure 5.6 Collector and Pump Configuration

Figure 5.7 Pumping Cycle

in Figure 5.8. The impedance between the pressure source and the pumping requirement can be matched by varying the length of time (or level of displacement) during which the pump chamber is connected to the pressure source.

The "fluid flywheel" concept can be readily extended to a mechanical system for deep-well pumping, as shown in Figure 5.9. A deep well requires some form of down-well pump, and a "sucker rod" pump is a simple, reliable, and efficient mechanical pump. The weight of the sucker rod adds to the system inertia and a counter weight can be placed on a lever arm to both decrease the mass required and increase the effective inertia. Such a mechanical system can be impedance matched by setting the mechanical advantage from the pump chamber piston rod to the suction rod to match the pumping requirements. The counter weight can be sized to take the best advantage of the condenser water temperature. An alternative to using the "fluid flywheel" is to make use of a mechanical system that has a nonlinear linkage. Such a system is described in Section 5.4.

The "fluid flywheel" cycle can be roughly approximated by a constant pressure/constant entropy expansion, as shown in Figure 5.4. The excess energy (EE) can be estimated from the following analysis:

$$EE = \int_{V_1}^{V_2} (P - P_H) dV + \int_{V_2}^{V_H} (P - P_H) dV$$

where P is the pressure in the pumping chamber and P_H is the pumping head, as shown in Figure 5.5. The absorbed energy (AE) is given by

$$AE = \int_{V_H}^{V_3} (P_H - P) dV.$$

Pentane and Freon, which are suitable working gases, have a pressure volume relationship for the isentropic expansion of superheated vapor that can be approximated by $Pv = \text{constant}$. If a lossless operation is assumed,

P-V RELATIONSHIP

T-S RELATIONSHIP

Figure 5.8 Thermodynamic Process Representations

Figure 5.9 Deep-Well Pump

then $EE = AE$. If the reservoir volume is very large compared to the pumping chamber volume $P_1 = P_2$, and if the dead volume in the pumping chamber is equal to zero ($V_1 = 0$), then

$$EE = (P_2 - P_H) V_2 + \int_{V_2}^{V_H} \left(\frac{K}{V} - P_H \right) dV$$

and

$$AE = \int_{V_H}^{V_3} \left(P_H - \frac{K}{V} \right) dV.$$

Equating the two expressions and solving the integrals yields

$$P_2 V_2 + K \ln \left(\frac{V_H}{V_2} \right) - P_H V_H = P_H V_3 - P_H V_H + K \ln \left(\frac{V_H}{V_3} \right)$$

but

$$P_2 V_2 = P_H V_H = P_3 V_3 = K$$

so

$$1 + \ln V_H - \ln V_2 - 1 = \frac{P_H}{P_3} = \frac{V_3}{V_H}$$

or

$$1 + \ln \frac{P_2}{P_3} = \frac{P_H}{P_3}$$

$$\text{The excess energy is } EE = K \ln \frac{V_H}{V_2} = P_2 V_2 \ln \frac{P_2}{P_H}.$$

From the above analysis P_3 can be determined given $P_1 = P_2$ and P_H . For $P_1 = P_2 = 53.11$ psia, which is equivalent to a pentane temperature of 175°F, values of P_3 have been determined for various pumping heads and these can be found in Table 5.3.

A realistic limit on the amount of possible work is set by the suction or condensation pressure. Pentane has a vapor pressure of 12 psia at a temperature of 86°F. If the condensing water is at 68°F (20°C), then an

TABLE 5.3
IDEAL VALUES FOR FINAL PRESSURES AND
FRACTION OF EXPANSION WORK EXTRACTED

Initial Pressure, 53.11 psia

Minimum Realistic Pressure, 12 psia

P_H (psia)	P_3 (psia)	M_{act}/M_{max}
19.7	6.3	1*
24.7	8.85	1*
34.7	15.6	0.96
44.7	26.2	0.67

*Calculated values of P_3 are below the minimum pressure.

18°F temperature difference exists for heat transfer and temperature rise. The maximum amount of expansion work can be estimated by

$$W_{\max} = \int_{V_2}^{V_3} (P - P_{\min}) dV$$

or

$$W_{\max} = \int_{V_2}^{P_2 V_2 / P_{\min}} \left(\frac{K}{V} - P_{\min} \right) dV.$$

Then

$$W_{\max} = K \ln \frac{P_2}{P_{\min}} - P_{\min} V_2 \left(\frac{P_2}{P_{\min}} - 1 \right)$$

and the actual work extracted is

$$W_{\text{act}} = K \ln \frac{P_2}{P_3} - P_{\min} V_2 \left(\frac{P_2}{P_3} - 1 \right).$$

The ratio between actual and maximum work is then

$$\frac{W_{\text{act}}}{W_{\max}} = \frac{\ln \frac{P_2}{P_3} - \frac{P_{\min}}{P_2} \left(\frac{P_2}{P_3} - 1 \right)}{\ln \frac{P_2}{P_{\min}} - \frac{P_{\min}}{P_2} \left(\frac{P_2}{P_{\min}} - 1 \right)}.$$

Values of $W_{\text{act}}/W_{\max}$ corresponding to $P_2 = 53.11$ psia, and $P_{\min} = 12$ psia can be found in Table 5.3.

The cycle is potentially quite efficient except when the pumping head approaches the source pressure. Even for the case when the pumping head is 44.7 psia with a source pressure of 53.11 psia, two-thirds of the expansion work is extracted.

Alternative thermodynamic cycles that utilize the "fluid flywheel" principle can be designed, and two examples are provided. Other cycles that are intermediate to these are evident, but the cycle of primary importance

is the one with an approximately constant pressure expansion followed by an isentropic expansion, because it provides good design flexibility. The first alternative cycle, shown in Figure 5.10, consists of a pressurized dead volume that is allowed to expand isentropically. Starting pressures must be very high to achieve higher pumping heads. The second alternative cycle, shown in Figure 5.11, relies upon the transfer of liquid, instead of vapor, to the pumping chamber. The liquid flashes and expands in the pumping chamber. The liquid system is less efficient but allows a metered quantity of working fluid to be introduced into the pumping chamber.

The major source of loss, and therefore inefficiency, for a "fluid flywheel" pump is pipe friction loss. The pipe flow velocity can be estimated from the excess energy and the mass of water, M , in the pipe by equating the excess energy with the fluid kinetic energy.

$$P_2 V_2 \ln \frac{P_2}{P_H} = \frac{1}{2} M \dot{x}^2$$

where $\dot{x}$ is the flow velocity, and

$$M = \rho_W L \frac{\pi d^2}{4}$$

where L is the pipe length, ρ_W the density of water, and d the pipe diameter; therefore,

$$\dot{x} = \left(\frac{2P_2 V_2}{M} \ln \frac{P_2}{P_H} \right)^{\frac{1}{2}}$$

By maintaining a large pipe diameter, flow velocity and losses can be minimized. The "fluid flywheel" will demand a larger pipe diameter than would be needed for a continuous pump of the same capacity because the flow pulsates in the "fluid flywheel" pump. A compliance chamber or accumulator can be located after a length of pipe so that the increased diameter is not required over the entire run of pipe. Care must be taken in the design and

P-V RELATIONSHIP

T-S RELATIONSHIP

Figure 5.10 Expansion Cycle Starting with a Fixed Volume of Vapor

P-V RELATIONSHIP

T-S RELATIONSHIP

Figure 5.11 Expansion Cycle with Liquid Injection

selection of components to avoid unnecessary flow losses from the pipe entrance, check valve, bends, and diffuser.

5.3 JET PUMP

Although direct pumping of water is simple and involves minimal moving parts, its thermodynamic efficiency is relatively poor. If some method of increasing this efficiency can be found, without seriously compromising reliability or cost, then the expense of pumped water will decrease in direct relation to the increase in efficiency. One approach to accomplishing this increase in efficiency is to raise the working fluid pressure above the head required by the pumping situation and then allow it to expand to the lower pressure. This will result in an increase in efficiency only if the added work available during the expansion is converted to useful work; i.e., pumped water. The excess pressure available must result in the increased flow of water.

One method of extracting some of the available work is with a jet pump, or water jet eductor. The jet pump is a simple device with no moving parts. It utilizes the kinetic energy of a liquid moving at high velocity to entrain and mix with a second liquid. The mixed liquids are then passed through a diffuser and discharged at some static pressure lower than that of the initial motive fluid.

A schematic drawing of a jet pump is shown in Figure 5.12. Motive fluid, supplied by the pumping tank, is accelerated to a high velocity through the nozzle. This high-velocity fluid entrains suction fluid in the mixing chamber to produce a uniformly mixed stream traveling at a lower velocity. The diffuser then reduces stream velocity, increasing static head to that required at discharge. The jet pump units are simple and compact, have no moving parts, and are self-priming. A pictorial drawing of the jet pump used in conjunction with a simple direct-acting pump is shown in Figure 5.13. A jet pump used in conjunction with the more complex reciprocating pump system described in Section 6.3 is shown in Figure 5.14. In each case, in addition to the jet pump, a check valve is

Figure 5.12 Jet Pump Configuration

Figure 5.13 Jet Pump Augmented Direct-Acting Pump

Figure 5.14 Jet Pump Augmented Reciprocating Pump System

required in the suction line. Proper utilization of jet pump augmentation for the extraction of expansion work also requires some minor modification of the working fluid system. Specifically, the working fluid inlet valve must close before the pumping cycle is completed to allow the vapor to expand fully.

The idealized cycle proposed is as follows:

- Working fluid is heated in the solar collector until it reaches a predetermined pressure greater than that required to pump.
- A valve between the tank and the collector is opened, allowing vapor to displace water. If the stored volume of hot working fluid is sufficiently large, this approximates constant pressure pumping. Flow is augmented by the jet pump.
- After a predetermined displacement (or drop in pressure) the working fluid valve is closed. The working fluid in the tank continues to expand until its pressure reaches the static head that the system is pumping against. During this phase of the pumping, jet pump augmentation will occur until the velocity through the nozzle drops below a lower limit, at which point suction flow will cease and total flow will equal motive flow.
- The vapor in the tank is condensed, drawing water into the tank and allowing the cycle to be restarted.

To determine the performance of a jet-pump-augmented system, it was deemed necessary to use real jet pump performance characteristics. These performance data are empirical and include the efficiencies of mixing, as well as nozzle and diffuser losses, etc. The operating characteristics of three standard jet pumps, each with a different nozzle to throat area ratio, were determined from extensive data published by the manufacturer.

It is a characteristic of the jet pump that, over a wide operating range, a dimensionless flow parameter,

$$M = \frac{Q_{\text{suction}}}{Q_{\text{motive}}}$$

is a linear function of a pressure parameter, N . N is usually defined as

$$\frac{P_{\text{out}} - P_{\text{suction}}}{P_{\text{motive}} - P_{\text{suction}}}$$

However, to simplify the calculation of the combined pumping cycle, the denominator in the equation for N was replaced with P_{motive} in absolute pressure units. A plot of M versus N , calculated for one of these pumps from tabular data, is shown in Figure 5.15. A linear regression program was then used to calculate the coefficients of an equation of the form $M = b - aN$ for the three selected pumps at two suction heads and three discharge heads. Correlation coefficients for these equations were very high, typically 99 percent. It was then a simple matter to set the limits of the equation. The upper limit is determined by the point at which suction flow ceases to increase with an increase in motive pressure; this flow is readily discernable from the tables. The lower limit is determined by the motive pressure at which suction flow goes to zero.

It is then necessary to relate jet pump suction flow to total flow. For the first pumping phase, when the valve from the solar boiler is open, this is straightforward. Pressure is considered constant, the displacement from the tank is predetermined, and the ratio of suction flow to motive flow (tank flow) is obtained from the previously derived linear equation.

For the second phase of the cycle, with the working fluid inlet valve closed, the expansion can be considered isentropic. For an isentropic expansion, absolute pressure can be related to specific volume by the equation $Pv^k = C$. For Freon 113 and pentane, setting $k = 1$ provides a reasonably close approximation (see Figure 5.16), so that

$$v_{\text{tank}} = C_1 / P_{\text{tank}}$$

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Figure 5.15 M vs. N Using Absolute Pressure

Figure 5.16 Actual PV Relationship of Superheated Pentane Compared to $P = \frac{C}{V}$

or, for a unit mass

$$V_{\text{tank}} = C_1 / P_{\text{tank}}$$

$$dV_{\text{tank}} = -C_1 / P_{\text{tank}}^2 dP_{\text{tank}}$$

For the jet pump

$$M = C_2 N + C_3$$

where

$$M = \frac{Q_{\text{jet}}}{Q_{\text{tank}}}$$

but

$$Q = \frac{dV}{dt}$$

$$M = \frac{dV_{\text{jet}}/dt}{dV_{\text{tank}}/dt} = C_2 \frac{P_{\text{out}} - P_{\text{suc}}}{P_{\text{tank}}} + C_3$$

$$dV_{\text{jet}} = \left(C_2 \frac{P_{\text{out}} - P_{\text{suc}}}{P_{\text{tank}}} + C_3 \right) dV_{\text{tank}}$$

$$= -C_1 \left(C_2 \frac{P_{\text{out}} - P_{\text{suc}}}{P_{\text{tank}}} + C_3 \right) \frac{1}{P_{\text{tank}}^2} dP_{\text{tank}}$$

$$dV_{\text{total}} = dV_{\text{jet}} + dV_{\text{tank}}$$

$$= -C_1 \left(C_2 \frac{P_{\text{out}} - P_{\text{suc}}}{P_{\text{tank}}} + C_3 + 1 \right) \frac{1}{P_{\text{tank}}^2} dP_{\text{tank}}$$

$$V_{\text{total}} = -C_1 \int_{P_1}^{P_2} \left(C_2 \frac{P_{\text{out}} - P_{\text{suc}}}{P_{\text{tank}}^3} + \frac{C_3 + 1}{P_{\text{tank}}^2} \right) dP_{\text{tank}}$$

$$V_{\text{total}} = \frac{C_1}{P_{\text{tank}}} \left(C_3 + 1 + C_2 \frac{P_{\text{out}} - P_{\text{suc}}}{2P_{\text{tank}}} \right) \Bigg|_{P_1}^{P_2}$$

The constants C_2 and C_3 are already determined from the equation relating M to N . C_1 is readily determined from the working fluid properties listed, and P_2 and P_1 are the upper and lower limits over which the linear relationship between M and N is valid, or over which the pump is operated, whichever is more restrictive.

The final pumping phase is again straightforward. During this phase there is no augmentation from the jet pump, and the water pumped is equal to the water displaced from the tank as the working fluid expands from P_2 to P_{out} . The total water pumped is then the sum of the water pumped in the three phases just discussed.

The thermodynamic efficiency is the work done on the water, divided by the heat added to the working fluid. Work done on the water is the product of the total volume of water pumped and the difference between output pressure and suction pressure. Not surprisingly, in all cases investigated, the thermodynamic efficiency of the jet-pump-augmented system is greater than the equivalent constant-pressure pumping system. However, since the working fluid must be raised to a higher temperature, the collector efficiency is lower. Figures 5.17 through 5.20 plot the thermodynamic and overall efficiencies of jet-pump-augmented systems, utilizing the single-glazed selective-surface collector described in Section 6.1. The efficiencies of the equivalent constant-pressure pump in each case are shown for comparison.

