A FEASIBILITY STUDY OF A SOLAR POWERED ELECTRICAL GENERATOR OPERATING ON A THREE PROCESS CYCLE

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ABSTRACT

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A design and feasibility study of an electrical generator, which derives its power from solar or industrial waste energy, is conducted. The generator operates on a three process cycle which consists of Isochoric heating, Isentropic expansion and Isobaric (Isothermal) cooling. The energy transformation of the generator is from thermal energy drawn from a heat source, into potential energy within a working fluid and then directly into electrical energy through the motion of a magnetic piston within a coil during the expansion process. The efficiency of the new cycle is comparable to that of a Rankine cycle, with no superheat, operating on the same working substance and between the same temperature limits. The construction of the electrical generator is simpler than that of a power plant operating on the Rankine cycle.

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LIST OF SYMBOLS

a - Acceleration of the piston, (m/s^2).

A - Area of the piston, (m^2)

b - Breadth of the condensing surface, (m)

B - Magnetic flux density, (Wb/m^2)

e - Voltage induced in the coil, (V)

F_e - Force on the piston due to the magnet-coil interaction, (N)

F_f - Force on the piston due to friction, (N)

F_p - Force on the piston due to a pressure differential, (N)

F_t - Total force on the piston, (N)

F_2 - Condensation factor based on the working fluid and the temperature at which condensation takes place

h - Specific enthalpy of the working fluid, (KJ/ Kg)

h_c - Heat transfer coefficient for condensation, ($KJ/s\text{-}m^2\text{-}{}^\circ K$)

H - Total enthalpy of the working fluid, (KJ)

ΔH_{12} - Heat added to the working fluid, (KJ)

ΔH_{12R} - Heat added to the working fluid in the Rankine cycle, (KJ)

ΔH_{31} - Heat rejected by the working fluid, (KJ)

I - Current induced in the coil, (A).

K - Friction Factor

- Length of one turn of the coil, (m)

m - Mass of the piston, (Kg)

$P(z)$ - Pressure acting on the piston (as a function of the displacement of the piston), (kN/m^2 , kPa)

s - Specific entropy of the working fluid, (KJ / Kg)

T	-	Temperature of the working fluid, (°K)
v	-	Specific volume of the working fluid, (m^3/ Kg)
V	-	Volume of the working fluid, (m^3)
w_l	-	Condensate flow rate over the condensing surface, (Kg/ s)
W	-	Mass of the working fluid, (Kg)
$\Delta\overline{W}$	-	Work done by the working fluid during the cycle, (KJ)
$\overline{W}_p$	-	Pump work done during the cycle, (KJ)
x	-	Quality of the working fluid
z	-	Displacement of the piston, (m)
Z	-	Number of turns of the coil
η	-	Efficiency of the cycle
ρ	-	Density of the working fluid, (Kg/ m^3)

Subscripts

c	-	Critical state of the working fluid, (Temperature, pressure and specific volume)
f13	-	Property of the working fluid at the intersection of the saturated liquid line and the line extended from the process line 1-3
g13	-	Property of the working fluid at the intersection of the saturated vapour line and the line extended from the process line 1-3
R	-	Rankine cycle
RP	-	Rankine cycle with the pump work included

v	- Vessel
1	- At point 1 of cycle
1R	- At point 1 of the Rankine cycle
2	- At point 2 of cycle
3	- At point 3 of cycle

INTRODUCTION

Till now, all vapour power plants operate on the Rankine cycle or its modified forms (regenerating or reheating). The machine operating on the Rankine cycle requires a minimum of four components: boiler, engine or turbine, condenser and pump. The initial costs of the machines for the Rankine cycle are high. An alternate method, now in operation, is the Minto Wheel (1). The Minto Wheel is a series of vessels arranged in a circle around a central hub. The vessels are connected by small tubes to the vessels on the opposite side of the circle. The working fluid is in the liquid state when the vessel containing it is rotated to the bottom of the circle. At this point, the vessel and the working fluid are heated, a part of the working fluid evaporates and expands, forcing the remaining liquid to pass through the tube to the vessel at the top of the circle. The assembly is now unstable with the mass concentrated at the top of the wheel, this will cause the rotation of the machine, creating useful work. The Minto Wheel is not comparable to the Rankine cycle in terms of efficiency but it can be operated on a small temperature differential.

The purpose of this report is to develope and examine the feasibility of a machine that is capable of transforming solar or industrial waste energy into useful work. In order to acheive this, a cycle and a machine to execute the cycle have been studied. This machine is referred to as the Free Piston Generator.(F.P.G.) because the piston of the machine, moving freely in a cylinder, is not physically connected to any outside mechanism and the machine generates electricity. The objectives of the study are to obtain a relatively good efficiency and a simple machine. This machine should be so designed that it can be fabricated with normal machine shop equipment.

The need for this type of equipment is shown by the present energy crisis and pollution. The use of solar and industrial waste energy will help reduce the acuteness of these problems.

The Rankine cycle is the standard vapour power cycle that is used for the comparison of the efficiency and the operation of the new cycle.

An important nature of the machine should be its simplicity, this will provide, not only, a low initial cost, but also, a low maintenance cost. The use of readily available materials can make the F.P.G. more acceptable. The simplicity of design is very conducive to the developement of a rugged piece of machinery. This quality of ruggedness serves to lower the operating costs of the machine and widens the range of applications of the F.P.G.. The equipment should also be versatile so that a greater number of possible applications can be found.

DESCRIPTION AND OPERATION OF THE MACHINE

The assembly of the Free Piston Generator (F.P.G.) is as shown in Figure 1. It is divided into three major sub-assemblies which are the piston-cylinder assembly, the valve assembly and the vessel assembly.

The piston-cylinder assembly consists of six major components as shown in Figure 2. The first of these is the piston which is fabricated from a highly magnetized material. The piston is equipped with seals that can withstand the maximum pressure of the process. It is also equipped with nylon slide bearings which provide low friction motion of the piston without the need for lubricants.

The next component of this assembly is the cylinder inner tube which is fabricated from a phenolic tube with a very thin wall, the bore must be honed to a fine surface finish. This material was chosen because of its high magnetic permeability and its high resistance to wear.

Around the inner tube is a coil made of copper wire, which is tightly wrapped around the phenolic tube. The coil is fixed in epoxy resin after the coil and the inner tube have been assembled in the outer shell.

The outer shell, which is the next component, is fabricated from a steel tube or pipe. This component is designed to strengthen the coil and inner tube. It also acts as a return path for the magnetic lines. This cylinder has its bottom end closed with a phenolic or stainless steel plate, which has a port providing mounting for the valve assembly. Also on the plate there are two insulated electrical contacts which control the operation of the valve. The other end of the cylinder has an opening, having a diameter smaller than that of the piston, so that it can provide a positive stop for the piston.

There is a bellows attached to the outer shell with a very good seal. It acts as an expansion compensator for the machine and is made from stainless steel. This material is selected for fatigue-proof operation and must be of a non-magnetic austenitic type of stainless steel. The positive stop on the outer shell prevents the piston from coming into contact with the bellows.

The final item in this assembly is a set of bellows guides which allow the bellows to expand and contract easily.

The second major sub-assembly is the valve assembly as shown in Figure 3. Valve a is the main valve, it controls the operation of the F.P.G.. It is a snap-over-centre design, pressure actuated for the opening of the valve and solenoid actuated for closing. The valve material could either be steel or phenolic. In the open position this valve will provide a large port area to give low resistance to the flow of the working fluid passing through it. In the closed position it effectively separates the piston-cylinder assembly from the vessel assembly. The valve has a slide which is fabricated from steel or stainless steel, and is equipped with seals and nylon bearings. There are two insulated electrical contacts in the valve body, which control the opening and closing of valves b and c which in turn control the heating and cooling processes. These two valves are located outside the valve assembly.

The third sub-assembly of the F.P.G. is the vessel assembly shown in Figure 4 as a hemispherical body which has two heat exchange surfaces.

The flat lower surface is used for pool boiling heat addition. If it is required this surface may be finned. The Hemispherical upper surface is used for heat rejection by film type condensation. Again this surface may be finned in order to increase the rate of heat transfer. This vessel is fabricated from mild or stainless steels and should be designed to withstand pressures 50 to 100% above the peak cycle pressure. There is also a safety valve which will blow out if the pressure exceeds the peak cycle pressure by 10 to 15%. This assembly has a large port which provides a mounting surface for the valve assembly.

The operation of the F.P.G. is a new closed non-flow cycle which consists of three processes as shown in Figure 5.

1-2 Isochoric heat addition

2-3 Isentropic expansion

3-1 Isobaric heat rejection

Conditions at the starting point 1 of the cycle are as follows:

- the piston is at the bottom of its stroke.
- valve a is closed which blocks flow from the vessel to the piston-cylinder assembly.
- valve b is opened which allows the hot liquid to heat the lower surface of the vessel.
- valve c is closed which blocks the cold liquid from the heat rejection surface of the vessel.

The fluid in the vessel is usually a refrigerant which is determined by the temperatures of the heating and cooling fluids.

The first process is an Isochoric heat addition because valve a is closed and the volume of the vessel cannot be changed. At the start of this process the working fluid is in the wet vapour region consisting of both liquid and vapour. As heat is added to the working fluid the temperature, enthalpy, quality and the pressure of the fluid increase. When the pressure reaches the value that is required to open valve a, the process ends. When the valve a opens, the slide of the valve touches the two insulated electrical contacts. This connection activates two relays, one causes the hot liquid valve b to close and the other causes the cold liquid valve c to open after a pre-set time delay.

The second process is Isentropic expansion. Valve a is now open, which allows the working fluid to pass from the vessel to the piston-cylinder assembly. This causes a pressure differential across the piston and produces an acceleration of the piston. The motion of the piston induces two forces, the first of these is a frictional force between the piston and the cylinder, which is in a direction opposing the motion of the piston. The second force is caused by the interaction of the coil and the magnetized moving piston, which tends to lower the velocity of the piston. The interaction between the coil and the piston causes an electrical current in the coil. The current is dependent on the velocity and the magnetic flux of the piston, as well as, the design of the coil. The voltage of the electrical output depends on the design of the coil. The electrical power output is the developed power of the F.P.G.. In the ideal case, the process ends when the pressure differential across the piston is zero.

The third process is the Isobaric heat rejection. It begins at the time when the cold liquid valve c is opened, which allows the cold liquid to flow to the upper heat exchange surface of the vessel. As the working fluid condenses on the heat exchange surface, its volume must decrease. This causes the piston to return. When the piston reaches the bottom of its stroke, it touches another set of two insulated electrical contacts. This electrical connection actuates a relay. The signal from this relay causes valve a to close, valve b, the hot liquid valve, to open and valve c, the cold liquid valve, to close. This is the end of the third process and it closes the cycle.

It should be pointed out that the F.P.G. consists of a few simple components, of these only two in the major assembly, the piston and the valve slide, are moving. The system is totally enclosed which minimizes the possibility of leakage. It is worthwhile to note that no ferromagnetic metals should be positioned at the ends of the cylinder. This is done in order to exclude the possibility of interaction of these components with the piston.

PERFORMANCE OF THE CYCLE AND MACHINE

1. Thermodynamic Relations

The operation and analysis of the F.P.G. machine is dependent on three factors:

- The working fluid
- The temperatures of the cycle
- The components of the machine

For the purpose of analysis, the cycle is considered to have:

- Isentropic expansion
- Maximum temperature and pressure at the critical point of the working fluid
- Minimum pressure of the working fluid to be atmospheric pressure

The cycle using Refrigerant 11 (Trichloroflouromethane) as the working fluid, given the above conditions, is shown in Figures 5 and 6, which are the T-s and P-v diagrams, respectively. Calculations for these figures were performed on a programmable calculator (TI-PROGRAMMABLE-59) as shown in Appendix A. The cycle shown in Figures 5 and 6 is as follows:

1-2 Isochoric heat addition

2-3 Isentropic expansion

3-1 Isobaric heat rejection

The sequence of the mechanism at each state of the cycle is:

1 Valve a closes

Valve b (hot liquid) opens

Valve c (cold liquid) closes

2 Valve a opens

Valve b (hot liquid) closes

3 Valve c (cold liquid) opens

Figure 7 is a graph of the thermal efficiency of the cycle with respect to the temperature of heat rejection using Refrigerant 11 as the working fluid. This form of graph is chosen because it shows clearly how the efficiency will change as the temperature of heat rejection is changed. This temperature can be raised by increasing the pressure at the end of the expansion stroke. It can be seen that the efficiency of the cycle increases as the heat rejection temperature is lowered. Efficiencies of this cycle below 24 °C have not been shown as the pressure of the working fluid would have to be below atmospheric pressure. Efficiency of this cycle is approximately 26 % at 24 °C and this is reduced to about 24 % when the heat rejection temperature is 38° C. Figure 8, which is similar in form to Figure 7, shows the efficiencies, for this cycle, of several fluids. In the analysis, Refrigerant 11 is chosen because water at 12° C is considered as the cooling fluid and because of its greater efficiency.

Given that all the pertinent data concerning the working fluid is available, the calculations, starting at point 2, will proceed as follows:

First the mass of the working fluid in the vessel is calculated:

$$W_2 = V_2 \rho_2 = W_1 = W_3 = W \qquad (1)$$

From this the total enthalpy of the fluid at point 2 can be calculated:

$$H_2 = h_2 W \quad (2)$$

At point 3 we know that the following relationship holds:

$$s_3 = s_{f13} + x_3 (s_{g13} - s_{f13}) \quad (3)$$

And because the expansion is considered to be isentropic:

$$s_3 = s_2 \quad (4)$$

By combining equations 3 and 4, it is found that the relationship for the quality of the fluid at point 3 is:

$$x_3 = \frac{s_2 - s_{f13}}{s_{g13} - s_{f13}} \quad (5)$$

From this the enthalpy at point 3 can be found, using the formula:

$$h_3 = h_{f13} + x_3 (h_{g13} - h_{f13}) \quad (6)$$

Therefore the total enthalpy at point 3 is:

$$H_3 = h_3 W \quad (7)$$

The volume of the working fluid after expansion is:

$$V_3 = W \left(v_{g13} x_3 + \frac{(1 - x_3)}{\rho_{f13}} \right) \quad (8)$$

Therefore the expansion ratio of the fluid is:

$$\frac{V_3}{V_1} = \frac{W \left(v_{g13} x_3 + \frac{(1 - x_3)}{\rho_{f13}} \right)}{V_1} \quad (9)$$

At point 1 it is known that the following relationship holds:

$$V_2 = V_1 = W \left(v_{g13} x_1 + \frac{(1 - x_1)}{\rho_{f13}} \right) \tag{10}$$

From this, the relationship for the quality at point 1 can be derived and it is:

$$x_1 = \frac{\frac{V_1}{W} - 1/\rho_{f13}}{v_{g13} - 1/\rho_{f13}} \tag{11}$$

Knowing the quality at this point, the entropy can be found as follows:

$$s_1 = s_{f13} + x_1 (s_{g13} - s_{f13}) \tag{12}$$

And the enthalpy can be found using the following relationship:

$$h_1 = h_{f13} + x_1 (h_{g13} - h_{f13}) \tag{13}$$

Therefore the total enthalpy of the system at point 1 is:

$$H_1 = h_1 W \tag{14}$$

From the previous calculations we can now find:

Heat added:

$$\Delta H_{12} = H_2 - H_1 \tag{15}$$

Heat rejected:

$$\Delta H_{31} = H_1 - H_3 \tag{16}$$

Work output:

$$\Delta \bar{W} = H_2 - H_3 \tag{17}$$

Knowing these values, the efficiency of the cycle can now be calculated using the relationship:

$$\eta = \frac{\Delta \bar{W}}{\Delta H_{12}} = \frac{H_2 - H_3}{H_2 - H_1} \tag{18}$$

As a guide, the following numerical calculations are submitted. The assumptions made are:

- Refrigerant 11 is the working fluid
- At point 2 the working fluid is at its critical conditions (4309.6 KiloPascals, 198° C)
- At points 1 and 3 the working fluid is at atmospheric pressure and 24° C
- The volume of the vessel is 0.0283 m^3 (.1 cubic foot)

Starting at point 2, the data is:

h_2 = 260.524 KJ/ Kg

s_2 = 0.40025 KJ/ Kg

V_2 = 0.0283 m^3

ρ_2 = 553.74 Kg/ m^3

The mass of the working fluid is:

$$W = V_2 \rho_2$$
$$= 0.0283 \times 553.74$$
$$= 15.680 \text{ Kg}$$

The total enthalpy at this point is:

$$H_2 = h_2 W$$
$$= 260.524 \times 15.680$$
$$= 4085.1 \ \text{KJ}$$

The properties at the intersections of the saturated liquid and the saturated vapour lines with the line extended from the process line 1-3 are:

$$v_{g13} = 0.17011 \ m^3/ \ Kg$$
$$\rho_{f13} = 1478.6 \ Kg/ \ m^3$$
$$h_{f13} = 54.597 \ KJ/ \ Kg$$
$$h_{g13} = 234.74 \ KJ/ \ Kg$$
$$s_{f13} = 0.11478 \ KJ/ \ Kg$$
$$s_{g13} = 0.45171 \ KJ/ \ Kg$$

At point 3 the quality of the working fluid is:

$$x_3 = \frac{s_2 - s_{f13}}{s_{g13} - s_{f13}}$$

$$= \frac{0.40025 - 0.11478}{0.45171 - 0.11478}$$

$$= 84.73 \ \%$$

The enthalpy at point 3 is:

$$h_3 = h_{f13} + x_3 \ (h_{g13} - h_{f13})$$
$$= 54.597 + 0.8473 \times (234.74 - 54.597)$$
$$= 207.23 \ KJ/ \ Kg$$