Although the jet pump offers a means of obtaining expansion work with the introduction of few complications or moving parts, the improvement in efficiency has proven to be minimal, at best. At very low pressures, the use of a jet pump will permit a smaller tank to be used for a given volume of water pumped, but this savings is obtained at the expense of somewhat increased control complexity. The jet pump seems to offer no clear advantage for the direct water pumping concept.

Figure 5.17 Thermodynamic Efficiency and Total Efficiency vs. Starting Pressure for a Jet-Pump-Augmented Pump. A Single-Glazed Collector is Assumed.

Figure 5.18 Thermodynamic Efficiency and Total Efficiency vs. Starting Pressure for a Jet-Pump-Augmented Pump. A Single-Glazed Collector is Assumed.

Figure 5.19 Thermodynamic Efficiency and Total Efficiency vs. Starting Pressure for a Jet-Pump-Augmented Pump. A Single-Glazed Collector Is Assumed.

Figure 5.20 Thermodynamic Efficiency and Total Efficiency vs. Starting Pressure for a Jet-Pump-Augmented Pump. A Single-Glazed Collector is Assumed.

5.4 NONLINEAR MECHANICAL LINKAGE FOR WORKING FLUID EXPANSION

A mechanical system was investigated for the expansion of the working fluid because deep-well pumping necessitates some form of down-well unit. Two alternatives to the transformation to mechanical energy were considered. The first was the piping of working fluid down well, and the second was the use of a hydraulic coupling to a down-well unit. The use of working fluid down well complicates the entire system, and a second pump must be used to return the condensate to the collector. Thermal losses are also increased and pump installation is far from simple. A hydraulic down-well unit, shown in Figure 5.21 and described in the proposal, requires two pipes, one to deliver hydraulic power to the down-well unit and the other to deliver pumped water to the surface. If the "fluid flywheel" principle is to be employed to expand the working fluid, then the diameter of the pipes should be as large as possible, which implies that each will be approximately one-half of the well hole diameter. The mechanical "fluid flywheel" permits either lower fluid flow losses or a smaller drilled hole because the mechanical connection can be solid and therefore of smaller diameter. The nonlinear mechanical linkage does not require flow rates as high as those for the "fluid flywheel" because the variable mechanical advantage closely matches the working fluid pressure during expansion. Because the impedances are well matched, kinetic energy storage requirements and actuating forces are greatly reduced. A moderately fast stroke can be used to reduce heat losses to the expansion chamber walls, but flow losses can still be reduced by an order of magnitude.

One nonlinear mechanical linkage selected for this application is shown in Figure 5.22. The linkage consists of two colinear points (X_1 and X_2) connected together at a common point (X_p, Y_p) by links R_1 and R_2 , respectively. The trajectory of X_p, Y_p is controlled, and mechanical advantage is set by this trajectory. The links can be operated in either a tension mode, as shown in Figure 5.23a, or a compression mode, as shown in Figure 5.23b. If a complementary linkage is added, then the transverse forces

Figure 5.21 (a) + (b) Deep-Well Foot Pump
(c) + (d) Nonexpanding Engine Combined with Deep-Well Pump

Figure 5.22 Trajectory Controlled Nonlinear Mechanical Linkage

Figure 5.23 Nonlinear Mechanical Linkages for Isoberic-Isentropic Expansion

on points X_1 and X_2 will be cancelled, reducing bearing loads considerably. The analysis of the motion of the three points X_1 , X_2 and X_p , Y_p found in the Appendix yields the following differential equation for the trajectory $f(X_p)$ of the point X_p , Y_p :

$$f'(X_p) = \frac{MA(X_1) - 1}{\left[\left(\frac{R_1}{f(X_p)} \right)^2 - 1 \right]^{-\frac{1}{2}} + MA(X_1) \left[\left(\frac{R_2}{f(X_p)} \right)^2 - 1 \right]^{-\frac{1}{2}}}$$

where $MA(X_1)$ is the mechanical advantage of the system referred to the motion of X_1 . Given R_1 , R_2 , $MA(X_1)$, and the distance between X_1 and X_2 , the differential equation can be numerically integrated to produce the required trajectory.

The mechanical advantage can be calculated by assuming a pressure volume characteristic for the working fluid. An idealized constant pressure expansion followed by an isentropic expansion can be assumed. If a relatively rapid stroke is desired, then an isentropic expansion based upon the combination of the collector, vapor reservoir, and expansion chamber is followed by an isentropic expansion based upon the expansion chamber alone. Two examples of idealized expansions with Freon 113 are shown in Figure 5.23.

A mechanical system can be biased and impedance matched to obtain the maximum performance from a particular working fluid. A bias force, F , which can be a spring or a weight, is used to match the suction or minimum working fluid pressure to the inlet pressure as follows:

$$F - P_{\min} A_1 = (P_a - P_i) \frac{A_2}{MA_{\min}}$$

where A_1 is the working fluid piston area, A_2 the pump chamber piston area, P_a the ambient pressure, P_i the inlet pressure, $P_{\min}$ the minimum working fluid pressure for the operating condenser temperature, and $MA_{\min}$ the minimum mechanical advantage.

The impedance is matched by the following relationship:

$$P_{\max} A_1 - F = (P_d - P_a) \frac{A_2}{MA_{\min}}$$

where $P_{\max}$ is the maximum working fluid pressure that occurs at the optimum operating conditions, and P_d is the delivery pressure. Combining these two equations yields:

$$F = \frac{A_1}{1 + \frac{P_a - P_i}{P_d - P_a}} \left[P_{\min} + P_{\max} \left(\frac{P_a - P_i}{P_d - P_a} \right) \right]$$

and

$$A_2 = \frac{MA_{\min}}{P_d - P_a} (P_{\max} A_1 - F).$$

Therefore, F is set by the pressure expected, A_2 is set by these pressures, and the minimum mechanical advantage designed into the linkage.

An inversor linkage, shown in Figure 5.24, is an alternative nonlinear linkage. It does not allow an exact match to expansion characteristics of a working fluid but it does allow a matching close enough that the energy storage requirements for the fluid/mechanical system are significantly below those of the "fluid flywheel" system. Its advantage over the tracked linkage of Figure 5.22 is the simplification of the linkage by replacing the track with a third link. A more complete analysis can be found in the Appendix.

5.5 REGENERATOR

One common method of increasing the efficiency of a thermodynamic engine is to use the exhaust from the engine to regeneratively heat fluid being returned from the condenser to the boiler. This recovers some of the heat available in the exhaust rather than rejecting it at the condenser. Due to the noncontinuous nature of the direct-acting pumps, regeneration becomes somewhat more difficult. At best, fluid must be stored and pumped back, through a heat exchanger, to the boiler at the start of a condensation

$$X_2 = \frac{A^2 - B^2}{X_1}$$

$$\sqrt{A^2 - B^2} < X_1 < A + B \text{ IF } A > B$$

$$B - A < X_1 < A + B \text{ IF } B > A$$

Figure 5.24 Inversor Linkage

cycle. This complexity would perhaps be worthwhile if a major increase in efficiency were obtained. The amount of heat that can be transferred is limited, however, due to the small average temperature difference.

Consider a direct-acting, constant-pressure pump, operating over the range of 30.4 meters-water column positive head to 3.7 meters-water column suction, with pentane as the working fluid.

For a bulk quantity of fluid being cooled by a second fluid that is pumped through a heat exchanger at a constant rate, the heat exchange is described by the following correlation:

$$-\frac{UA}{WC} = \ln \left(1 - \frac{MC}{Wc\theta} \ln \frac{T_1 - t_1}{T_2 - t_1} \right)$$

where

U = overall heat transfer coefficient

A = transfer area (relative to U)

W = mass flow rate of cooling fluid

M = mass of fluid to be condensed

c = heat capacity of liquid pentane

C = a dummy heat capacity, explained in Section 6.2

θ = time

T_1 = initial tank temperature

T_2 = final tank temperature

t_1 = initial cooling fluid temperature

Letting UA be infinitely large,

$$\frac{MC}{Wc\theta} \ln \frac{T_1 - t_1}{T_2 - t_1} = 1$$

From the properties of pentane,

$$\ln \frac{T_1 - t_1}{T_2 - t_2} = 0.197$$

$$\frac{T_1 - t_1}{T_2 - t_2} = 1.217$$

$$\frac{83.3 - 23.9}{T_2 - 23.9} = 1.217$$

$$T_2 = 72.7$$

$$\frac{h}{h_{\text{total}}} = 0.078.$$

The total amount of heat that could be transferred with even an infinitely large heat exchanger is less than 8 percent of the reject heat, even at a moderately large head. In a solar-heated system, this would be reflected in a somewhat smaller solar collector. Due to condensing time constraints, the size of the condenser would be essentially unaffected. Offsetting the savings in collector size would be the need for an additional heat exchanger, the regenerator, and a considerable increase in control complexity. The savings in system cost using a regenerator would be minimal, at best. The additional control complexity would, moreover, degrade system efficiency. Accordingly, regeneration was not considered in any of the cycles studied in this program.

6. COMPONENT STUDIES

6.1 NATURAL CONVECTION BOILER

An alternative to heating the working fluid directly in the pumping tank is to utilize an external solar-powered boiler. Separating the heating and pumping functions allows optimization of each component for its particular function, and eliminates, or at least minimizes, some conflicting requirements between the two. The boiler, perhaps a flat-plate solar collector, can be properly tilted and oriented for maximum collecting of available insolation. The tank, on the other hand, can be fully insulated. Also, the tank head design is limited only by the constraints of pressure and cost. In many cases, it could prove quite practical to use readily available water tanks.

In the remote locations where pumps such as these will be operated, it is desirable that any fluid circulation through the boiler be accomplished without external power. This is not, however, a particularly serious restraint, since natural convection will provide more than adequate circulation. A schematic drawing of such a boiler is shown in Figure 6.1. Fluid circulates through the boiler because of differences in density between the riser and the downcomer.

If the tanks, headers, and return tubes of the boiler are properly insulated, all fluid in the system will be at the same temperature. In a properly evacuated system, this temperature will be the saturation temperature. As heat is added to the riser, which in this case is part of the solar collector, some of the saturated liquid boils to vapor. As the fluid continues up the tube, its vapor quality continues to increase, with an accompanying decrease in density. In the separator tank, the vapor rises out of the liquid to be tapped off for pumping. The liquid then returns through the downcomer to make up for lost volume.

It is the difference in density between the liquid in the downcomer, at essentially zero vapor quality, and the fluid in the riser, at a vapor

Figure 6.1 Natural Convection Boiler

quality greater than zero, that provides the motive force for the boiler. A force balance of a system in equilibrium shows that the motive force, which is a direct function of the height, Z , and the density difference between the fluid in the downcomer and that in the risers, must balance frictional losses through the system. An interesting and useful aspect of this mode of boiler operation is that the circulation through the individual tubes tends to be self-balancing. If heat input to a tube should change, the density difference would change accordingly. The flow through the tube would increase or decrease in response to the density change. Thus, a shaded tube, for instance, would receive less flow than would parallel tubes in full sunlight. Also, since each riser tube is essentially self-pumping, orifices or other devices are not required to balance flow.

A flat-plate collector developed at Thermo Electron was investigated for use in a natural convection mode for direct water pumping, and a scaled-down module of this collector, only slightly modified, was used for pump testing. The section used for testing compromised one bay of the collector shown in Figure 6.2. A drawing of a cross section of the collector is shown in Figure 6.3, and its collection performance in Figure 6.4. Each absorber plate of the collector is a 13.2-cm-wide aluminum extrusion. This extrusion is formed so as to tightly grip a standard $\frac{1}{2}$ -in.-o.d. copper tube. The absorber has a selective surface of black chrome plating.

The only modification to the collector mandated by the natural circulation mode was the addition of downcomers. One downcomer, also of $\frac{1}{2}$ -in.-o.d. copper tubing, per module, was found to be more than adequate for this function.

Fossil-fuel-fired boilers generally incorporate rather elaborately baffled vapor separators at the top of the risers to ensure high-quality vapor leaving the tank. However, with the low power densities encountered in solar collectors, this complexity is not required. At vapor velocities below 1 m/sec, liquid entrainment is essentially nonexistent. With the

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Figure 6.2 Thermo Electron Solar Collector

Figure 6.3 Solar Collector Cross Section

Figure 6.4 Performance of Solar Collector

standard header on this collector, the vapor velocity is substantially below this upper limit.

Natural convection is a time-proven method of boiler circulation. It allows circulation of fluid without the need for external power or auxiliary pumps. This approach works well with normal tilt angles of collectors, and allows the collector to be oriented as required to maximize solar collection. This boiler must run nearly full at all times, however, and does not lend itself to schemes requiring the boiler to run dry. In pumping schemes that are not hampered by this limitation, the natural convection boiler offers the best collection efficiency of any of the approaches considered, with no cost penalty involved.

6.2 CONDENSER

In all systems, except the single-cycle-per-day pump, where the working fluid can cool overnight, some form of active condensing is required. This condensing is most reasonably accomplished using some, or all, of the water pumped. The exact configuration of the condenser, the amount of water used for condensing, and even whether to locate the condenser inside or outside the pumping chamber, are subject to a number of system considerations. Each pumping scheme elaborated in this report has unique requirements and constraints that affect the condenser design. Additionally, individual operating parameters will have an effect on the condenser.

The final difference between the cooling-water temperature and that of the working fluid condensate is a major factor in the size of the condenser. The pumping head, which determines the work required per unit of water, can also be a major factor. The ability to condense fully the working fluid with the water pumped can in fact constitute one system limit on pumping depth. This limitation will be explored briefly to establish a perspective on the upper limit of water-cooled condensing.

Because of limitations on maximum system temperatures, a deep-well pump will probably require mechanical augmentation. It is necessary to assume a number of operating conditions to arrive at the limiting head. Assume a system using pentane as the working fluid operating between 84° and 24°C. Assume further that the pumping is done at constant pressure, with no expansion work, and that the mechanical pump efficiency is 90 percent. Let the initial water temperature be 10°C.

A diagram of temperature versus enthalpy is shown in Figure 6.5 for condensation of pentane over the operating range chosen. The broken line shows a straight-line approximation of this curve, based on the end points, over the constant volume phase of condensation. To provide a first approximation of the quantity of water required for cooling, the condensation can be modeled as two problems. The first of these, the constant volume condensation, can be modeled as a tank of fluid with a dummy heat capacity, C , defined as the change in enthalpy divided by the change in temperature; i.e., the slope of the line in Figure 6.5. The second phase, the constant pressure condensation, can be modeled as a counterflow heat exchanger, with the pentane having a heat capacity approaching infinity.

Since we are attempting to determine the minimum quantity of water required, we can assume an infinitely large exchanger area.