The total enthalpy at this point is:

$$H_3 = h_3 W$$
$$= 207.23 \times 15.680$$
$$= 3249.4 \ \text{KJ}$$

The volume of the working fluid at point 3 is:

$$V_3 = W \left(v_{g13}\, x_3 + \frac{(1 - x_3)}{\rho_{f13}} \right)$$

$$= 15.680 \times \left(0.17011 \times 0.8473 + \frac{(1 - 0.8473)}{1478.6} \right)$$

$$= 2.2619 \ m^3$$

And the expansion ratio is:

$$= \frac{V_3}{V_1}$$

$$= \frac{2.2619}{0.0283}$$

$$= 79.88$$

At point 1 the quality of the fluid is:

$$x_1 = \frac{\frac{V_1}{W} - 1/\rho_{f13}}{v_{g13} - 1/\rho_{f13}}$$

$$= \frac{\frac{0.0283}{15.680} - 1/\ 1478.6}{0.17011 - 1/\ 1478.6}$$

$$= 0.6667 \ \%$$

Therefore, the entropy of the working fluid at point 1 is:

$$s_1 = s_{f13} + x_1 \left(s_{g13} - s_{f13} \right)$$

$$= 0.11478 + 0.006667 \times (0.45171 - 0.11478)$$

$$= 0.11703 \ \ KJ/\ Kg$$

The enthalpy of the fluid at point 1 is:

$$h_1 = h_{f13} + x_1 (h_{g13} - h_{f13})$$

$$= 54.597 + 0.006667 \times (234.74 - 54.597)$$

$$= 55.798 \text{ KJ/ Kg}$$

The total enthalpy at this point is:

$$H_1 = h_1 W$$

$$= 55.798 \times 15.680$$

$$= 874.92 \text{ KJ}$$

The heat added during the process 1-2 is:

$$\Delta H_{12} = H_2 - H_1$$

$$= 4085.1 - 874.92$$

$$= 3210.2 \text{ KJ}$$

The heat rejected during the process 3-1 is:

$$\Delta H_{31} = H_1 - H_3$$

$$= 874.92 - 3249.4$$

$$= 2374.5 \text{ KJ}$$

The work done during the cycle is:

$$\Delta \overline{W} = H_2 - H_3$$

$$= 4085.1 - 3249.4$$

$$= 835.7 \text{ KJ}$$

Finally, the efficiency of the cycle is:

$$\eta = \frac{\Delta \overline{W}}{\Delta H_{12}}$$

$$= \frac{835.7}{3210.2}$$

$$= 26.032\ \%$$

To compare the efficiency of this cycle the efficiency of the Rankine cycle, with no superheat, with the same maximum and minimum temperatures can be calculated. The difference between the Rankine and F.P.G. cycles is in the heat addition process. At the start of the heat addition process of the F.P.G. cycle, the working fluid is in the wet vapour region, whereas, in the Rankine cycle, the working fluid is in the supercooled liquid region. Additionally, the Isochoric heat addition of the F.P.G. cycle is different from the heat addition of the Rankine cycle, which is Isobaric. The Rankine cycle also has a pumping process which does not form part of the F.P.G. cycle. Assuming that the working fluid of the Rankine cycle is a saturated liquid, then at the start of the pumping and heat addition processes, the enthalpy of the fluid will be 54.597 KJ/ Kg. Therefore, the total enthalpy of the working fluid at this point is:

$$H_{1R} = h_{1R}\ W$$
$$= 54.597 \times 15.680$$
$$= 856.09 \text{ KJ}$$

The heat addition process must therefore supply:

$$\Delta H_{12R} = H_2 - H_{1R}$$
$$= 4085.1 - 856.09$$
$$= 3229.0 \text{ KJ}$$

If the pump work is neglected, then the efficiency of the cycle is:

$$\eta_R = \frac{\Delta \bar{W}}{\Delta \bar{H}_{12R}}$$

$$= \frac{835.7}{3229.0}$$

$$= 25.88\ \%$$

If the pump work is included, it is approximately:

$$\bar{W}_p = \frac{P_2 V_2 \rho_2}{\rho_1}$$

$$= \frac{4309.6017 \times 0.0283 \times 553.74}{1478.6}$$

$$= 45.701 \text{ KJ}$$

Therefore the total work output is:

$$\Delta \bar{W}_R = \Delta \bar{W} - \bar{W}_p$$

$$= 835.7 - 45.701$$

$$= 790.0 \text{ KJ}$$

This is the real work output of the Rankine cycle and, therefore, the efficiency is:

$$\eta_{RP} = \frac{\bar{W}_R}{\bar{H}_{12R}}$$

$$= \frac{790.0}{3229.0}$$

$$= 24.47\ \%$$

It can be seen that, under the given conditions, the F.P.G. cycle is more efficient than the Rankine cycle. It must be pointed out, however, that the efficiencies, as calculated, are ideal. In comparison to the Rankine cycle, the F.P.G. cycle has other advantages, including, negligible piping (no flow losses), no pumping and , as shown in the preceding calculations, less heat is added and rejected.

The next step in the analysis of the F.P.G. cycle is to improve the efficiency. Two methods of obtaining a better efficiency are to be examined. The first of these is accomplished by changing the Isochoric heat addition process to a combination of an Isochoric and an Isobaric heat addition process. The second modification involves increasing the volume of the vessel while retaining the same mass of working fluid, this increasing the quality of the working fluid at the end of the heat rejection process. Both of the modifications involve the adjustment of the maximum temperature and pressure of the cycle. in all cases it is considered to be on the saturated vapour line. In both cases, the maximum operating conditions are lower than the critical conditions, both in temperature and pressure. It must be decided whether it is preferable to obtain a maximum output or a maximum efficiency. in this case, the modified cycles are analysed with a maximum output cycle.

The first of these two modifications is the two process heat addition. This cycle is shown in Figures 9 and 10, which are the T-s and P-v diagrams, respectively. The Figures show the maximum output cycle for Refrigerant 11. The cycle modifications, in this case, are in the heat addition process, combined with the adjustment of the maximum cycle temperature and pressure. The heat addition process in this case is:

1-1'   Isochoric heat addition

1'-2   Isobaric heat addition

The efficiency of this cycle with a heat rejection temperature of 24° C is 26.51 %, an increase of 0.48 %. The output has been raised by 16 % to 968.5682 KiloJoules per stroke. The maximum cycle temperature has been lowered to 182° C and the pressure has been lowered to 3408.34 KiloPascals (21 % lower than the original cycle pressure).

Figure 11 is a graph of the temperature at point 2 with respect to the cycle efficiency and work output. It can be seen that the maximum efficiency of this cycle is obtained at 193° C. The efficiency, in this case, is 26.84 %, 0.81 % higher than the original cycle. The output of the cycle at this point is 950.498 KiloJoules, which is 13.7 % higher than the original cycle.

The advantages of this modification are:

a) Lower temperature

- Lower heat losses
- Lower temperature heating fluid can be used

b) Lower pressure

- The shell of the vessel can be thinner

c) Increased efficiency

d) Increased output

In order to accomplish this modification the vessel must be redesigned in order to accomodate the equipment that would produce an Isobaric heat addition. This equipment would probably be a piston and spring arrangement, similar to that shown in Figure 12. The action of this mechanism would be to stay in its least volume state (spring fully extended) until the desired cycle pressure is reached. At this point the pressure force would be enough to move the piston against the spring. From this point the piston would compress the spring in order to maintain the pressure as the volume of the working fluid increases. The spring would have to apply a constant force on the piston throughout the stroke of the piston. When the maximum volume of the mechanism is reached valve a would be opened releasing the working fluid to work on the magnetic piston. For a part of the stroke the mechanism would maintain the pressure in the vessel, this would transfer the work done on the constant pressure mechanism into useful work. As the pressure drops, the piston would return to its least volume state through the action of the spring. This mechanism would give an Isobaric heat addition process.

The disadvantages of this system are:

a) Additional equipment
- More design problems
- More maintenance is required

b) Additional seals
- More possibility of leakage

The second modification is to simply increase the volume of the vessel while retaining the same mass of working fluid. The T-s and P-v diagrams for the maximum output cycle of this modification using Refrigerant 11 are Figures 13 and 14, respectively. The efficiency of this cycle is 26.8 %, 0.77 % higher than the cycle operating with critical temperature and pressure. The work output, maximum cycle temperature and maximum cycle pressure are the same as the maximum output cycle operating with the Isochoric-Isobaric heat addition. Figure 15 shows the graph of the efficiency and the work output with respect to the temperature at point 2. The maximum efficiency is at 193° C and is now 26.98 %, 0.95 % higher than the original cycle.