For a tank of fluid being cooled by a constant water flow through a coil or jacket, the heat transfer is described by the relation:

$$e^{\frac{-UA}{WC}} = 1 - \frac{MC}{W_c \theta} \ln \frac{T_1 - t_1}{T_2 - t_1}$$

where T_1 = initial temperature of the tank fluid
 T_2 = final temperature of the tank fluid

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Figure 6.5 Condensing Thermodynamics Chart

t_1 = initial temperature of the cooling water

θ = time

W = mass flow rate of the cooling water

c = heat capacity of the water

C = heat capacity of the tank fluid

M = mass of tank fluid

U = overall heat transfer coefficient

A = effective transfer area

Since UA has been considered infinitely large,

$$\frac{MC}{Wc\theta} \ln \frac{T_1 - t_1}{T_2 - t_1} = 1$$

The mass of water used $m = W\theta$.

Inserting the dummy heat capacity described earlier, the heat capacity of water, and the temperatures stated for the problem,

$$\frac{m}{M} = 2.71.$$

For the isobaric condensation, again assuming an infinitely large heat exchanger, the mass ratio of water required would be the change in enthalpy of the pentane, divided by the product of the heat capacity of the water and the temperature difference between the incoming water and the pentane:

$$\frac{m}{M} = \frac{\Delta h}{c\Delta T} = 1.14.$$

Summing the two

$$\left(\frac{m}{M}\right)_{\text{total}} = 3.85.$$

This value is of course based on an infinitely large heat exchanger. A real system will require a higher ratio of water mass to pentane mass since the theoretical maximum quantity of heat cannot be exchanged. Also, with a real heat exchanger, with a finite area, the overall transfer coefficient is important. This coefficient changes considerably over the isovolumic condensing phase, due both to the dependence of the thermal transport properties of the fluids on temperature and to the sensitivity of the condensing coefficient to the temperature gradient across the condensate film. To allow for this temperature dependence, as well as the nonlinear relationship between temperature and enthalpy, the isovolumic condensation was solved in successive temperature steps. A multiple-pass, shell-and-tube heat exchanger was adopted as the condenser style to be used. A condenser design was then developed based on the stated operating conditions and the volume displacement of the test pumps constructed for this program. A mass ratio of 4.3 was found to provide a good compromise among condenser size, condensing time, and water consumption.

The thermodynamic work available over the range of 24° to 84°C for a constant pressure pump is approximately 8.5 kWh/kg of pentane. For the assumed 90-percent mechanical conversion efficiency, 7.6 kWh/kg is put into the water to be pumped. If this amount of water is 4.3 kg/kg of pentane, then it can be pumped against a head of 650 meters-water column. For a greater temperature range, or a more efficient thermodynamic cycle, even higher heads are obtainable.

Several methods of using pumped water to provide condensation were considered. They include:

- A heat exchanger in a water storage tank or pond.
- Pumping all the water to a higher head than required by the application and allowing it to drain back through a condenser.

- Storing some of the pumped water and running it through the condenser when required.
- Opening a valve to a lower head storage container, after the constant volume pumping has ceased, allowing expansion work to supply cooling water.
- Providing a sufficiently large heat sink to start suction, then using suction flow to finish condensing and cool the heat sink.
- Increasing the volume of the system sufficiently to start suction flow, and then using suction water for condensation.
- Using the output from one pump to cool and condense the working fluid of the other, in a multiple pump system, where the pumps work out of phase.

Consider first a multiple pump system with the pumps operating out of phase in a manner described in Section 6.3.2.2. For this type of system, an external condenser is the obvious choice. One condenser is adequate for the two or more pumping tanks in use. All of the water output can be used for cooling, with the only loss in efficiency being the work required to overcome the small pressure drop necessary to maintain turbulence in the water as it flows through the heat exchanger. Since all of the output water is used to absorb heat, the water temperature rise is kept to a minimum. The greater temperature difference allows a decrease in the surface area and the cost of the condenser. A single external condenser does require appropriate valving to switch between tanks. This valve can be operated in conjunction with the boiler valve, however, so it adds little to the complexity of the system. In addition, having a valve between the condenser and the pumping tank eliminates a heat loss to the condenser during condensation.

With a single pump system or, alternately, a multiple pump system with all the pumps operated in parallel, the ejected water cannot be used directly, but must be stored at least temporarily for use in condensing. Water on the suction side of the pump can be used directly, since flow

occurs at the proper time. To obtain this flow, however, pressure in the pumping tank must first be dropped to a level where suction flow can occur. Two methods of obtaining this pressure drop have been investigated: providing a heat sink with a capacity adequate to induce suction, and which is subsequently re-cooled once suction flow has begun; or increasing the dead volume by connecting a second low-pressure chamber sufficiently large that the working fluid expansion results in an adequately low pressure.

The feasibility of the heat sink concept depends on the quantity of heat that must be accepted before active cooling begins, and upon the temperature difference over its operating range. Unfortunately, isovolumic condensation represents a substantial amount of the heat that must be removed during the condensation phase of the constant pressure cycle. The percentage of heat that must be stored in the heat sink compared to the total heat that must be removed is shown in Figure 6.6; this value is plotted for a range of discharge heads at two suction heads. To further aggravate the already large heat capacity requirement, the temperature range over which the sink can operate is very limited. At best, the temperature of the sink can be no greater than the temperature at which isobaric condensation starts, and no less than the temperature of the pumped water. Practical considerations as to condensing time and available surface area require that this temperature range be even less.

One possible heat sink would be some or all of the water previously pumped. The pumped water has many times the heat capacity required for total condensation of the working fluid. To provide heat exchange between the working fluid and the stored water, it is necessary to provide a large contact area between them. A coil connected to the working fluid could be immersed in a tank of water, for instance, or tubes full of water could be exposed to the working fluid.

An approach such as this is certainly possible. The thermal conductivity of water is quite low, however, and without turbulent flow to provide mixing, the water-side heat transfer coefficient is poor. To

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Figure 6.6 Heat Transferred during Isovolumic Condensation as a Percentage of Total Heat Transferred During Condensation

offset the poor heat transfer, it is necessary to greatly increase the surface area. This is generally accomplished with finned surfaces. In this case, however, consideration must be given to fouling and corrosion, since untreated water will be used. This severely limits surface enhancement and requires, instead, many more tubes or passages - an expensive solution.

Alternatively, a large mass of metal could be used. Aluminum, for instance, is relatively inexpensive and light and has excellent thermal conductivity. Choice of the proper alloy, or of the proper surface treatment, will provide corrosion resistance in most water conditions that will be encountered. The worst and best cases of Figure 6.6 give a perspective on the quantity of metal required. In the worst case, the water is drawn up 5 m and discharged against a head of 50 m. At the best, it is drawn up 1 m and discharged against a head of 5 m. In both cases, the working fluid is pentane. Assume an initial water temperature of 10°C. A stored heat capacity of 4.8×10^5 J/kg is required for the isovolumic condensation in the high head case; for the low head case, 1.7 J/kg is adequate. The temperatures of pentane at -5 meters-water column pressure is 18.6°C and, at -1 meters-water column pressure, 33°C. Allowing a 2°C differential between the heat sink and each of the fluids, the sink will operate over a 4.6°C range in the former case, and a 19°C range in the latter. Using aluminum for the heat sink, with a heat capacity of 962 J/kg-°C, 109 kg aluminum per kg of pentane is required for the high head case, and 9.39 kg of aluminum per kg of pentane for the low head case. The amount of metal required for the high head case is unrealistically large; even for the low head case, it is questionable, and other alternatives are available.

An alternative method of dropping the pentane pressure is to connect the pumping chamber to a large-volume condenser at low pressure. A valve is required to isolate the condenser during the pumping cycle. At the end of the pumping stroke, the valve is opened allowing the working fluid to flow into the condenser until the pressures are in equilibrium.

Even making the unrealistic assumption that a perfect vacuum can be obtained in the second chamber, its volume will have to be greater than 1.6 times the volume of water displaced for the low head situation, and more than 9 times as great for the high head case.

One final method would be to cool liquid pentane to near water temperature in the condenser and then, when required, mix it with the pentane vapor. The quantity of liquid working fluid required is 37.88 kg/kg and 4.22 kg/kg for the same two cases. The mass for the low head case is certainly manageable. This approach, however, does require extra components, including valves and a second heat exchanger, to cool the next charge of liquid working fluid. Also, since all the heat is transferred at the lowest temperatures, much more exchange area is required.

Two of the schemes for utilizing the pumped water are quite straightforward. Either all of the water pumped can be stored in an intermediate reservoir and then allowed to run down through a heat exchanger when required, or only as much water as is required can be stored for use. The differences between the two approaches are minor. The first, since it uses all the available water, requires a smaller heat exchanger. It requires a much larger intermediate storage tank, however, and the water must be pumped to a higher head than would be required by site conditions. In the second case, the amount of water that must be stored is much smaller, typically 5 percent or less, and the discharge head requirement is not changed; but the cooling water is lost unless a use can be found for it at a lower head. In each of these approaches, condensing represents a parasitic drain on pumping work. The overall effect on cycle efficiency is quite low, however, and either alternative appears quite practical. The choice between them will depend on a number of system considerations, and must include the site requirements and expected ambient conditions.

For systems pumping against a relatively high head, a third alternative is available. When the pump has reached the end of its constant pressure stroke, a valve is closed between the boiler and the tank, and another valve in the water line is opened connecting the remaining water in the tank to a storage container at a head lower than the original pumping head. The working fluid in the tank will expand to this lower head, expelling more water and losing some heat. This water is then used for condensing. Since the water is obtained with no additional heat input, it causes no loss in efficiency. Indeed, if a use can be found for this water, it represents an increase in pump work and system efficiency.

The use of suction and discharge water can also be combined. Discharge water can be used for the isovolumic condensation, and suction water for the isobaric condensation. This would decrease the parasitic loss more than is immediately apparent, since the mean temperature difference is much greater during the isovolumic phase; consequently, less water is required per unit of heat rejected. However, this loss reduction is obtained only at an increase in system complexity, a complexity probably not warranted in most situations with a direct-acting pump.

We have been concerned thus far with condensing in relation to direct-acting pumps. At least two other types of pumps require consideration. These two are the stored momentum, or "fluid flywheel", expansion pump and the deep-well pump requiring a mechanical interface for force amplification.

With the "fluid flywheel" pump, the period of pump ejection is, by design, very short. In addition, it is important to minimize the restriction in the pump output plumbing due to the high velocities involved. These restrictions preclude the direct use of output water from interacting multiple pumps for condensing. The use of temporarily stored water is, of course, as applicable to the "fluid flywheel" pump as it is to the direct-acting pump. The "fluid flywheel" pump offers a unique opportunity to utilize suction water, however. In many instances, the pump will expand

to near atmospheric, or even subatmospheric, pressures; consequently, little or no isovolumic condensation is required. In addition, because of the rapidness of the expansion cycle, no effort is made to prevent heat transfer from the working fluid to its surroundings. If the pumping tank is cooled, probably by surrounding it with a commercially available water jacket, then the tank can provide any isovolumic condensing that may be required, as well as the isobaric condensation once suction flow commences. This approach is very simple, requires no additional valves, causes no efficiency losses, and is quite cost effective.

The indirect pump, operating through a mechanical linkage, will most probably be employed in deep-well pumping applications, where the available suction head of a direct-acting pump is exceeded. This application precludes the use of suction water even if the working fluid is expanded to below atmospheric pressure. Also, the quantity of water pumped for a given amount of work will decrease with increasing head. This will make the tapping-off of a portion of the pumped water for condensing increasingly less attractive with increasing pumping depth. Temporary storage of all of the pumped water, at a slightly higher head than the final head required, is the best solution for a single-cylinder pump. It is quite simple and will have the least effect on efficiency. For multiple pumps operating out of phase, or for a dual-cylinder system driving a double-acting pump, the use of water from one pump to cool the working fluid of the second is still the best choice.

In selecting the configuration of the heat exchanger used for the condenser, consideration should be given to cost, reliability, and maintenance. It must be capable of operating for extended periods of time using low-quality water, and it should be possible to clean away accumulated silt and scale periodically to provide for long-term use. The shell-and-tube heat exchanger fits these requirements quite well. It is a type of heat exchanger that is readily available in a large number of sizes and configurations from several manufacturers, and is easily tailored to site requirements. The inside diameter of the tubes provides an unobstructed

water flow path, and the outside of horizontal tubes provides an excellent self-draining surface for condensation. Although a number of surface-enhancement techniques could reduce the size of the exchanger, most are not particularly worthwhile. Within reason, size is not a major consideration for this component. Surface extenders and fins would add to the cost per foot so that overall price reduction would be minimal. Any extended surfaces on the water side would increase the likelihood of fouling.

Convolute tubing, properly sized, has the potential of improving heat transfer on both the water and condensing sides of the tube, although at the cost of increased pressure drop. The tube would also be more prone to fouling than would a round tube, and it would be harder to clean once fouling did occur. Also, exchangers of this type would be a custom design. One technique that might prove worthwhile would be to coat the tubes with Teflon. The nonwetting characteristic of this surface would improve the condensing film coefficient; the Teflon coating, as long as its integrity was maintained, would protect the tube. Such exchangers are available commercially, although on a much more limited scale than are conventional shell-and-tube heat exchangers. The worth of the coating would best be established experimentally.

The shell-and-tube exchanger would be mounted externally to the pumping tank, and generally higher than the boiler. It is also possible to mount condensing tubes, or a coil, inside the pumping tank. This type of mounting would eliminate the need for the housing or "shell" of the heat exchanger, but would complicate the construction of the pumping tank. In addition, any water remaining in the tubes would be exposed to working fluid during the next pumping cycle, and would extract heat from the working fluid, decreasing system efficiency.

In any pumping scheme, but particularly in the multiple-cycle-per-day pumping systems, certain problems of fluid relocation must be addressed. Specifically, noncondensable gas that infiltrates the system must be removed, and condensed working fluid must be returned to the

boiler. In each case, the fluid must be moved from a source at a low pressure to a sink at a higher pressure; therefore, some form of active pumping must be employed.

The noncondensable gases in the system, primarily air, can most readily be removed from the condenser, where they will be highly concentrated. One method of moving these gases from the subatmospheric condenser to a receiver at atmospheric pressure is with a jet pump. This device, discussed in Section 5.3, operates on the principle of momentum exchange. High-pressure vapor, bled from the boiler, is accelerated through a nozzle to near sonic velocities. Upon leaving the nozzle, the vapor mixes with the gas to be pumped, imparting velocity to it. Leaving the mixing chamber at a lower, but still high, velocity, the combined gas and vapor enters a diffuser where the velocity is reduced to near zero, causing a large rise in static pressure. This mixture of noncondensable gas and working fluid vapor then enters a reservoir, where the vapor condenses for return to the system after gravity separation of the noncondensable gases.

This pump represents a parasitic drain on the pumping system. Vapor that would otherwise pump water is used to pump gas. The magnitude of this loss is of course dependent on the quantity of gas to be pumped. In a well-designed system, the loss can be kept acceptably low.

Two methods have been considered for returning condensed working fluid to the boiler. The first uses a jet pump or, more properly, a variation of the jet pump, to return the fluid. The second uses an intermediate receiver and an arrangement of valves.

The latter case is shown schematically in Figure 6.7. During one condensing cycle, the three-way selector valve connects the condenser to the reservoir, which is lower and, since both tanks are at equal pressure, fills by gravity. The check valve below the reservoir prevents gas from the boiler from flowing into the reservoir.

Figure 6.7 Method of Returning Working Fluid from Condenser Using an Intermediate Storage Tank

During the next pumping cycle, the selector valve closes off the condenser and connects the reservoir to the boiler. With pressure in the reservoir identical with that in the boiler, condenser fluid can return to the boiler under the influence of gravity. The selector valve can, in a multitank pumping system, operate concurrently with the pumping tank selector valves.

The jet pump mentioned earlier is the automatic boiler injector. This device, as might be inferred from the name, is used to inject feed water into boilers and is available as a stock item, although some modifications will probably be required to accommodate the special conditions of this pump.

The boiler injector functions in a manner identical to the jet pump described for use in the removal of noncondensable gases, except that the high-velocity vapor leaving the nozzle condenses immediately upon contact with the condensate to be pumped. The large reduction in volume, and therefore velocity, of the vapor results in a large rise in static pressure. This rise in static pressure can result in a pressure higher than the pressure of either the suction or motive fluids. Since the pressure is higher than motive (or boiler) pressure, return flow of the condensate can occur.

By the classic thermodynamic definition of efficiency; that is, the quotient of work out divided by heat in, the boiler injector does quite poorly. The efficiency is typically on the order of a few percent. This poor efficiency is attributable to the large amount of heat given up to the suction fluid. In this particular application, heat given up to the pumped condensate is not really lost, but serves to preheat the condensate prior to its return to the boiler. If the heat recycled to the boiler is included with the pumping work done, the boiler injector approaches 100-percent efficiency.

6.3 PUMP ACTUATORS

Preliminary testing of the open-tank, directly coupled water pump system showed the importance of physical isolation between the water being pumped and the thermodynamic working fluid. In addition to the need for isolation is the requirement for actuator simplicity, economy, and reliability. The actuator designs chosen for this system involve the use of an elastomeric diaphragm. The fabric-reinforced elastomeric diaphragm is commonly used for many types of actuators and has demonstrated high reliability when used in a water pumping application. Diaphragm pumping units are especially valuable if the pumped water contains particulates, which tend to accelerate wear in many other types of pumps. A thermal system of this nature requires good thermal isolation between the pumped water and the system working fluid. The diaphragm is constructed of low-conductivity materials, and its inherent insulating ability can be increased by the addition of elastomeric insulating materials during fabrication.

Diaphragms can be used purely as isolators with no appreciable pressure drop taken across the unit, or they can be used to extract reciprocating shaft power, taking the full system pressure load. Commercially available diaphragm pumps operate in both modes; reliability estimates are in the range of 1 million cycles, with some reports of pumps lasting more than 2 million cycles.

Because the flat diaphragm in its simplest form is an inherently short stroke device, variations upon that simple configuration are necessary in order to make a pump actuator of practical diameter with a suitable output displacement. A commonly used technique is to make the diaphragm dish-shaped with steep sides, allowing the "dish" to turn inside out, giving the unit a stroke capacity equal to twice the dish depth. This type of diaphragm is best used as an isolation unit, with its maximum stroke limited to about half of its diameter. If this type of diaphragm is to be used as a power actuator, its thickness must be increased to the point where a significant amount of power is lost while flexing the reinforced elastomer.

The rolling diaphragm concept is useful for both power actuation and long strokes. It has a very thin sidewall operating in tension and has very low friction and hysteresis properties, resulting in efficient operation with high reliability. This type of diaphragm requires an outer cylinder and an adequately guided inner piston to properly restrain the unit and ensure smooth rolling action. Its stroke capacity is about $1\frac{1}{2}$ times its diameter. It is important to note that the rolling diaphragm must be provided with a positive pressure differential to properly form the rolling shape and to avoid abrasion of the sidewall.

6.3.1 Deep-Well System

The deep-well pumping-system actuator will be a power actuator with the output shaft suitably coupled to a down-well pumping unit, as shown in Figures 6.8 and 6.9. Because the actuator needs to provide long-stroke reciprocating shaft power to the pumping mechanism, the rolling diaphragm principle is best suited for this unit. Consultation with a local diaphragm manufacturer resulted in a preliminary design specification estimating the minimum cycle life of the unit to be 750,000 cycles. The manufacturer warns, however, that this specification is valid only if there are no pressure reversals seen by the diaphragm.

The actuator design for the deep-well system, presented in Figure 6.10, will provide reciprocating shaft power to operate the pushrod of the down-well pumping unit. The actuator involves a piston centrally guided within a cylinder. The diaphragm rolls between the cylinder and the piston, isolating the system working fluid and converting system gas pressure to shaft force. The design for this actuator is based on a 10-in.-diameter rolling diaphragm with a stroke capability of about 8 in. The effective area of this diaphragm is 72.75 in.² when it is used with the specified 9.25-in.-diameter piston, giving a volume displacement of 582 in.³ at an 8-in. stroke.

Figure 6.8 Deep-Well Actuator

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Figure 6.9 Mechanism to Connect Deep-Well Actuator to Down-Well Pump

Figure 6.10 Single-Diaphragm Shallow-Well Actuator

The housing will be of thin-walled steel construction insulated to limit heat losses from the system. A control valve is mounted on the housing and will be operated by a follower rod attached to the piston. The valve is ported directly into the actuator cylinder and is positioned so as to collect and return any condensate from the cylinder.

6.3.2 Shallow-Well System

Two actuator designs are planned for the shallow-well pumping system, using dished diaphragms as the isolation between the pumped water and the system working fluid. The first actuator design involves a single diaphragm relying on subatmospheric condensing pressure to provide suction, whereas the second actuator uses two diaphragms connected by a pushrod and operating in a double-acting mode.

6.3.2.1 Single-Diaphragm, Shallow-Well Actuator

The design for the single-diaphragm, shallow-well pump actuator, shown in Figure 6.10 consists of two thin-walled steel dishpan-shaped housing shells clamping a similarly shaped diaphragm. One housing shell is the pumped water side and it carries the water inflow and outflow check-valves. The other housing shell is the system working fluid side and to it is attached the control valve. As in the deep-well actuator, the housing shell is penetrated by a follower rod to actuate the control valve. The diaphragm used in this actuator has an 8-in. diameter and a 2-in. height, with an effective area of 47 in.², giving a volume displacement of about 190 in.³.

6.3.2.2 Twin-Diaphragm, Double-Acting Pump Actuator

A design for the double-acting pumping unit is presented in Figure 6.11. It consists of two actuators, similar to the single diaphragm unit, that are connected by a rod. The connecting rod is used to actuate the control valve; it also causes one diaphragm to have positive inlet suction while the opposite diaphragm is expelling water. This principle reduces the pumping system's dependence on low condensing temperatures to ensure

Figure 6.11 Double-Acting Pump Actuator

inlet suction. This actuator uses the same diaphragm that is used in the single-diaphragm unit, giving a volume displacement of 190 in.³ per half-stroke, or 380 in.³ total displacement per stroke. The control valve, mounted between the two diaphragm pumping units, supplies pressure alternatively to the two pumping chambers.

6.4 CONTROL VALVES

6.4.1 Single-Acting Actuator Valves for Shallow- and Deep-Well Systems

Two control valves have been designed for the single-acting pumping systems, one for the constant-pressure operating mode and another for the "fluid flywheel" mode. Either valve can be used on either single-acting actuator, depending on which operating mode is desired.

6.4.2 Constant-Pressure Control Valve

The control valve for the constant-pressure system is a directly actuated spool valve controlled by a follower rod attached to the actuator diaphragm. The diaphragm follower rod shifts the pilot spool from one control position to the other at the completion of the actuator stroke, the spool being magnetically latched at either control position. Following is a detailed description of the operation of the valve shown in Figure 6.12 as one complete cycle is analyzed:

- The deep-well-system pumping cycle will begin with the pump having completed its fill stroke. At this time the valve spool has been pushed to its eject position by the diaphragm follower rod and is magnetically latched into that position.
- With the spool in the eject position, the boiler pressure is introduced into the pump chamber, causing the actuator diaphragm and follower rod to move in the eject direction. The control-valve spool remains latched in the eject position as the diaphragm continues to stroke.

Figure 6.12 Control Valve for Constant Pressure Deep-Well System

- Near the end of the eject stroke, the condense spring contacts the valve spool and begins to compress. The spring continues to compress until it reaches its solid height, at which point the force applied to the spool is sufficient to free it from its magnetic latch. The compressed spring pushes the spool to its latched condense position, connecting the pump chamber to the condenser.
- As the working fluid vapors condense, the actuator diaphragm and follower rod begin to return to the filled position, and the spool remains latched in the condense mode. As the actuator nears the end of its fill stroke, the eject spring compresses, pushing the spool free of its condense latch when the spring reaches its solid height. The eject spring then pushes the spool to its eject position, beginning the actuator eject cycle again.

6.4.3 "Fluid Flywheel" System Control Valve

The control valve for the "fluid flywheel" pump system (Figure 6.13) is a spool valve arranged to remain open for a fixed percentage of the eject stroke, limiting the amount of vapor introduced into the pump actuator. It will then return to neutral for the remainder of the inlet stroke and switch to the condense mode at the end of the eject stroke. It will remain in the condense mode until the completion of the fill stroke, at which time it will switch to the open position, beginning the eject stroke again. The control valve is used in series with a boiler sequence valve (Figure 6.14) to ensure full-stroke system startup when the boiler is slowly increasing in pressure, such as when starting from a cold condition. The valve is designed to remain in the closed position until the boiler pressure rises to the minimum required for full-stroke operation. The valve will then open fully and remain open during the stroke. In the condense mode, the sequence valve will close if the boiler pressure drops to a level below the opening pressure.

Figure 6.13 Control Valve for "Fluid Flywheel" System

Figure 6.14 Sequence Valve

The "fluid flywheel" system control valve is operated as follows:

- The pumping cycle begins with the sudden application of pressure, onto the inlet port, caused by the opening of the boiler sequence valve. At this point, the pump is fully filled, the diaphragm plate is fully returned, and the follower rod eject pin is holding the valve spool in the open position. With the spool in this position, the boiler pressure is connected to the pump actuator.
- The eject pilot valve is closed, preventing the pump pressure (P_3) from being able to pressurize the pilot reservoir. This resultant spool pressure imbalance ($P_3 - P_4$) holds the spool against its eject stop, supplying the pump actuator with vapor from the boiler. The spool pressure imbalance is maintained by the pilot valve during the beginning of the eject stroke. As the pump actuator strokes, the follower rod moves the pilot valve toward its open position. When the pilot valve opens, the pilot pressure (P_4) equalizes with the pump pressure (P_3), balancing the spool and allowing the centering springs to return the spool to the neutral position. The length of the pilot valve determines the fraction of the pump stroke during which the control valve is open.
- The pump eject stroke continues with the control valve closed, expanding the vapor introduced during the first part of the stroke. Near the end of the eject stroke, the follower rod condense pin pulls the spool toward the condense position, and the spool magnetic latch holds it in the condense mode. At this point the spool is pressure balanced and the pump actuator is connected to the condenser.
- The pump pressure collapses into the condenser and the pump begins its fill stroke. As the diaphragm plate and follower rod nears the end of its fill stroke, the eject pilot valve closes, and the spool is pushed off its condense latch and the compressed spring forces it towards the eject position.

- The spool is pushed through neutral, and the inlet port begins to open. As the pump pressure (P_3) increases, the spool is again forced against the eject stop because of the pressure imbalance ($P_3 - P_4$). The pump begins to move toward the eject position, beginning the pumping cycle again.

During all normal operating modes the pump will stroke fully, shifting the valve spool from the eject position to the condense position. There is, however, the possibility that the pump may not complete its eject stroke, preventing the valve spool from being pulled to its condense position. This is possible if there is a temporary blockage of the pump outflow or if the pressure head becomes too high. In this case, the actuator will stop moving in mid-stroke. If this happens, the system will correct itself by the over-pressure valve, shown in Figure 6.15, which will open as the collector temperature begins to rise. When the over-pressure valve opens, it permits the boiler to pressurize the actuator even though the spool is in the neutral position. The actuator is displaced to the condense position in a constant pressure stroke; then the normal sequence will resume.

6.4.4 System Sequence Valve

The sequence valve is normally closed, the valve body being forced to the left by the spring. Boiler pressure is applied to the valve body end, and actuator pressure is applied to the differential area at the first spool land. The actuator pressure remains at condenser pressure until the boiler pressure exceeds the level required to compress the sequence valve spring. As the valve opens, boiler pressure is connected to the actuator through the actuator control valve, and the increasing actuator pressure acting upon the differential area forces the sequence valve spool to the right, against the spring. The pumping chamber pressure maintains the sequence valve in the open position even though the boiler pressure has decreased to a level below the opening requirement. When the actuator reaches the end of stroke, it is connected to the condenser, and the sequence valve is forced to the left and closed by the compressed spring unless the boiler pressure is above the opening value.

Figure 6.15 Boiler Excess Pressure Valve

6.4.5 Double-Acting, Shallow-Well System

The double-acting, shallow-well pump will operate on either pumping principle. The control valve for this actuator is mounted between the two pumping diaphragms. It is of the spool type and is directly operated by the diaphragm connecting rod, being alternately shifted from one magnetic latch to the other. A description of the constant pressure valve operation through one couple pumping cycle follows:

- The cycle begins with the actuation diaphragms shifted fully to the left (Figure 6.11). At this point, the right-side pumping chamber is filled and the left pumping chamber is emptied. The spool has been shifted to the left by the diaphragm connecting rod, opening the pressure port to the right-side pumping chamber and opening the condenser to the left-side chamber.
- The right-side pumping chamber pressurizes, beginning diaphragm motion to the right. The diaphragm connecting rod causes the left-side pumping diaphragm to follow the displacement of the right-side diaphragm, creating a positive suction on the left side as water is being expelled from the right side.
- Near the end of the displacement toward the right, the spool is contacted by the left-side spool control spring. As the stroke continues, the spring exerts sufficient force to break the spool away from its left-side magnetic latch and push it through neutral to its right-side magnetic latch. Now the porting is reversed, with the right-side pump chamber connected to the condenser and the left-side chamber connected to the boiler. The pumping direction is now to the left, with the left-side chamber expelling the water it drew in and with the right-side diaphragm being pulled to the left, drawing in water to be pumped on the next cycle.
- As the diaphragms approach full stroke to the left, the control valve is shifted to the left, beginning the pumping cycle over again.

6.5 BOILER FEEDPUMP

It is planned to use an injector pump to return the condensate to the boiler. This type of pump uses the velocity and condensation of a jet of vapor taken directly from the boiler to pump back into the boiler an amount of condensate many times the weight of the vapor consumed. In so doing, it also heats the condensate to a temperature close to the boiler temperature. The scheme is to provide a reservoir for condensate with a float valve to operate the injector pump. In this manner, the condensate will be returned to the boiler when a sufficient quantity has been accumulated. A design for this system is shown in Figure 6.16. The condensate enters the reservoir from the bottom of the condenser. When the condensate level reaches a suitable level, the float valve opens, beginning the flow of hot pressurized vapor through the injection pump. The injection pump returns fluid under pressure to the boiler until the level in the condensate reservoir has dropped to the level at which the float valve closes. Injection pumps for boiler feed applications are available as an off-the-shelf component, and have a reputation for being reliable and effective.