The advantages of this modification are:

a) No major equipment modification (except in size)

b) Larger surface areas for heat exchange in the vessel

c) No increase in the number of moving parts

In comparison with the cycle operating at critical temperature and pressure, there appears to be no major disadvantage, however the larger surface area of the vessel might increase the amount of heat lost to atmosphere.

As this cycle is the most efficient of the three discussed, it will be used for subsequent calculations. Instead of choosing the cycle with the best efficiency or with the best output, these calculations are done on an intermediate point. Referring to Figure 15, it is seen that at 188° C, that the efficiency is near the maximum as is the work output. The following numerical calculations give the values that will be used in the subsequent analyses.

The data for this cycle is based on the following assumptions:

- Refrigerant 11 is the working fluid
- Maximum cycle temperature is 188° C and the maximum cycle pressure 3707.58 KiloPascals
- Minimum cycle pressure is atmospheric pressure (24° C)
- The weight of the working fluid is 15.680 Kg
- Isentropic expansion

The volume of the vessel with this weight of working fluid is found using the following equation:

$$V_v = \frac{v_{g2} V_c}{v_c} \tag{19}$$

Where v_c and V_c are the specific volume of the working fluid and the volume of the vessel, respectively, for the cycle operating at critical temperature and pressure.

At point 2, the data for the working fluid is:

h_2 = 286.58 KJ/ Kg

s_2 = 0.43318 KJ/ Kg

v_{g2} = 0.003569 m^3/ Kg

ρ_{g2} = 280.17 Kg/ m^3

At the critical temperature and pressure the data is:

v_c = 0.001806 m^3/ Kg

V_c = 0.0283 m^3

The volume of the vessel is:

$$V_v = \frac{v_{g2} V_c}{v_c}$$

$$= \frac{0.03569 \text{ X } 0.0283}{0.001806}$$

$$= 0.05597 \text{ m}^3$$

The mass of the working fluid is:

$$W = V_v \rho_{g2}$$

$$= 0.05597 \text{ X } 280.17$$

$$= 15.680 \text{ Kg}$$

The total enthalpy of the working fluid is:

$$H_2 = h_2 W$$

$= 286.58 \times 15.680$

$= 4493.7$ KJ

The data for the working fluid at the saturated liquid and saturated vapour lines intersecting with the extended process line 1-3 are:

$v_{g13} = 0.17011\ m^3/\ Kg$

$\rho_{f13} = 1478.6\ Kg/\ m^3$

$h_{f13} = 54.597\ KJ/\ Kg$

$h_{g13} = 234.74\ KJ/\ Kg$

$s_{f13} = 0.11478\ KJ/\ Kg$

$s_{g13} = 0.45171\ KJ/\ Kg$

At point 3, the quality of the working fluid is:

$$x_3 = \frac{s_2 - s_{f13}}{s_{g13} - s_{f13}}$$

$$= \frac{0.43318 - 0.11478}{0.45171 - 0.11478}$$

$$= 94.50\ \%$$

The enthalpy of the working fluid at point 3 is:

$$h_3 = h_{f13} + x_3\ (\ h_{g13} - h_{f13}\ )$$

$$= 54.597 + 0.9450 \times (\ 234.74 - 54.597\ )$$

$$= 224.84\ KJ\ /\ Kg$$

The total enthalpy of the fluid at point 3 is:

$$H_3 = h_3\ W$$

$$= 224.84 \times 15.680$$

$$= 3525.5\ KJ$$

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The volume of the working fluid at point 3 is:

$$V_3 = W \left(v_{g13}\, x_3 + \frac{(1 - x_3)}{\rho_{f13}} \right)$$

$$= 15.680 \times \left(0.17011 \times 0.9450 + \frac{(1 - 0.9450)}{1478.6} \right)$$

$$= 2.5213 \ m^3$$

The Expansion ratio of the fluid is:

$$= \frac{V_3}{V_1}$$

$$= \frac{2.5213}{0.05597}$$

$$= 45.05$$

At point 1, the quality of the fluid is:

$$x_1 = \frac{\frac{V_1}{W} - 1/\rho_{f13}}{v_{g13} - 1/\rho_{f13}}$$

$$= \frac{\frac{0.05597}{15.680} - \frac{1}{1478.6}}{0.17011 - \frac{1}{1478.6}}$$

$$= 1.707 \ \%$$

The entropy of the fluid at point 1 is:

$$s_1 = s_{f13} + x_1 (s_{g13} - s_{f13})$$

$$= 0.11478 + 0.01707 \times (0.45171 - 0.11478)$$

$$= 0.1205 \ KJ/\ Kg$$

The enthalpy at point 1 is:

$$h_1 = h_{f13} + x_1 (h_{g13} - h_{f13})$$
$$= 54.597 + 0.01707 \times (234.74 - 54.597)$$
$$= 57.674 \text{ KJ/ Kg}$$

The total enthalpy of the working fluid at point 1 is:

$$H_1 = h_1 W$$
$$= 57.674 \times 15.680$$
$$= 904.32 \text{ KJ}$$

Therefore the heat added during the cycle is:

$$\Delta H_{12} = H_2 - H_1$$
$$= 4493.7 - 904.32$$
$$= 3589.4 \text{ KJ}$$

The heat rejected by the system is:

$$\Delta H_{31} = H_1 - H_3$$
$$= 904.32 - 3525.5$$
$$= - 2621.2 \text{ KJ}$$

The work done during each cycle is:

$$\Delta W = H_2 - H_3$$
$$= 4493.7 - 3525.5$$
$$= 968.2 \text{ KJ}$$

Finally, the efficiency of the cycle is:

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$$\eta = \frac{\Delta \overline{W}}{\Delta H_{12}}$$

$$= \frac{968.2}{3589.4}$$

$$= 26.97 \%$$

It is possible to increase the efficiency of all of the previously described cycles, by superheating the vapour. For this possibility, no precise calculations were performed. However, rough calculations were made showing an efficiency of about 31 % and an output 44 % greater than the cycle working from the critical temperature and pressure. Care must be taken in this case, as it is desirable to maintain the conditions suitable for condensation. This means that no superheated vapour can be tolerated at the end of the expansion process. With a real process (non-isentropic) these conditions might be hard to obtain consistently, because the end state of the fluid might still be located in the superheated vapour region.

The advantages of this modification are the increase in efficiency and output. However, there are disadvantages, such as:

a) Higher maximum temperatures

b) Higher maximum pressures

c) Controls for the opening of the valve a would have to be more accurate in order to obtain the correct conditions for condensation

2. Heat Transfer Processes

For the heat transfer processes care must be taken to maximize the heat tmansfer rates and thereby minimize the time. Two heat transfer processes are involved, these are the heat addition (1-2) and the heat rejection (3-1) processes. These are first analysed with respect to the hemispherically shaped vessel as shown in Figure 4.

The heat addition process (1-2) is a horizontal flat plate heat addition. It is assumed that the temperature of the plate surface varies with respect to the temperature of the working fluid, so as to maintain a constant heat flux of 63.09 KJ/ m^2- sec (20000 BTU/ ft^2-hr). The volume of the vessel is calculated for 15.680 Kg of Refrigerant 11 as the working fluid and is 0.05597 m^3. For a hemispherical container, it has a diameter of 0.598 m. The flat surface area that is available for heat addition is 0.2808 m^2 and the hemispherical area that is available for heat rejection is 0.5615 m^2.

The heat transfer coefficients can be obtained from the ASHRAE handbook (2). Assuming that there exists a heat flux density of 63.09 KJ/ m^2- sec (20000 BTU/ ft^2-hr), the heat transfer coefficient at 188 °C is 3.237 KJ/ m^2-sec- °C (570 BTU/ ft^2-hr- °F). This gives a temperature differential between the surface and the working fluid of 20 °C and a surface temperature of 208 °C. At 24 °C, the heat transfer coefficient is 2.36 KJ/ m^2-sec- °C (415 BTU/ ft^2- hr- °F) and the temperature differential is 27 °C, then the surface temperature is 51 °C. As the heat addition process requires 3589.4 KJ and the heat is transferred at the rate of 17.72 KJ/ sec, this heating process takes 3 minutes and 23 seconds. This time can be lowered only slightly by increasing the temperature differential between the working fluid and the heating surface because the temperature differential for the maximum heat flux is 28 °C. If the temperature differential is more than 28 °C, the heat flux is reduced due to the transition from nucleate boiling to film boiling.

The heat rejection process (3-1) is Isobaric. This process rejects 2621.2 KJ per stroke. In order to determine the heat transfer coefficient in process 3-1, the following assumptions are made: The heat is rejected by film type condensation and the temperature differential is 11 °C. The area available for the heat rejection is 0.5615 m^2.