Figure 6.16 Boiler Feedpump

7. COMPUTER SIMULATION STUDIES

Analytic models of both the constant-pressure (isobaric) expansion system and the isobaric-isentropic expansion system were constructed. The isobaric-isentropic system is actually an isentropic-isentropic system but with differing system volumes. The dynamics of the system are important because pipe friction and heat losses are important mechanisms affecting system efficiency. The transient dynamics of both the "fluid flywheel" system and the simple constant pressure system are orders of magnitude faster than those during the more steady-state heating of the working fluid.

A proprietary simulation procedure was utilized that dynamically optimizes the time step of the simulation so that excess computer time is not utilized. Differential equations governing both the thermodynamics of the working fluid and the thermal and mechanical dynamics of the remainder of the system were derived and implemented in the system model. Simulations have been written utilizing n-Pentane, Freon 113, and air as the working fluids of interest. Air was included because it was used, along with n-Pentane, as a working fluid in the "fluid flywheel" pumps.

7.1 THERMODYNAMIC MODEL

The thermodynamic characteristics for both the saturated region and the superheated region were included in the simulation. When the fluid was in the two-phase region, the following analysis was utilized to yield pressure as the dependent variable:

$$h = (1 - x)h_f + xh_g$$

$$\text{or } h = h_f + xh_{fg}$$

$$v = v_f + xv_{fg}$$

and

$$v = \frac{V}{M} = \frac{AL + V_o}{M}$$

where M is the working fluid mass, V the volume, A the pumping chamber area, and L the displacement of water from the chamber.

$$dq = Pd v + du$$

$$h = u + Pv$$

$$dh = du + Pd v + v dP$$

Therefore,

$$dh = dq + v dP$$

or

$$dP = \frac{dh - dq}{v}$$

$$x = \frac{v - v_f}{v_{fg}}$$

$$x = \frac{\frac{V_o}{M} - v_f}{v_{fg}}$$

$$dh = dh_f + x dh_{fg} + h_{fg} dx$$

$$dv = dv_f + x dv_{fg} + v_{fg} dx$$

and

$$dv = (A dL - v dM) / M$$

$$dL = \dot{L} dt$$

The enthalpy, h , and the specific volume, v , are functions of pressure, P , as well as quality, x , in the two-phase region. Therefore,

$$dh = \frac{\partial h_f}{\partial P} dP + x \frac{\partial h_{fg}}{\partial P} dP +$$

$$\frac{h_{fg}}{v_{fg}} \left(\frac{A}{M} dL + v \frac{dM}{M} - \frac{\partial v_f}{\partial P} dP - x \frac{\partial v_{fg}}{\partial P} dP \right).$$

Solving for dP yields

$$dP = \frac{\frac{h_{fg}}{v_{fg}} \left(\frac{A}{M} dL + v \frac{dM}{M} \right) - dq}{v - \frac{\partial h_f}{\partial P} + x \left(\frac{h_{fg}}{v_{fg}} \frac{\partial v_{fg}}{\partial P} - \frac{\partial h_{fg}}{\partial P} \right) + \frac{h_{fg}}{v_{fg}} \frac{\partial v_f}{\partial P}}$$

In the saturated region, $T = f(P)$.

The thermodynamics in the superheated region were simulated by implementing the following analysis:

$$dh = dq + v dP$$

and

$$z = \frac{Pv}{RT_a} \quad (\text{note that } T_a \text{ is absolute temperature})$$

$$h = f_h(T, P) \text{ and } z = f_z(T, P)$$

$$dh = \frac{\partial h}{\partial T} dT + \frac{\partial h}{\partial P} dP$$

$$P = \frac{zRT_a}{v}$$

$$\begin{aligned} dP &= \frac{\partial}{\partial T} \left(\frac{zRT_a}{v} \right)_v dT + \frac{\partial}{\partial v} \left(\frac{zRT_a}{v} \right)_T dv \\ &= \frac{R}{v} \left(z + T_a \frac{\partial z}{\partial T} \right) dT - \frac{RT_a}{v} \left(\frac{1}{v} - \frac{\partial z}{\partial v} \right) dv. \end{aligned}$$

The term $\partial z / \partial v$ can be shown to be negligible in comparison with $1/v$ for the range of interest, so

$$dP = \frac{R}{v} \left(z + T_a \frac{\partial z}{\partial T} \right) dT - \frac{RT_a}{v^2} dv$$

and

$$T_a = \frac{Pv}{Rz}$$

A similar expression can be derived for the differential of temperature:

$$dT_a = \frac{v}{R} \left(\frac{1}{z} + P \frac{\partial \left(\frac{1}{z} \right)}{\partial P} \right) dP + \left(\frac{P}{Rz} + \frac{Pv}{R} \frac{\partial \left(\frac{1}{z} \right)}{\partial v} \right) dv$$

and neglecting the last term gives

$$dT_a = \frac{v}{Rz} \left(1 - \frac{P}{z} \frac{\partial z}{\partial P} \right) dP + \frac{P}{Rz} dv.$$

Correlations for h and z were derived for n-Pentane and Freon 113.

Combining the differential equation for temperature and the differential equation for enthalpy yields the following differential for pressure, which is utilized in the simulation:

$$dP = \frac{-dq + \frac{\partial h}{\partial T} \frac{P}{Rz} dv}{v - \frac{\partial h}{\partial P} - \frac{v}{Rz} \frac{\partial h}{\partial T} \left(1 - \frac{P}{z} \frac{\partial z}{\partial P} \right)}.$$

The absolute temperature is derived from the definition for compressibility z.

7.2 SYSTEM MODEL

The model for the simulation for the system is shown in Figure 7.1. Various valve arrangements were implemented which enabled several versions of constant pressure and "fluid flywheel" systems to be simulated.

The governing differential equations and other relevant equations follow:

$$dT_2 = \left(\frac{A}{CM} \right)_2 [K_{T1} - (T_2 - T_a) U_{2a} - (T_2 - T_1) U_{21} K_1] dt$$

$$dT_3 = \left(\frac{A}{CM} \right)_3 [-(T_3 - T_a) U_{3a} - (T_3 - T_1) U_{31} - (T_3 - T_w) U_{3w}] dt$$

$$dT_4 = \left(\frac{A}{CM} \right)_4 \left[-(T_4 - T_a) \left(U_{4a} + \frac{4 \Sigma \dot{L} \left(\frac{CM}{A} \right)_4}{A_4 D_4 + 4L} \right) - (T_4 - T_1) U_{41} \right] dt$$

Figure 7.1 Model of Simulated System

$$dM_5 = \frac{U_{41} A_4}{h_{fg}} (T_4 - T_1) dt$$

$$dM_6 = \frac{U_{31} A_3}{h_{fg}} (T_3 - T_1) dt$$

$$dq = \frac{1}{M_1} \left[(T_2 - T_1) U_{21} K_1 A_2 + (T_3 - T_1) U_{31} A_3 + (T_4 - T_1) U_{41} A_4 \right] dt$$

$$dL = \dot{L} dt$$

$$\dot{dL} = \frac{P_A \cdot A}{M_{H_2O}} dt$$

The variables dP and dv are given by the equations of the thermodynamic model. In the simulation of the "fluid flywheel" pumping system, an additional set of thermodynamic variables was included to account for the working fluid that was contained within the pumping chamber during the period when the chamber was isolated from the remainder of the system.

The remaining equations are:

$$A_4 = \pi D_4 L + A \text{ when } L > 0$$

$$A = \frac{\pi D_4^2}{4}$$

$$M_1 = M_1^0 - M_5 - M_6$$

$$P_A = \begin{cases} P + (H - L) \rho g - P_F = P_d, & \text{when pumping} \\ P + (H - L) \rho g + P_F = P_i, & \text{when filling} \end{cases}$$

$$M_{H_2O} = \begin{cases} A_d \rho L_d \left(\frac{D_4}{D_d} \right)^4, & \text{when pumping} \\ A_i \rho L_i \left(\frac{D_4}{D_i} \right)^4, & \text{when filling} \end{cases}$$

The flow is assumed to be turbulent for a rapid pumping stroke, as in the "fluid flywheel", and laminar for a constant pressure stroke.

$$P_F = \frac{\rho V_p^2}{2} \cdot 4f \cdot \frac{L_p}{D_p}$$

$$V_p = L \left(\frac{D_4}{D_p} \right)^2$$

for turbulent flow,

$$f = \frac{0.046}{Re_p^{0.2}}$$

The subscript p becomes d for pumping and i for filling. For laminar flow,

$$f = \frac{16}{Re_p}$$

The boiling correlation for n-Pentane was generated by extrapolating the pool boiling data for n-Pentane found in the Handbook of Heat Transfer. The curves relate the difference between wall temperature and saturation temperature to the heat flux as a function of absolute pressure. The following equation for the film coefficient was derived and provides a good approximation to the extrapolated curves in the region of operation:

$$h = 6.6 \times 10^{-9} P^{4.1} \Delta T (10.75 - 4.69 P^{0.1}).$$

The film coefficient, h, is in Btu/hr-ft²-°F when P is in psia and ΔT is the difference between the wall temperature and the saturation temperature in °F. A discussion of the verification of this correlation can be found in Chapter 9.

The effective condensing film coefficient for n-Pentane and the effective film coefficient for the condensing water were determined by first considering a specific condensing case as described in Section 6.2.

Based upon these calculations, a general equation for an effective overall film coefficient was derived so that condenser parameters could be varied for simulation runs. The equation for the effective overall film coefficient

$$U_{EFF} = \frac{W}{A} \left[1 - e^{-\frac{A}{W} \left(\frac{1}{U_{Cond}} + \frac{.0684}{\left(\frac{W}{A}\right)^{0.8}} \right)^{-1}} \right]$$

was then separated into water and working fluid film coefficients as follows

$$U_{3w} = 3.0 U_{EFF}$$

$$U_{31} = 1.5 U_{EFF}$$

In the equation for U_{EFF} , U_{Cond} is the actual condensing film coefficient and W and A are the water mass flow rate and the condenser surface area, respectively. The effective film coefficients account for the temperature rise in the water as it passes through the condenser.

8. LABORATORY TESTING AND EXPERIMENTAL VERIFICATION

The primary purposes for the laboratory testing of the directly coupled solar thermal water pump were the experimental verifications of the analytic model and computer simulation, and the proof-of-concept testing of the components. The two systems selected for testing were the slow-stroke, constant-pressure system and the rapid-stroke, "fluid flywheel" concept. Problems in the operation of the systems or components uncovered at this stage enable a more reliable design to be generated and reduce the "debugging" problems associated with a design prototype. Verification of boiling and condensing film coefficients, as well as performance correlations, were sought.

8.1 TEST APPARATUS

The test setup for the constant-pressure pump is shown schematically in Figure 8.1. The working fluid, which was pentane, was vaporized in a horizontal collector while connected to both the condenser and the pumping chamber. At the end of the pumping stroke, condensing water passes through the condenser until the pumping chamber is refilled with water. Condenser water flow is then terminated and condensed pentane is released from the storage tank to the collector.

The four photographs, Figures 8.2 through 8.5, show the actual apparatus utilized. Figure 8.2 shows the constant-pressure, directly coupled solar water pump. The rack of instrumentation on the left is used to supply and monitor the electric heating elements attached to the collector absorber plates. The two tall cylinders in Figure 8.3 are the pumping chambers, which can be operated in parallel or sequentially. The condenser is above the pumping chambers, and the small tank slightly above the collector is the pentane reservoir. In Figure 8.4, the system's water heater and pumped-water holding tank are directly to the left of the instrumentation rack. The water heater is used to prevent freezing of the water if the need arises. Pumped water is held in the holding tank until it is returned to the pumping chambers during the condensation phase.

Figure 8.1 Schematic of Experimental Constant-Pressure Pumping System

Figure 8.2 Directly Coupled Solar Thermal Water Pump - Front View

Figure 8.3 Pumping Chambers, Condenser and Pentane Reservoir

Figure 8.4 Rear View of System

Figure 3.5 Pressure Regulators

The lift and delivery heads are independently adjustable by means of the regulators shown in Figure 8.5. This arrangement allows a continual recycling of the pumped water with the ability to simulate a wide range of pumping conditions. As a result of the difficulties with utilizing a rolling diaphragm without a unidirectional pressure gradient to ensure proper diaphragm behavior, the system was tested without a diaphragm. The wood piston was kept in the cylinder to act as an insulator between the water and the working fluid. A layer of pentane, which is not a good thermal conductor, was used around and over the top of the piston to further reduce heat losses to the water.

The test setup for the "fluid flywheel" pump is shown schematically in Figure 8.6. The working fluid, which is again pentane, is vaporized in the tilted collector, and pressurized vapor is stored in the storage tank above the collector until the cycle is initiated. During the first phase of the pumping cycle, the vapor storage tank is connected to the pumping chamber, and the condenser is isolated from the system. In the second phase, the pumping chamber is isolated from the vapor storage tank, and the kinetic energy stored in the moving water expands the vapor in the pumping chamber to a pressure level below the pumping head. In the third phase, the pumping chamber is connected to the condenser, and water is allowed to reenter the pumping chamber. Condensed pentane is then reintroduced into the collector by a pair of valves and an intermediate storage tank.

The photographs in Figures 8.7 through 8.9 show the actual apparatus used for the "fluid flywheel" pump. Figure 8.7 shows the covered tilted collector, heater controls, vapor reservoir above the collector, vertical pumping chamber, and condenser above the pumping chamber. Figure 8.8 shows the pumping chamber, piping run, and receiving chamber on top of the scaffolding. A detailed photograph of the low-pressure-drop check valve can be seen in Figure 8.9. A diaphragm to separate the water from the working fluid was not utilized but the wood piston was included to reduce the interaction between the vapor jet and the free surface of the water.

8-8

Figure 8.7 "Fluid Flywheel" Pumping Chamber and Collector

Figure 8.8 "Fluid Flywheel" Delivery Piping and Receiver

Figure 8.9 "Fluid Flywheel" Low-Pressure Drop Check Valve

8.2 RESULTS

The constant-pressure system with a horizontal collector was operated in a continuously cycling mode in order to obtain performance data on pumping efficiency. The pumping efficiency of the system was close to the value predicted by the simulation model, but the actual operation of the system differed from the simulated operation in several significant ways. The most probable causes of the operating differences originate from the degradation of the heat transfer coefficient during condensation and the change in heat transfer coefficient during boiling.

The condensing film coefficient was degraded by the presence of non-condensable vapor (probably air) that accumulated around the condenser tubes. Tests of the condenser under varying conditions, which are discussed later in this section, confirm this mechanism. The boiling film coefficient is disguised, in the case of a horizontal boiler, by the non-uniform and unsteady flow resulting from the two-phase flow conditions. The net boiling heat transfer coefficient appears to be fairly constant, but differences in temperature between collector tubes as large as 10°C have been observed. The horizontal collector position allows the collector to "boil dry" while maintaining good fluid distribution. This was a desired feature of the simple single-acting, constant-pressure system, but it is not required for the constant pressure system with collector valving. Tests of the tilted collector with natural convection circulation do not show the flow nonuniformity or degraded boiling coefficient that was observed for the horizontal collector.

The major operational differences between the constant pressure test system and the simulated model were a significantly longer time required for condensing (filling) and a significantly shorter time required for pumping in the real system. The longer condensing time results in an increase in energy stored as sensible heat in the dry collector and, therefore, a reduced pumping time when the liquid is returned to the collector. The pressure during pumping was not constant because of the relatively higher pumping rate

at the start of the cycle and the inability of the pressure regulator to maintain a constant pressure with this elevated flow rate. Pressure-volume graphs of two sample test runs are shown in Figure 8.10.

The condenser and horizontal collector were tested under steady-state conditions by operating the collector and condenser simultaneously. The overall heat transfer coefficient for the condenser was calculated by measuring the inlet and outlet temperatures of the condensing water, the water flow rate, the heat input, and the vapor temperature. The boiling heat transfer coefficient was similarly calculated by measuring the collector tube temperature, the heat input, and the vapor temperature. The testing procedure consisted of first isolating the pumping chambers from the remainder of the system, evacuating the system, then filling with liquid pentane. Even with a leaktight system, the vapor pressure differed from the saturation pressure expected with a corresponding vapor temperature. The suspected source for the noncondensable vapor (probably air) was the pentane itself, since the excess pressure was an approximately linear function of the amount of liquid pentane with which the system was filled (see Table 8.1 for the data). Measurements taken after the system had been connected with the pumping chamber and cycled show a rise in the noncondensable vapor pressure. The overall condensing film coefficient was related to the amount of noncondensable vapor, as shown in Table 8.2.

The boiling film coefficient for the horizontal collector was calculated for the horizontal collector with the assumption that the tubes were half filled with liquid. The measured and calculated temperature differences for one tube were within 1°F (out of a 20°F temperature change) at a thermal input of 1500 W less collector losses; but another tube differed in temperature by nearly 20°F, showing the nonuniformity of fluid flow within the collector. The measurements and theory differed significantly as the heat input was increased to 3000 W. The overall correlation was too unpredictable to be useful.

TOTAL TIME = 23 MIN
 $\eta = 0.0054$
 $I = 0.95$ SJN

TOTAL TIME = 23 MIN
 $\eta = 0.0048$
 $I = 0.95$ SUN

Figure 8.10 Constant Pressure Pumping Cycle

TABLE 8.1
NONCONDENSIBLE VAPOR PRESSURE

Vapor Temperature (°C)	Equivalent Saturation Pressure (psia)	Vapor Pressure (psia)	Difference (psia)	Volume Pentane (liters)	Ratio
<u>Before Cycling</u>					
16.7	7.0	9.82	2.82	3.5	0.81
14.1	6.2	7.08	0.88	1.0	0.83 avg.
14.8	6.4	7.17	0.77	1.0	
<u>After Cycling</u>					
18.0	7.4	12.36	4.96	3.5	
18.2	7.46	12.45	4.99	3.5	

TABLE 8.2
RELATIONSHIP OF CONDENSING FILM COEFFICIENT
NONCONDENSIBLE VAPOR

Noncondensible Vapor Pressure (psi)	Overall Condensing Film Coefficient (Btu/hr-ft ² -°F)
<u>Before Cycling</u>	
0.83	338
2.82	177
<u>After Cycling</u>	
4.97	168

The boiling tests for the tilted collector with natural convection flow gave consistent readings between tubes, and the temperature rise between the wall and the fluid was consistently lower than the value predicted by pool boiling correlations. This was expected since the rising vapor disturbs the boundary layer, thereby improving the heat transfer. Data for various conditions are shown in Table 8.3.

Air was used as the working fluid in the "fluid flywheel" pump in order to evaluate the pump behavior with a well-behaved single-phase fluid. Data for the actual tests and for simulation runs are shown in Table 8.4. Actual and simulated pump dynamics as a function of time are shown in Figures 8.11 and 8.12, respectively. The simulated results show excellent agreement with test results, with one exception - the time scale is diminished by a factor of $\sqrt{2}$, which could relate to a simulation error, possibly in the pipe impedance, of a factor of 2. This error had not been located at the time of writing, but the simulation correlation on the more important variables is very good. The actual water displacement at valve closure in the test runs was difficult to determine because of signal noise, but the values given in the table are considered to be within the tolerance shown. The response times for the tests attempt to account for the time during which dead volume in the pump chamber is filled.

Operation of the "fluid flywheel" pump with pentane as the working fluid resulted in significant differences from the air-operated case. The major difference was the heat transfer from the pentane vapor to the free surface of the water. The magnitude of heat transfer, although it was originally thought to be small because of the very short time interval, was exceedingly large because of the mixing of either (or both) the pentane vapor in the water or the water droplets in the pentane. The result was sufficient heat transfer surface area to quench the incoming pentane vapor.

This problem would not be present if the water and pentane were separated by a diaphragm but, since one was not operational, the solution for the experimental apparatus was to lower the initial level of the water

TABLE 8.3
HEAT TRANSFER DURING BOILING
FOR A TILTED COLLECTOR

Electrical Power Input: 76.9 W/ft²*

Thermal Input: $(262 - 1.755 \frac{\Delta T}{^\circ\text{C}})$ Btu/hr-ft²*

Tube Wall Temperature (°C)		Vapor Temperature (°C)	Average Temperature Difference (°C)	Equivalent Pressure (psia)	Predicted Temperature Difference (°C)
A	B				
<u>Quasi-steady</u>					
68.3	68.9	65.1	3.5	36.2	4.7
69.6	69.6	66.2	3.4	37.4	4.6
70.9	70.9	67.6	3.3	38.8	4.5
72.1	72.2	68.7	3.45	40.0	4.4
73.7	72.8	69.5	3.75	40.8	4.3
<u>Transient</u>					
36.0	34.7	30.5	4.85	12.2	8.4
50.4	49.6	45.6	4.4	20.4	6.5
64.5	62.5	59.2	4.3	30.7	5.2

*Based upon collector area; collector area = 5 x tube area.

TABLE 8.4
DATA FOR "FLUID FLYWHEEL" PUMP UTILIZING AIR

Data Source	At Valve Closure					At End of Cycle		
	Initial Pressure (psia)	Reservoir Pressure (psia)	Pumping Chamber Pressure (psia)	Displacement (ft)	Time (sec)	Pumping Chamber Pressure (psia)	Displacement (ft)	Time (sec)
Test Results	45.7	33.5	30.7	0.325	0.82	11.7	1.1	2.4
	65.7	47.7	39.2	0.375	0.7	12.7	1.7	2.5
Tolerance	±1.0	±1.0	±1.0	±0.05	±0.05		±0.025	
Simulation	45.7	34.6	31.5	0.325	0.59	10.4	1.1	1.8
	65.7	47.2	39.0	0.375	0.47	10.8	1.65	2.0
Simulation with 2x Pipe Impedence	45.7	35.1	32.1	0.325	0.83	10.5	1.1	2.6
	65.7	47.5	39.3	0.375	0.68	11.1	1.65	2.8

Delivery Pressure: 19.53 and 19.8 psia

Pumped Volume: $0.833 \text{ ft}^3/\text{ft displacement}$

Figure 8.11 Actual Dynamic Response for an Air-Powered "Fluic Flywheel" Pump

Figure 8.12 Simulation Dynamic Response for an Air-Powered "Fluid Flywheel" Pump

and piston in the pumping chamber. Approximately 1.5 to 2.0 liters of water were removed to reduce the magnitude of this effect but, in doing so, a dead volume is created which introduces some system inefficiency. The general effect of this dead volume on system operation can be seen by examining the dynamic response of the pump shown in Figure 8.13. The reservoir pressure drops quickly in the first 100 msec, and the pumping chamber pressure rises with a longer time constant.

Some liquid pentane entered the pumping chamber during the cycle. Two sources of the liquid were identified and both can be eliminated in a prototype design. One source was the condensate that accumulated next to the unheated valves. The other was liquid that was entrained by the pentane vapor as the bulk liquid flashes to vapor during the pumping cycle. The valve source can be eliminated by maintaining the valves at an elevated temperature and allowing any condensate to return to the collector. The entrained liquid can be recovered by employing baffles or other conventional techniques for liquid separation.

A modified version of the simulation program was prepared to account for some of the operational changes encountered. The results for a simulated run are shown in Figure 8.14, and they compare reasonably well with the actual performance data. Table 8.5 compares the data for the actual and simulated runs. The higher pressure at the end of the actual run is felt to be due to the presence of liquid pentane, which will produce additional vapor. Calculations of the mass of pentane in the pumping chamber, made by assuming only vapor present, indicate an increase in mass from the time of valve closure to the end of the cycle.

Figure 8.13 Actual Dynamic Response of a Pentane Powered "Fluid Flywheel" Pump

Figure 8.14 Simulated Dynamic Response of a Pentane Powered "Fluid Flywheel" Pump

TABLE 8.5
DATA FOR "FLUID FLYWHEEL" PUMP UTILIZING PENTANE

Data Source	Cycle Point	Reservoir Pressure (psia)	Pumping Chamber Pressure (psia)	Displacement (ft)	Time (sec)
Test Results	Initial	43.7	16.5	0	0
	At Valve Closure	34.7	27.8	0.35	1.15
	At End of Cycle	38.5	15.5	0.85	2.7
Tolerance		±1.0	±1.0	±0.05	±0.05
Simulation (2x Pipe Impedence)	Initial	42.0*	34.5	0	0
	At Valve Closure	36.3	28.0	0.35	1.16
	At End of Cycle	38.0	12.0	0.95	2.84

* A lower valve was required due to simulation characteristics.

9. FABRICATION COSTS FOR ACTUATORS AND VALVES

The fabrication costs will depend on both the materials selected and the quantity to be produced. A total production quantity of 10,000 units is assumed. All tooling and fixtures are amortized over this number of units. The life expectancy of the actuators will be dependent on the choice of materials, and we will be looking for a minimum service life of 10 years for all parts except the elastomeric diaphragms, which will probably need to be replaced annually.

The useable life of the pump actuator housing will depend on the corrosion rate of the waterside housing parts. It is planned to make them of stainless steel to ensure suitable service life. The water-side housing parts are relatively simple and can be stampings, which will lower parts costs in spite of the high cost of stainless steel. It would be reasonable to make the gas-side housings out of carbon steel, because the corrosion rate of these parts will be quite low. Water certainly will be in contact occasionally with the gas-side actuator housing, but will not be continuously pumped through, as in the case of the water-side housings. For the same reasons given above, the piston, shaft, and follower rod assemblies will be made of carbon steel.

The control valve assemblies will be fabricated of aluminum and, because the gas being used has poor lubricating properties, the parts will be hardcoated and Teflon impregnated to ensure long life and low sliding friction characteristics. This method of spool valve construction has been well proven in industry for pneumatic service and should be equally useful for pentane vapor service.

9.1 DEEP-WELL ACTUATOR

The deep-well actuator does not pump water directly, so carbon steel can be used to make all housing parts, and corrosion will not be a problem. The gas-side housing can be made by the deep drawing process while the base housing shell can be fabricated from sheet or tubing. The piston

and shaft assembly can be made by welding a deep-drawn piston to the shaft, and the diaphragm retainer can be a stamping. It is planned to make the base plate of steel plate and to fabricate the guide assembly from tubing. Approximate costs for these parts, assuming quantities of 10,000 parts are as follows:

Gas-Side Housing	\$10.00
Base Housing	10.00
Piston & Shaft Assembly	9.00
Bladder Retainer	1.00
Base Plate	9.00
Guide Assembly	12.00
<hr/>	
Total Cost for Mechanical Parts	\$51.00

Adding to this the bladder cost of \$15 and about \$9 for assembly and fasteners, the total actuator cost is approximately \$75.

9.2 SHALLOW-WELL ACTUATORS

Referring to the shallow-well single-diaphragm actuator design shown in Figure 6.10, it can be seen that one housing shell is in direct and continuous contact with water and, therefore, should be made of stainless steel to avoid corrosion. This part can be a deep-drawn part as can the steel gas-side housing, with the bladder clamp plates both made of stainless steel. Cost estimates for the single-diaphragm shallow-well actuator are:

Gas-Side Housing	\$10.00
Water-Side Housing	15.00
Diaphragm Clamp Plates	2.00
Check Valves	20.00
<hr/>	
Total Cost for Mechanical Parts	\$47.00

Adding to this the diaphragm cost of \$15 and an assembly and fastener cost of \$9, the total actuator cost will be \$71.

The double-acting dual-diaphragm shallow-well pump actuator is essentially two single-diaphragm actuators connected by the control valve housing. Its cost, therefore, will be double the cost of the single-diaphragm actuator plus the cost of the connecting housing. The connecting housing will contain the control valve assembly and can be fabricated from steel tubing with welded-on mounting flanges. An approximate cost for this assembly is \$25 which, when added to twice the single-diaphragm actuator cost (\$142), brings the cost of the double-acting actuator to \$167.

9.3 CONTROL VALVES

The control valve costs for the pump actuators will be based on the selling price of commercially available valves of similar size and complexity. An estimate will be made using one half of the valve's selling price as the valve cost.

VALVE	APPROXIMATE COST
Constant-Pressure Single-Action Valve	\$ 55.00
"Fluid Flywheel" Single-Action Valve	95.00
Constant-Pressure Double-Action Valve	75.00
"Fluid Flywheel" Double-Action Valve	130.00
Sequence Valve	30.00
Boiler Over-Pressure Valve	25.00

9.4 BOILER FEEDPUMP SYSTEM

The cost of the jet pump boiler feed system will include a tank, a float valve, and an injector. The costs for the tank and valve are estimated; the injector pump cost is based on a commercially available unit.

Tank	\$ 25.00
Float Valve	35.00
Injector	100.00

A mechanical feedpump, although less efficient, would cost \$20 and would not require a tank and float valve. If a pump efficiency of 50 percent is assumed for a mechanical pump, then the change in performance is only 3 percent.

9.5 SOLAR COLLECTOR AND VAPOR RESERVOIR

The solar collector used for the boiler of the water pumping systems will cost about \$12 per square foot. The efficiency of the constant pressure system is estimated to be 2.5 percent, which gives an equivalent collector cost of \$5/W. The "fluid flywheel" efficiency is estimated to be 8 percent, giving an equivalent collector cost of \$1.60/W.

9.6 CONDENSER

The cost of the condenser was obtained by first sizing a unit, then determining the pricing on standard shell-and-tube heat exchangers. The condenser was sized by the maximum load expected from a unit capable of delivering the low-head pumping requirement described in Chapter 10. The smallest standard size unit had four times the capacity required, and its cost would be approximately \$60. Since the condenser scales with the collector size, a cost adder of \$0.50/ft² will be included with the collector cost.

TOTAL SYSTEM COST

System Type	Single-Acting Actuator	Double-Acting* Actuator
Constant Pressure	** \$75.00 + \$5.40/W Plus	
Shallow Well	\$126.00	\$242.00
Deep Well	170.00	250.00
"Fluid Flywheel"	** \$75.00 + \$1.70/W Plus	
Shallow Well	\$165.00	\$297.00
Deep Well	170.00	305.00

*The double-acting actuators have double the pumping capacity of the single-acting actuators.