Referring to the ASHRAE handbook (3), the following relationship (converted to SI units) is found:

$$h_c = 2624\, F_2 \left(b / w_1 \right)^{1/3} \tag{20}$$

Where h_c is the heat transfer coefficient (KJ/ sec- m^2 - K), F_2 is a condensation factor based on the fluid and the temperature at which the condensation takes place and w_1 is the condensate flow rate (Kg/ sec) over the condensing surface. The symbol b, is the breadth (m) of the condensing surface over which the condensed fluid passes. In this case, b is the perimeter at the base of the hemispherical condensing surface.

Using a series of successive approximations, it is found that the cooling process takes approximately 5 minutes and 22 seconds.

This gives a total heat exchange time for the two processes of about 8 minutes and 45 seconds. It is expected that the time required for the expansion process is much faster than that required for the heating and cooling processes. The time for a complete cycle is only slightly longer than the time for the heat transfer processes (1-2) and (3-1). It must be pointed out that the above estimated time for the heat transfer processes does not include the time that is required to heat or cool the body of the vessel.

In order to reduce the heat exchange time, the vessel can be modified as shown in Figure 16 or 17. Figure 16 shows a horizontal tube with a heat rejection surface on the upper portion, A, and a heat addition surface on the lower portion, B, of the vessel. To match the requirements of the volume of the working fluid, the inside diameter and the length of the tube are chosen as 10.8 cm and 6.1 m, respectively. The liquid part of the working fluid, at the start of the heating process, covers about 30.6 % of the total surface area of the vessel, this is considered to be the heating surface. The rest of the surface is used for the heat rejection process. As the total inner surface of the tube has an area of 2.07 m^2, 0.6234 m^2 is used for the heat addition process and 1.4376 m^2 is used for the heat rejection process. This modification of the vessel reduces the total heat exchange time to about 3 minutes and 20 seconds. The total heat exchange time can be further reduced to 2 minutes and 15 seconds if the diameter and the length of the tube are changed to 7.62 cm and 12.272 m, respectively.

Using the modification as shown in Figure 17 with a 15.24 cm inside diameter tube as the outer shell and a 13.2 cm outside diameter tube as the inner tube, the heat exchange time can be reduced to about 1 minute and 13 seconds. In this case, the inner tube has the functions of

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increasing the heat rejection area and decreasing the condensed liquid film thickness.

The advantages of these modifications, in comparison with the original vessel, include increasing the heat transfer areas with thinner vessel walls and shortening the heat exchange times.

The disadvantages of these modifications are that more space is required and an increased external surface area increases the heat lost from the heating fluid. Also, a larger volume of material is used in the fabrication of the vessel and more liquids are held by the jackets, these will have to be heated during the heat addition process and cooled during the heat rejection process. This results in a longer heat exchange time and a greater heat input required.

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3. Magnetic-Electric Interaction

Finally, the coil-magnet interaction will be analysed. Similar to an electrical generator, a relative motion of a magnetic field with respect to a coil induces a current in the coil (winding) of the unit. This current causes a force that opposes the relative motion of the magnetic field. The relationship between the voltage e, the induced current I, the force F_e and the relative velocity of the field v is given by the formula:

$$e\ I = F_e\ v \tag{21}$$

Where the following relationships hold:

$$F_e = Z\ B\ \ell\ I \tag{22}$$

$$e = Z\ B\ \ell\ v \tag{23}$$

And the symbols are:

Z : the number of turns of the coil

B : the magnetic flux density

ℓ : the length of one turn of the coil

In these equations, the values of Z, B and ℓ are knowns, e, I, F_e and v are four unknown variables.

Newtons equation applied to the piston gives:

$$F_t = m\ a = m\ \frac{dv}{dt} \tag{24}$$

Where F_t is the total force acting on the piston, m, a and v are respectively the mass, acceleration and velocity of the piston. F_t is the summation of the pressure force F_p, the frictional force F_f and the force caused by the electrical field F_e, as follows:

$$F_t = F_p - F_f - F_e \tag{25}$$

The force caused by the pressure is a function of the displacement of the piston:

$$F_p = A\,P(z) \tag{26}$$

Where A is the area of the piston and P(z) is the pressure as a function of the displacement z.

The force caused by friction is dependent on the velocity of the piston:

$$F_f = K\,v \tag{27}$$

Where K is a constant which depends on the materials of the two bearings and the material of the inner shell of the cylinder.

As shown earlier, in equation (21), the electrical field force can be given by:

$$F_e = \frac{e\,I}{v} \tag{28}$$

It can be seen that the magnetic-electrical interaction is very complex and interdependent. Because the velocity of the piston is an important factor affecting this interaction, the diameter and stroke of the piston must be chosen very carefully.

DESIGN CONSIDERATIONS

This section of the report deals with the design of the components of the F.P.G. machine. The ideas mentioned below should be incorporated into the design of a prototype, in order to insure the proper operation of the machine.

The first sub-assembly is the piston-cylinder assembly. The piston is a moving part and it should be stable. In order to make the piston stable, the length of the piston should be greater than its diameter and the nylon slide bearings should be put as close to the ends of the piston as is possible. The poles of the magnet should be along the length of the piston. The seals that are selected for the piston should be capable of withstanding high linear speeds. The material that is used to fabricate the seals should have a low coefficient of friction without the need of lubricants and should be compatible with the working fluid.

The cylinder length of the prototype should be longer than the expected stroke of the piston, this is in order to insure that the piston will not hit the end and be damaged. The inner tube should be made of a single piece of phenolic tubing. If this is not possible, great care must be taken in order to insure the alignment of the segments of this tube. To align the segments correctly, they can be assembled on an expanding mandrel. The coil can then be wound around the inner tube while still on the mandrel. The coil and inner tube can then be inserted into the outer shell of the cylinder. Resin can then be poured between the inner tube and the outer shell, when the resin sets, it secures the inner tube and coil to the outer shell. After the resin has set, the mandrel can be removed.and the tube can be honed. This will produce a cylinder with a straight bore and no misalignments of the segments which could wear the nylon slide bearings.

The coil should be designed to maintain a velocity profile of the piston as shown in Figure 18, with the maximum velocity segment as long as possible. To accomplish this, the coil may not be of a uniform depth, with respect to the number of layers, along its length.

The bellows should have the ability to withstand more expansion than expected. It must be designed to operate well under conditions of rapid expansion. The space between the bellows and the piston should be filled with the same refrigerant as used as the working fluid. Care must be taken in the design of the bellows and its guides, as these components are not very strong and they can effect the operation of the cycle if they malfunction.

The complete piston-cylinder assembly should be inclined so as to promote the flow of condensed working fluid back to the vessel. This lowers the efficiency of the cycle slightly by increasing the pressure at the end of the expansion stroke, due to the weight of the piston. The increased pressure causes an increase in temperature of the working fluid, this reduces the time required for heat rejection by increasing the temperature differential between the working fluid and the cooling medium. It is also important to keep the clearance volume between the piston and valve assembly to a minimum, this not only includes the space between the piston and the closed end of the cylinder, but also the lines that connect the piston-cylinder assembly to the valve assembly.

The action of the valve a in the valve assembly must be positive and extremely rapid. When the valve slide moves to open the port and the snap-over-centre device is in its unstable position (centred), the port of valve a must be closed. If there is any opening of the port of valve a some vapour is allowed to escape from the vessel assembly to the piston-cylinder assembly. This would lower the pressure in the vessel, which in turn, would allow the valve slide to return to the closed position. The port area of the valve a should be the same as the lines conecting the valve to the vessel assembly and to the piston-cylinder assembly. The valve should be as thin as possible.

The vessel should, preferably, be of the horizontal cylinder type, as shown in Figures 16 and 17. The bottom of the vessel must always be kept level. To minimize the loss of heat from the heating fluid to the cooling fluid, as the heating process begins, the cooling jacket should be drained at the end of its process. Similarly, this should be done to the heating fluid at the end of its process. If the heating fluid is gaseous, then the jacket should be designed to hold a minimum of it. In order to increase the heat transfer rates, it might be desirable to add fins to the heat exchange surfaces.

The choice of a working fluid is very important. For the analysis in this report Refrigerant 11 was chosen. The temperature of the R-11 at atmospheric pressure is 24° C and the temperature of the cooling fluid (water) is 13° C, this gives a temperature differential of 11° C between the working fluid and the cooling fluid. Among the refrigerants, for which material properties are available (4), R-11, used in the new cycle, possesses the best calculated efficiency. There are other refrigerants for which material properties are not available, but could be used in this application. Among these are R-611 (Methyl Formate), R-610 (Ethyl Ether), R-30 (Methylene Chloride), R-1130 (Dichloroethylene) and R-1120 (Trichloroethylene), these refrigerants have boiling temperatures at atmospheric pressure close to that of R-11. If the temperatures of the heating and cooling fluids are changed, the working fluid must be changed in order to match the new heating and cooling conditions. The materials that are used in the construction of the F.P.G. must be selected knowing that they are compatible with the working fluid.