**A mechanical boiler feedpump has been assumed.

10. SYSTEM STUDIES

Three solar thermal pumping systems have been described in this report. Two of these, the constant pressure pump and the "fluid flywheel" pump, utilize the directly coupled pumping concepts. The third concept, which makes use of a nonlinear mechanical linkage, is an interesting and viable method to utilize solar thermal energy; however, it was not thoroughly investigated because of time limitations. The two directly coupled concepts differ mainly in the utilization of the expansion work of the working fluid. Both directly coupled concepts can employ similar actuators. A relatively rapid stroke can be achieved for the constant-pressure system by applying a slight over pressure, set by a sequence valve; by this method, thermal losses can be reduced. The "fluid flywheel" system is capable of achieving substantially higher system efficiencies than is the constant pressure system, and was therefore the main subject of the design tradeoff studies.

The "fluid flywheel" concept allows higher collector temperatures and pressures to be utilized for pumping even low heads; therefore, the system analysis included a double-glazed collector with a selectively coated absorber for its improved high-temperature efficiency. Performance with a concentrating collector can be extrapolated from the flat panel data. The collector configuration is identical to the singly glazed Thermo Electron collector, except for the addition of an inner glazing of FEP Teflon film. The Teflon film has a low refractive index, which reduces reflection losses at the air/dielectric interfaces. The correlation for the collector performance is

$$\eta_{\text{collector}} = 0.8 - 3.1 \frac{\Delta T}{I} \frac{\text{W/M}^2}{^\circ\text{C}}$$

This correlation is consistent with data obtained from Thermo Electron collectors.

10.1 DESIGN TRADEOFF STUDY OF THE "FLUID FLYWHEEL" PUMP

Pentane is the modeled working fluid for an idealized thermodynamic cycle that approaches a Carnot cycle. The process is an isothermal expansion with heat addition (boiling), followed by two adiabatic expansions; then an isothermal compression with heat extraction (condensing), followed by the adiabatic compression (working fluid pumping), which is provided by a jet pump driven from the collector reservoir. The jet pump itself is not adiabatic, but the total system can be considered adiabatic during the process. The actual simulation model does not assume ideal cycle conditions, but allows the process to be self-driven except at transition points. Simplified equations of state for pentane were used in the simulation; therefore, some error (which has not been quantified) in performance calculations can be expected in the simulation results.

A general analysis of the "fluid flywheel" system, in which flow losses were neglected, was performed; the results are presented in Figure 10.1. The delivery pressure was set at 43.3 psia and the inlet pressure was 10 psia, which corresponds to a condensing temperature of 77°F or 25°C. The 10-psia condensing pressure is used throughout the system study. Three insulation cases are plotted, demonstrating a broad maximum in system efficiency at intermediate levels. System operation near 225°F provides a good efficiency level over a variety of operating conditions and does not introduce extraordinary requirements for the system's components.

The system efficiency for both an ideal, frictionless system, and for a practical system with flow losses, has been determined as a function of delivery pressure. The data shown in Figure 10.2 indicate that a frictionless system operating at 225°F (102 psia) would have an efficiency peak near 32 psia delivery pressure, but a high level of efficiency in the range of 20 to 60 psia. The efficiency peaks near 32 psia because, with an initial pressure of about 100 psia, the pressure at the end of the expansion is approximately 10 psia, the condensing pressure. At higher

Figure 10.1 Ideal System and Collector Efficiency vs. Temperature as a Function of Insolation

Figure 10.2 System Efficiency vs. Delivery Pressure

delivery pressures, the pressure at the end of the cycle is greater than 10 psia, indicating remaining work that could be extracted. At lower delivery pressures, the end-of-cycle pressure is below 10 psia, resulting in negative work.

The negative work that is calculated during the simulation is a result of the simulation model. The model does not allow for simultaneous flow in both the inlet and discharge pipes, so the subinlet pressure that is present during the expansion cannot result in the acceleration of water in the inlet pipe. If the model allowed this condition, efficiency would not decrease as the discharge pressure decreases below the level required to reach 10 psia, but would remain constant.

The practical system includes the flow losses associated with pipe fittings, check valve, and pipe length, as well as entrance and exit losses due to the nozzle and diffuser. The efficiency peak for the practical system occurs at a higher delivery pressure if a constant volume of water is delivered per stroke. Again the efficiency remains high over a broad range from 20 to 60 psia. The efficiency peak is shifted to a higher delivery pressure because, at the higher pressure, less energy must be stored in the form of fluid kinetic energy. A lower level of energy translates into lower flow losses and, therefore, a more efficient system.

Figure 10.3 shows the effect of delivery-pipe diameter on system efficiency. The inertia increases as the diameter is squared; therefore, the velocity is an inverse function of diameter. Pressure drop is a function of velocity squared, length-to-diameter ratio, and a weak power of the Reynolds number; therefore, the pressure drop decreases approximately as the diameter to the 2.8 power. But, as the efficiency increases, so does the energy storage requirement, and the resulting power law factor is about 2.2.

The results in Figure 11.3 are for a pumping chamber with a displacement of about 3 gal or 0.4 ft³. The displacement cycle time for a chamber of this size is about 1.4 sec. This time, plus the time to condense and

Figure 10.3 System Efficiency vs. Delivery Pipe Diameter

refill the chamber, results in the minimum cycle time or maximum cycle frequency. The displacement cycle time does not vary significantly with pumping chamber volume if the delivery-pipe diameter is increased, as the pumping chamber volume is increased, to maintain a constant system efficiency. A longer length of pipe, for a given diameter, does not have a major effect on system efficiency, but a longer pipe generally results in higher efficiency. Two principles are operating. The first is that the mass increases as the pipe length increases; therefore, the fluid velocity decreases by the square root of pipe length. The effect on pressure drop is then only the Reynold's number effect, which causes an increase in pressure drop by the $1/10$ power of the pipe length. The second effect is that pressure losses associated with the valves and fittings become a smaller fraction of the total impedance. Since valve and fitting impedance is primarily resistive, whereas pipe-length impedance is both resistive and inductive, the net effect is to improve the quality of the fluid impedance (i.e., the ratio of inductive to resistive impedance), thereby reducing losses. This later effect is the predominant one.

A "fluid flywheel" pump does not require a large pipe diameter for the entire length of a long pipeline. Once sufficient pipe length has been provided to give a good impedance quality, the large-diameter pipe can be terminated by a holding tank or compliance chamber (accumulator) before transition to the smaller pipe diameter required by the steady-state flow requirement. The utilization of the pipeline can be improved if multiple pumping chambers are employed. This is necessary because the chamber filling time will typically be longer than the discharge time, and therefore the duty cycle of a single chamber system will be low. For a larger system, multiple chambers that are properly sequenced will result in a quasi-steady water flow.

The reservoir specific volume has an effect on system performance because it changes the location of the initial fluid expansion in the

two-phase region. Higher specific volumes result in higher efficiencies, but sufficient working fluid mass must be present to allow for losses and inventory in the condenser and pumping chamber. Table 10.1 shows the effect of specific volume on efficiency. Reservoir volume does not directly affect system efficiency, but the minimum volume is set by the mass of working fluid required and the desired specific volume.

The inner pumping chamber walls of a "fluid flywheel" pump can be maintained at an elevated temperature by either a heat source, such as a jacket of hot working fluid, or by an insulating layer, such as an elastomer. The elevated wall temperature prevents condensation during the cycle. Because of the rapid cycle, condensation is not important unless the walls have a short thermal time constant and a high surface-area-to-volume ratio. Small pumping chambers with inner metal walls will require a heat source to prevent working fluid condensation.

A principal design concern in a "fluid flywheel" pump is the actuator valving. The valving must sequence the actuator properly with low flow losses, and recover from regions of operation that are outside the design range; e.g., actuator stall midstroke in the final expansion. The requirement for low valve flow losses is most critical when the pumping chamber must be isolated from the boiler. At this point, the flow rate is highest and, therefore, valve closing speed is critical. The flow rate sets the valve size. The valve size required is about 3/4 in. equivalent orifice diameter for the actuator modeled in the simulation. The closing time should be on the order of 10 to 30 msec, which is within the capability of conventional valve designs.

The nonlinear mechanisms, previously described, have some inherent system advantages over the "fluid flywheel", although they require additional mechanical complexity. The valving logic is identical to the "fluid flywheel" logic but the speed and flow requirements are reduced because of an improved impedance match with the load. With a close impedance match, flow velocities and therefore pipe size can approach the values

TABLE 10.1
RELATIONSHIP BETWEEN SYSTEM EFFICIENCY
AND WORKING FLUID SPECIFIC VOLUME

Initial Specific Volume	System Efficiency
0.4	0.091
0.2	0.076
0.1	0.070

* Temperature, 225°F

Delivery pressure, 58 psia

Pipe diameter, 3 in.

Insolation, 1 kW/m²

associated with a constant-flow system. An exact impedance match is not required since some energy can be stored as kinetic energy. A relaxation on impedance matching allows a simple mechanism, like the inverter linkage, to be employed.

10.2 DESIGN RECOMMENDATIONS

The directly coupled solar thermal water pump is probably most suitable for remote pumping applications with relatively small volumetric requirements. The hydraulic power range for these sites would be between 100 W and 4 kW. Larger volumetric requirements can be met either by using multiple units or by accepting a shorter maintenance interval on the actuator diaphragm. The design recommendations are oriented toward unattended operation during the irrigation season. Three basic designs are recommended, one design for shallow wells or surface water pumping, and two for deep well operation.

A twin diaphragm double-acting pump actuator, (as described in Section 6.3.2.2), using the "fluid flywheel" principle is recommended for low suction lift applications. The use of the double-acting actuator eliminates the possible mismatch between achievable condenser temperature and actual suction pressure. If the condenser temperature and, therefore, pressure is below the required suction pressure the net force is available to assist the discharge side. Conversely, if the condenser pressure is above the required suction pressure the discharge side will assist thereby providing the additional force. A deep-well type actuator, with a nonlinear mechanical linkage, could be used in this application, but the added mechanical complexity is not required. The substitution of Freon 113 for n-Pentane as the working fluid would allow additional suction at the same condenser temperature, but this is not a necessary substitution.

The recommended design for a deep-well application would utilize the deep-well actuator as described in Section 6.3.1. The actuator can be used

directly in a "fluid flywheel" mode or indirectly with a nonlinear linkage to reduce flow losses. Either system can use the mechanical advantage of a lever to match the load requirements to the actuator. The rolling diaphragm of the deep-well actuator requires a unidirectional pressure gradient to ensure proper diaphragm operation. Premature diaphragm failure usually results when pressure reversals are present. The requirement for no pressure reversals can be adequately met for most applications by replacing the working fluid with one whose boiling point at atmospheric pressure occurs at a temperature below the expected ambient range. Both n-Butane and Freon 114, with boiling points at 31.1°F and 38.8°F, respectively, can be used. Their critical temperatures are sufficiently above the expected operating temperature of the collector so that the thermodynamic properties will not appreciably affect the operation of the system. A switch to such a working fluid will result in condenser pressures above atmospheric pressure; therefore, the pump must either be operated with a spring or counterweight for a single-acting actuator, or a double-acting actuator must be used. The double-acting actuator would consist of two single-acting actuators and deep-well pumps coupled together to operate out of phase. The double-acting system in a deep-well application has the advantage, as in the shallow-well system, of having a condenser temperature which is not critical to system operation. The deep-well system will typically have a larger condensing load than a comparable volume shallow-well system because of the increased delivery head requirement. A time constraint on condensing is imposed as the power output increases, since the time for the collector pressure to recover then decreases.

The collector recommended for remote applications is a double-glazed, flat-panel collector with a selective absorbing surface, or a moderate temperature concentrating collector employing a low concentration ratio. Either collector can be erected and left unattended during the irrigation season.

A mechanically driven feedpump, rather than a jet pump, is recommended for small or medium size systems because of the high estimated price differential between the two types of pumps. The performance improvement would warrant the higher cost jet pump in a system size over 2 kW.

10.3 COMPARISON OF WATER PUMP METHODS

Small capacity irrigation pumps have been chosen for this economic comparison because the directly coupled solar thermal water pump is intended for remote applications when the total volumetric requirements are relatively small. A recent study of remote area farms by Tabors (1978)* indicates that there is a large demand for irrigation on small one-hectare farms with a water lift of 1.5 to 4.5 meters. The yearly pumping requirement of 10,000 m³ of water per cultivated hectare is cited in the report. This is consistent with an average of one-eighth inch of water per day for a yearly requirement. If the irrigation demand during certain periods of the year, is twice the average demand, a pumping rate of approximately 60 gal/min would be required for a 4.5-hour equivalent solar day. A requirement for the same volume of water, but at a 30-meter head, is also considered in the analysis. Gasoline, diesel, wind, photovoltaic pumping systems, and the directly coupled solar thermal water pump are compared by using the present value method of analysis. Two rates for the value of capital, a general inflation rate, and a fuel escalation rate have been used in the present value calculations.

10.3.1 Engine Driven Pump

The cost of operating an engine-driven pump is usually dominated by the fuel cost; however, purchase price, maintenance cost, and replacement

* "The Economics of Water Lifting for Small Scale Irrigation in the Third World: Traditional and Photovoltaic Technologies," Richard D. Tabors, MIT Energy Laboratory Working Paper, No. MIT-EL-78-015, presented at the International Conference: Energy and Community Development, Athens, Greece, July 11, 1978.

cost are also significant considerations. The two system requirements are summarized below.

	Low-Head System	High-Head System
Daily capacity	61.3 m ³	61.3 m ³
Total lift	3 m	30 m
Yearly volume	10 ⁴ m ³	10 ⁴ m ³

Commercially available pumps, of this size range have efficiencies approaching 50 percent if they are well matched to the delivery conditions. It is common, however, to find pumps that operate at only 25 percent efficiency. For this analysis the pump will be considered well-designed and reasonably matched so that the efficiency is 40 percent for the high-head condition. At the low-head conditions, pipe losses will probably diminish this efficiency by about one-third, to 25 percent. The characteristics of two small, heavy duty gasoline and diesel engines typical of those used in pumping applications are given below:

	Gasoline	Diesel
Maximum Continuous Power	2.55 hp	3.3 hp
Specific Fuel Consumption (SFC) at full load	0.16 gal/hr/kW	0.125 gal/hr/kW
Full load pumping rate, $\eta_p = 50\%$	0.9 kW	1.0 kW

The engines have a higher specific consumption when they operate at the partial loads encountered at most installations.

A correlation for the normalized, specific fuel consumption of the gasoline engine is

$$NSFC = 0.4/f + 0.6$$

where f is the fraction of full load.

When the engines are running at an assumed two-thirds load with a 40 percent efficient pump, the actual fuel consumption per delivered hydraulic kW is 0.48 gal/hr/kW for the gasoline engine and 0.375 gal/hr/kW for the diesel engine. Operating at two-thirds load, the gasoline engine would operate for only 1.7 hr/day at the low-head condition and 10.6 hr/day at the high-head condition. The fuel consumption would be 0.4 gal and 2.4 gal for the two cases. Similarly, the diesel would operate 1.5 hr/day and 9.5 hr/day and consume 0.3 gal and 1.9 gal for the two cases. The yearly fuel and operating requirements are:

	Gasoline	Diesel
Low Head	70 gal 306 hr	54 gal 270 hr
High Head	440 gal 1980 hr	342 gal 1710 hr

Maintenance costs of engines and pumps in remote areas are difficult to estimate because of high labor and transportation charges, but a rough guess can be made. The pump seal should be replaced regularly at an estimated parts and labor cost of \$0.06/hr. The gasoline engine, which costs \$546 (including pump), has an estimated nonfuel operating cost of \$0.25 per hour and should be replaced after 3600 hours of operation. The diesel engine which costs \$1829 (including pump), should be rebuilt after 5000 hours of operation at a cost of \$500 and replaced after 10,000 hours of operation. Other nonfuel operating costs are estimated at \$0.04/hour.

10.3.2 Wind Powered Pump

The use of wind power for remote location water pumping is highly sensitive to site location. Although solar pumping is also site sensitive, the availability of winds during the irrigation season is probably more variable than the solar insolation. If only "good" wind sites are considered a comparison can be made. A "good" site is one that has a 15 mph typical

wind speed. This speed was selected because a constant 15 mph is equivalent, in daily energy, to a 4.5 hour solar day. It is also assumed that the wind at this "good" site blows during one-half of the average day.

Windmills have evolved considerably during the past decade, but little information could be obtained on the maintenance and durability of the newer machines. An example of one newer machine is the Sencenbaugh Model 500 which is rated at an electrical output of 500 W in a 24 mph wind. This unit, coupled with an electric motor and pump, could provide the output required by the low head requirement. The price of this unit, including the tower, motor, and pump, is \$4045. A mechanical coupling of the windmill and the pump would be a more efficient, although less flexible, alternative.

The older style, multivaned windmills are available and are designed to be coupled with a sucker rod pump. An 8-foot diameter machine would be capable of delivering our low-head requirement in the "good" location. Its cost, without the pump, is \$1,787. The high-head requirement can nearly be met by a 16-foot diameter unit at a cost of \$8,181 without the pump. The annual service requirements of the multivaned windmill pump are estimated to be 5 percent of the windmill's cost. This is equivalent to \$37 for the 8-foot machine and \$254 for the 16-foot machine. A pump service cost of \$20/year is also assumed.

10.3.3 Photovoltaic System

A comparable flat-panel photovoltaic system to pump 60 gal/min, using reasonable values for conversion and pumping efficiencies, would require approximately 290 W. At \$9/W panel cost, assuming the remaining components of the system at \$3/W, the total photovoltaic system would cost \$3,500. Prototypes of an operable system, suitable for remote locations, are currently available at \$7,500. The capacity of this unit is 240 W, and rated flow against a low discharge head is a similar 60 gal/min. The assumed estimate of \$3,500 for a production system is, therefore, a reasonable reflection of what a remote area photovoltaic pumping system could cost.

For the purposes of the economic analysis a 20-year lifetime for the flat panels has been assumed with no intervening maintenance required. The motor and pump would be replaced every five years at a cost of \$150 (\$300 for the high-head case) and the power supply replaced every ten years at a cost of \$190 (\$450 for the high-head case).

10.3.4 Directly Coupled Solar Thermal Water Pump

The costs associated with the directly coupled solar thermal water pump can be found in Chapter 9. The double-acting shallow-well actuator would undergo 1.5 million cycles/yr for the required output. This number of cycles is the approximate lifetime of the diaphragm; therefore, a yearly replacement would be necessary. The deep-well actuator would also require yearly replacement if used in a single action mode.

The actuator valves will probably require replacement after 10 million cycles, based upon experience with commercially available valves. Other maintenance costs would probably include periodic replenishment of the working fluid and replacement of seals in the deep-well pump.

A factor of 2 has been used with the system cost to generate the system retail selling price. Maintenance costs are estimated at \$60/yr for the low-head pump and \$50/yr for a double-acting, high-head pump.

10.3.5 Cost Comparisons

The alternative remote location irrigation systems were compared for cost by using a present value analysis. Capital and maintenance costs were escalated at an annual rate of 10 percent and the fuel cost escalated at an annual rate of 15 percent. Two rates for the value of capital, 11.5 percent and 15 percent were included. The lower rate is close to the prime lending rate and the higher rate represents a level consistent with the rate for higher risk investments. Figures 10.4 through 10.7 display the results in graphic form. The first cost for the low-head condition is lowest for a gasoline pumping system during the first four years of operation. After four years the directly coupled solar thermal system provides

Figure 10.4 Present Value of Water Pumping Systems
Low-Head Condition: 11.5% Annual Value of Capital

Figure 10.5 Present Value of Water Pumping Systems
Low-Head Condition: 15% Annual Value of Capital

Figure 10.6 Present Value of Water Pumping Systems
High-Head Condition: 11.5% Annual Value of Capital

Figure 10.7 Present Value of Water Pumping Systems
High-Head Condition: 15% Annual Value of Capital

the lowest cost water-pumping method. The initial cost of the directly coupled solar thermal system is high because the valves and actuators represent a high fraction of the system cost. The value of capital does not have a significant effect on the crossover time. For the high-head system, gasoline again has the lowest first cost, but it rises above the diesel system cost by the end of the second year. After four years, the directly coupled solar thermal system provides the lowest-cost pumping system, and the long-term costs of this system are significantly below the alternatives.

11. CONCLUSIONS AND PROGRAM RECOMMENDATIONS

The directly coupled solar thermal water pumping system has been analyzed for both performance and cost. Performance estimates for a real operating system exceed 8 percent for the typical environmental conditions expected. The cost of such a system, using currently available flat-panel collectors, is highly competitive with alternative, remote location, water pumping systems. Although the system has a fairly low relative cost, there are areas such as valve and actuator cost that have a high potential for reducing system cost, especially for smaller systems. The performance of the system can be improved by high-performance, moderate-temperature collectors such as evacuated glass tubes. The advantages of the directly coupled solar thermal system are simplicity and limited failure modes. Results of the experimental phase indicate that the analytic models provide an accurate estimate of achievable system performance.

The viability of the directly coupled solar water pumping concept requires the construction and testing of prototypic units. Because of the demonstrated potential for a low-cost, remote pumping system, a continuation of this program is recommended.

APPENDIX

NONLINEAR MECHANICAL LINKAGES

A nonlinear mechanical linkage can be synthesized to match the exact requirements of an expanding fluid by utilizing the model shown in Figure 5.22 and the following analysis:

$$(X_p - X_1)^2 + Y_p^2 = R_1^2$$

$$(X_2 - X_p)^2 + Y_p^2 = R_2^2$$

$$X_p - X_1 = [R_1^2 - f^2(x_p)]^{\frac{1}{2}}$$

$$\frac{dx_1}{dx_p} = 1 + [R_1^2 - f^2(x_p)]^{\frac{1}{2}} f(x_p) f'(x_p) \quad (1)$$

$$\frac{dx_2}{dx_p} = 1 - [R_2^2 - f^2(x_p)]^{\frac{1}{2}} f(x_p) f'(x_p) \quad (2)$$

$$dx_1 = MA(x_1) dx_2 \quad (3)$$

Combining Equations 1, 2, and 3 yields:

$$f'(x_p) = \frac{MA(x_1) - 1}{\left[\left(\frac{R_1}{f(x_p)} \right)^2 - 1 \right]^{-\frac{1}{2}} + \left[\left(\frac{R_2}{f(x_p)} \right)^2 - 1 \right]^{-\frac{1}{2}} MA(x_1)} \quad (4)$$

Therefore, given R_1 , R_2 , and $MA(x_1)$, along with starting points for x_1 and x_2 , a solution for $f(x_p)$ can be obtained by numerically integrating Equation 4. The mechanical advantage can be approximated, a priori, for a known expansion; some allowance for friction and errors must be provided to ensure a rapid and complete stroke.

A computer program has been written for an idealized expansion of Freon 113; a listing of the program and two runs have been included. The

input parameter lengths are normalized to the link length (R_2). Required parameters are the link length (R_1), the initial position of the point (x_2), the position of x_1 for the isentropic expansion, the initial specific volume, the final volume, the minimum or back pressure the system is working against, and the number of position increments desired. The outputs are positions of the linkages, angles of the linkages and track, maximum forces encountered, and mechanical advantage.

The invisor linkage analysis that follows relates the displacement of the colinear points at distances x_1 and x_2 to the volumes of the working fluid chamber (v) and the pump chamber (v_p). The initial x_1 and x_2 positions are x_{10} and x_{20} , and the volume at the end of the idealized isobaric expansion is v_o .

$$x_2 = \frac{A^2 - B^2}{x_1}$$

The following normalized variables are utilized:

$$\bar{x}_2 = \frac{x_2}{A} \quad \bar{x}_1 = \frac{x_1}{A} \quad \bar{v}_o = \frac{v}{v_o} \quad \bar{v}_p = \frac{v_p}{v_o}$$

$$R = B/A \quad \text{and} \quad 1 - R^2 = 1.$$

$$\bar{x}_2 = -\frac{\alpha}{\bar{x}_1}$$

$$\bar{v} = \bar{x}_1 - \bar{x}_{10} \quad \text{and} \quad \bar{v}_p = \bar{x}_2 - \bar{x}_{20}$$

Combining the above equations yields

$$\bar{v}_p = \frac{-\bar{x}}{\bar{v} + \bar{x}_{10}} - \bar{x}_{20}$$

at

$$\bar{v} = 0, \quad \bar{x}_1 = \bar{x}_{10} \quad \text{and} \quad \bar{v}_p = 0.$$

Therefore,

$$\bar{x}_{20} = \frac{-\alpha}{x_{10}} \text{ and } \bar{v}_p = \alpha \left(\frac{1}{\bar{x}_{10}} - \frac{1}{\bar{v} + \bar{x}_{10}} \right).$$

If the end points of the invisor are constrained to match the end points of the working fluid expansion, then a quadratic solution for $\bar{x}_{10}$ in terms of x can be obtained.

LIST

LINKAGE 12:03EDT 06/25/79

```

100 PROGRAM LINKAGE
1100
1200 FREON 113 IS THE WORKING FLUID
1300
140 WRITE(' ',FMT=10)
150 10 FORMAT('0PROGRAM TO CALCULATE LINKAGE POSITIONS',
160 & ' AND ANGLES'/'FOR AN EXPANDER/PUMP'/' )
170 WRITE(' ',FMT=20)
180 20 FORMAT(' INPUT THE FOLLOWING PARAMETERS: R1/R2, D0/R2, A*U0/R2,',
185 & ' U0, UMAX/U0, PMIN, N'/' )
190 INPUT, R1,D0,AU0,U0,RATIO,PMIN,N
200 R12=R1**2
201 UOE=U0**(-1.02)
202 C2=PMIN/34.0
203 C1=UOE-C2
204 RN=N
210 RD=57.296
220 WRITE(' ',FMT=30)
230 30 FORMAT(' X1/R2',T11,'X2/R2',T20,'XP/R2',T29,'YP/R2',T37,'ANG1',
240 & T45,'ANG2',T53,'ANG3',T61,'FMAX',T69,'MA'/' )
250 DO 100 I=0,N
260 RI=I
270 RAT=RI*(RATIO-1.0)/RN+1.0
275 ADU=C1/(RAT**(-1.02)*UOE-C2)
280 D=D0-AU0*(50.0*UOE/C1*(RAT**(-1.02)-1.0)+(1.0+C2/C1)*(RAT-1.0))
290 DP=(R12+D**2-1.0)/(2.0*D)
300 YP=SQRT(R12-DP**2)
310 X1=AU0*RAT
320 X2=X1+D
330 XP=X1+DP
340 ANG1=ASIN(YP/R1)*RD
350 ANG2=ACOS(ABS(D-DP)*RD)
353 ANG3=ATAN((ADU-1.0)/(ADU*YP/(ABS(D-DP))+YP/ABS(DP))))*RD
354 F1=ABS(R1/(ABS(DP)*ADU))
355 F2=ABS(1/(ABS(D-DP)))
356 FT=ABS((YP/(ABS(D-DP))+YP/(ABS(DP)*ADU))/COS(ANG3/RD))
357 FMAX=AMAX1(F1,F2,FT)
360 WRITE(' ',FMT=40)X1,X2,XP,YP,ANG1,ANG2,ANG3,FMAX,ADU
370 40 FORMAT(1X,4(F6.3,3X),5(F5.2,3X))
380 100 CONTINUE
385 PRINT
390 STOP
400 END

```

READY

RUN

LINKAGE 12:05EDT 06/25/79

PROGRAM TO CALCULATE LINKAGE POSITIONS AND ANGLES
OR AN EXPANDER/PUMP

INPUT THE FOLLOWING PARAMETERS: R1/R2, D0/R2, A*U0/R2, U0, UMAX/U0, PMIN, N

?1. -1.414 .5 .5705 3. 8. 10

X1/R2	X2/R2	XP/R2	YP/R2	ANG1	ANG2	ANG3	FMAX	MA
0.500	-0.914	-0.287	0.707	45.01	45.01	-0.00	2.00	1.00
0.600	-0.824	-0.112	0.702	44.59	44.59	6.28	1.79	1.24
0.700	-0.751	-0.026	0.688	43.48	43.48	11.98	1.61	1.50
0.800	-0.690	0.055	0.667	41.83	41.83	17.45	1.46	1.78
0.900	-0.638	0.131	0.639	39.72	39.72	22.92	1.34	2.08
1.000	-0.594	0.203	0.604	37.17	37.17	28.57	1.25	2.41
1.100	-0.555	0.273	0.562	34.17	34.17	34.56	1.21	2.76
1.200	-0.521	0.340	0.510	30.64	30.64	41.08	1.16	3.13
1.300	-0.491	0.405	0.445	26.44	26.44	48.40	1.12	3.55
1.400	-0.464	0.468	0.362	21.23	21.23	57.05	1.07	3.99
1.500	-0.441	0.530	0.242	14.00	14.00	68.57	1.03	4.49

PROGRAM STOP AT 390

RUN

LINKAGE 12:18EDT 06/25/79

PROGRAM TO CALCULATE LINKAGE POSITIONS AND ANGLES
OR AN EXPANDER/PUMP

INPUT THE FOLLOWING PARAMETERS: R1/R2, D0/R2, A*U0/R2, U0, UMAX/U0, PMIN, N

?1. 1.961 .5 .5705 3. 8. 10

X1/R2	X2/R2	XP/R2	YP/R2	ANG1	ANG2	ANG3	FMAX	MA
0.500	2.461	1.480	0.197	11.33	11.33	-0.00	1.02	1.00
0.600	2.551	1.575	0.221	12.76	12.76	25.59	1.03	1.24
0.700	2.624	1.662	0.274	15.87	15.87	35.28	1.04	1.50
0.800	2.685	1.742	0.335	19.54	19.54	38.40	1.06	1.78
0.900	2.737	1.818	0.396	23.32	23.32	39.18	1.09	2.08
1.000	2.781	1.891	0.455	27.04	27.04	38.97	1.12	2.41
1.100	2.820	1.960	0.510	30.67	30.67	38.24	1.16	2.76
1.200	2.854	2.027	0.562	34.20	34.20	37.22	1.21	3.13
1.300	2.884	2.092	0.610	37.62	37.62	36.01	1.26	3.55
1.400	2.911	2.155	0.655	40.94	40.94	34.65	1.32	3.99
1.500	2.934	2.217	0.697	44.18	44.18	33.18	1.42	4.49

PROGRAM STOP AT 390