CONCLUSIONS

To evaluate the feasibility of any new developement, the positive and negative aspects of the design must be considered.

The major difficulties of the F.P.G. are the heat-exchange times and the design of the bellows. The heat-exchange times can be reduced by increasing the heat transfer surface area, as shown in the Heat Transfer Processes section of the Performance chapter. With the reduction of the time for the heat exchange processes, a reduction in the efficiency of the cycle is expected. The requirement for a larger heat exchange surface causes an increase in the material required to fabricate the vessel, this requires additional heat to heat the vessel. Similarly, the cooling fluid must cool down the vessel before condensation can commence. To obtain a reasonable efficiency from the F.P.G., the sizing of the vessel must be analysed as to its effect on the heating and cooling fluids.

The design of the bellows is a different type of problem, being related to the material used in its construction and the method of fabrication. It would be possible to operate the F.P.G. without the bellows but the refrigerant would leak to atmosphere through the piston seals. This would mean that the refrigerant would require continuous monitoring and refilling. However, the F.P.G. can be modified as shown in Figure 19. This modification does not require bellows. An additional vessel assembly and an additional valve assembly are used to replace the bellows. This modification has the advantage of producing a power output from the piston moving in both directions.

Because the cycle produces an intermittent electrical power output, it presents another problem of the utilization of the power output. The solution of this problem is to use the power output to charge batteries. The batteries act as a buffer and an accumulator between the F.P.G. and the equipment that uses the developed power.

A characteristic of the F.P.G. is the simplicity of construction and maintenance. Another characteristic is that the heat can be supplied to the machine across the heating surface from a variety of sources, such as solar energy or industrial waste energy.

In comparison to the Rankine cycle, the F.P.G. has several advantages. As mentioned previously, the construction of the F.P.G. is much simpler than that required for a machine operating on the Rankine cycle. With the same temperature limits and the working fluid as a saturated vapour at the end of the heat addition process, the new cycle has an efficiency comparable to that of the Rankine cycle. However, the Rankine cycle has the advantage of being able to extend into the superheated region, whereas, the F.P.G. has a very limited range of application in the superheated region.

It is felt that the developement of the F.P.G. has the potential of being an alternative method of energy conversion. It would be wise to fabricate a prototype before any further studies are undertaken. This could be done on a relatively small scale so that the cost would not be great.

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2. ASHRAE, Handbook of Fundamentals, (1972), p. 46, Figure 10.

3. ASHRAE, Handbook of Fundamentals, (1972), p. 52, Table 10, Equation (2).

4. ASHRAE, Handbook of Fundamentals, (1972), p. 246, Table 2.

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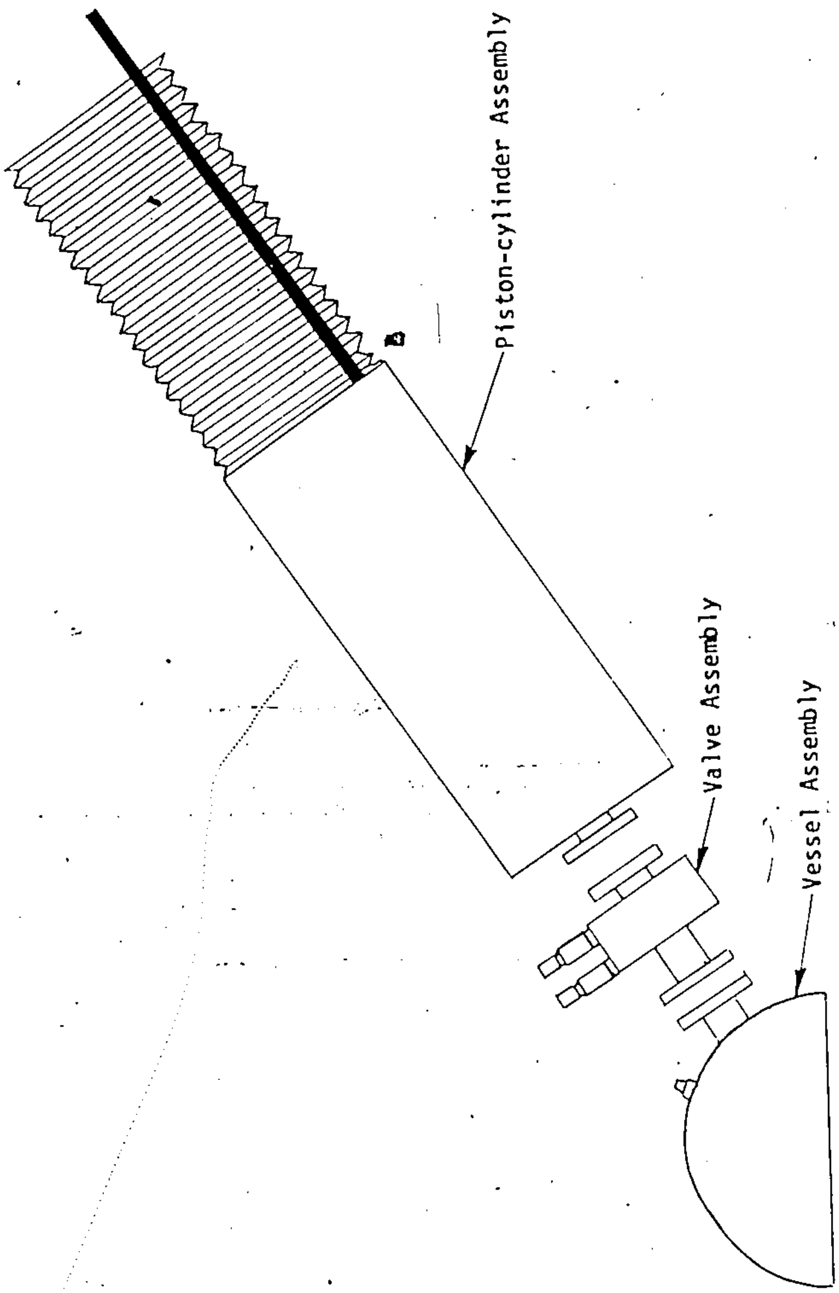

Figure 1. Assembly of the Free Piston Generator

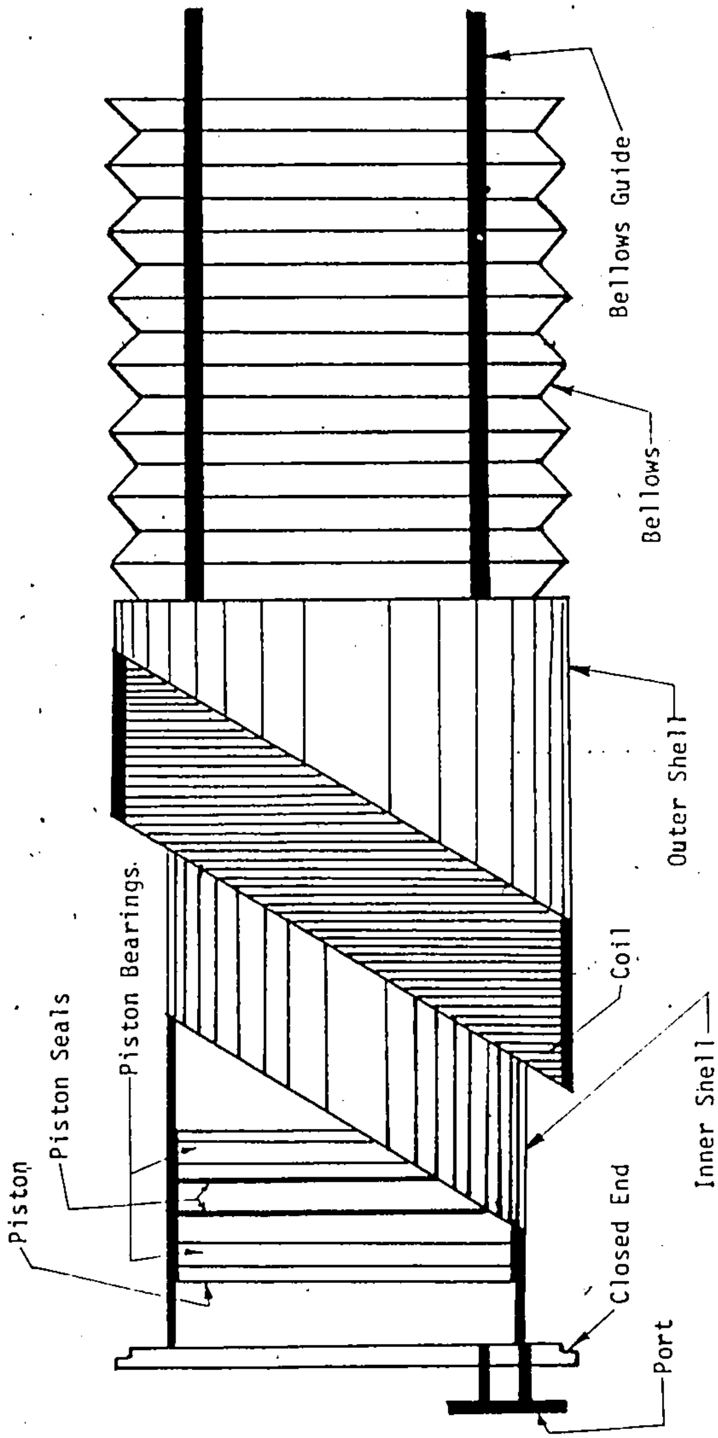

Figure 2. Piston-cylinder Assembly

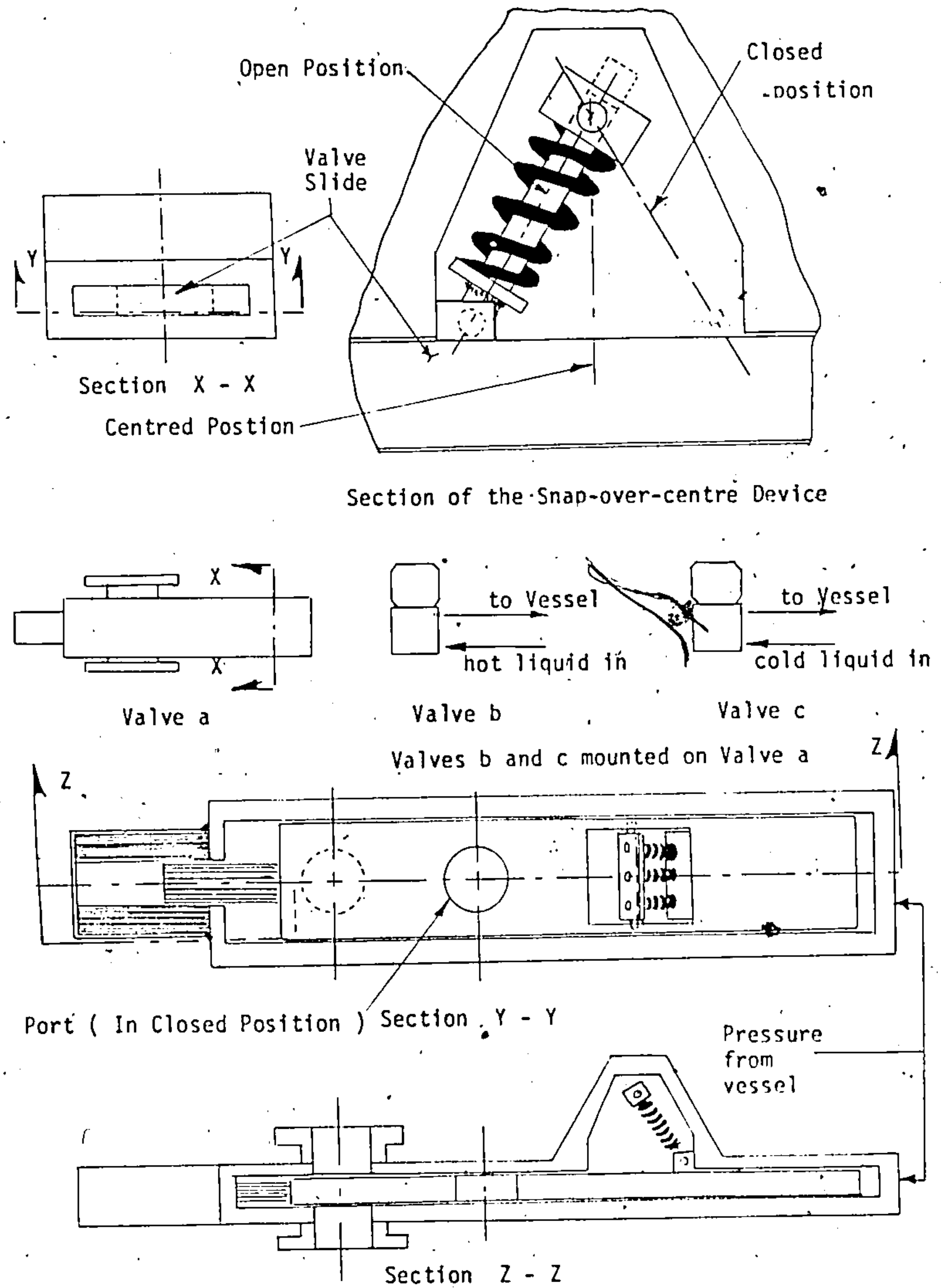

Figure 3. Valve Assembly

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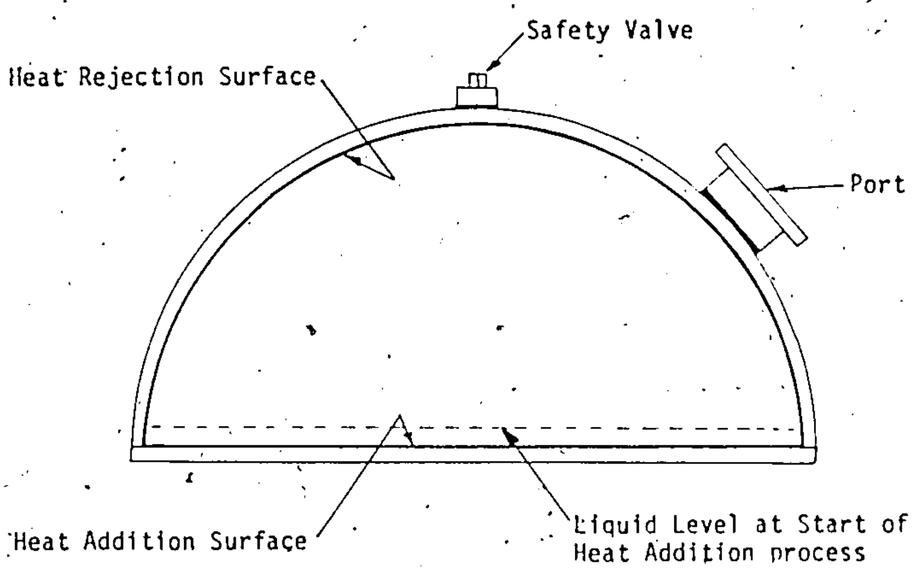

Figure 4. Vessel Assembly

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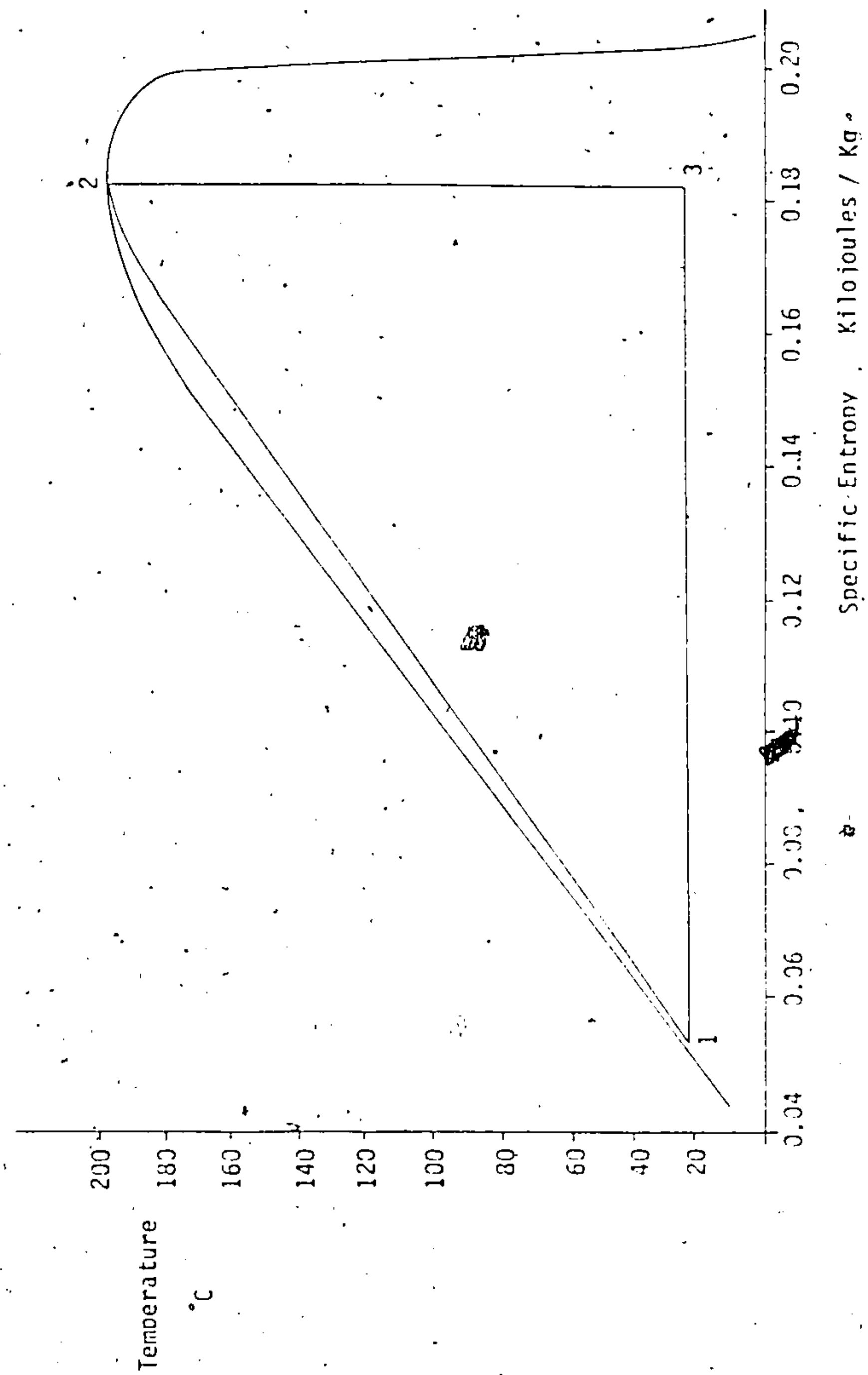

Figure 5. T-s Diagram

(Critical Conditions at Point 2, R-11)

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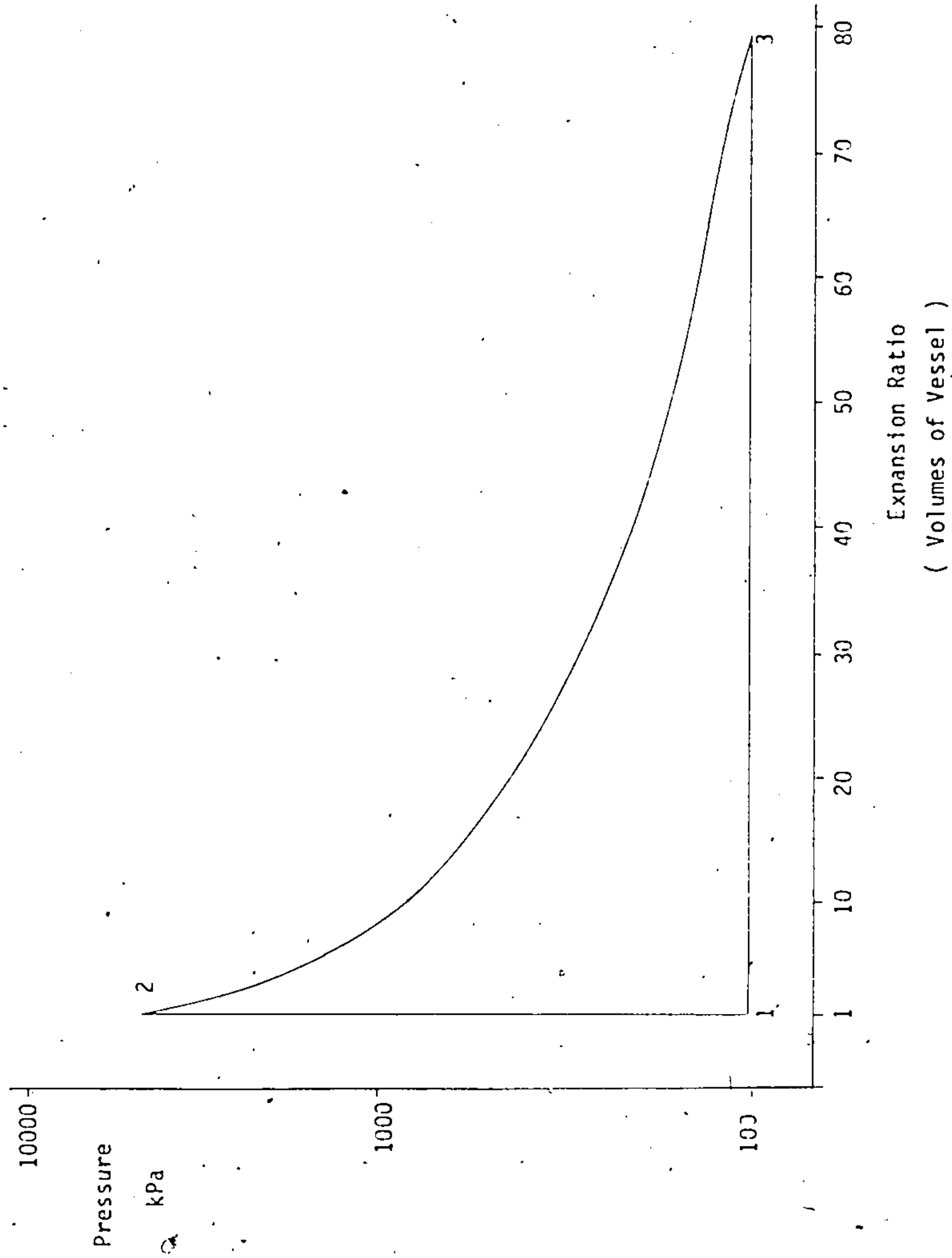

Figure 6. P-v Diagram

(Critical Conditions at Point 2, R-11)

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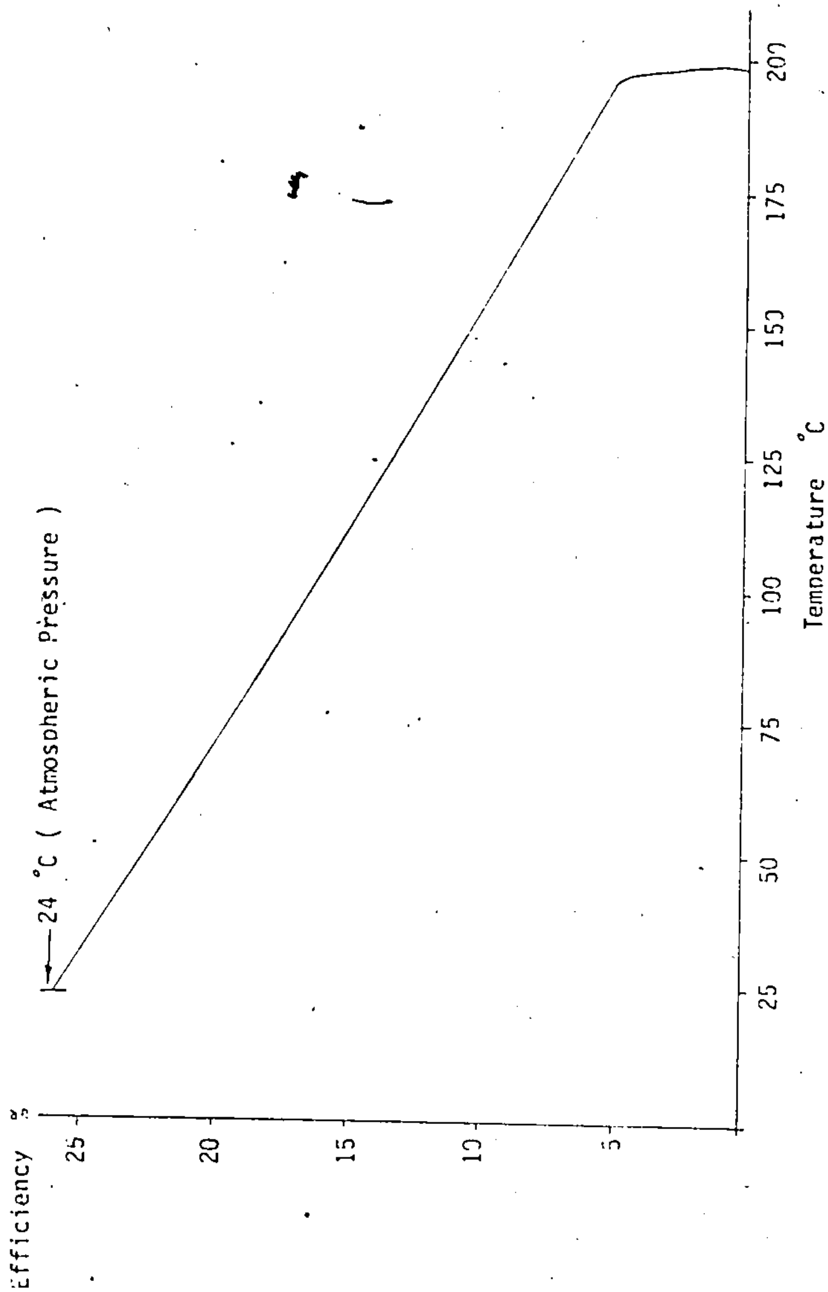

Figure 7. Efficiency vs. Temperature at Point 1. (R-11)

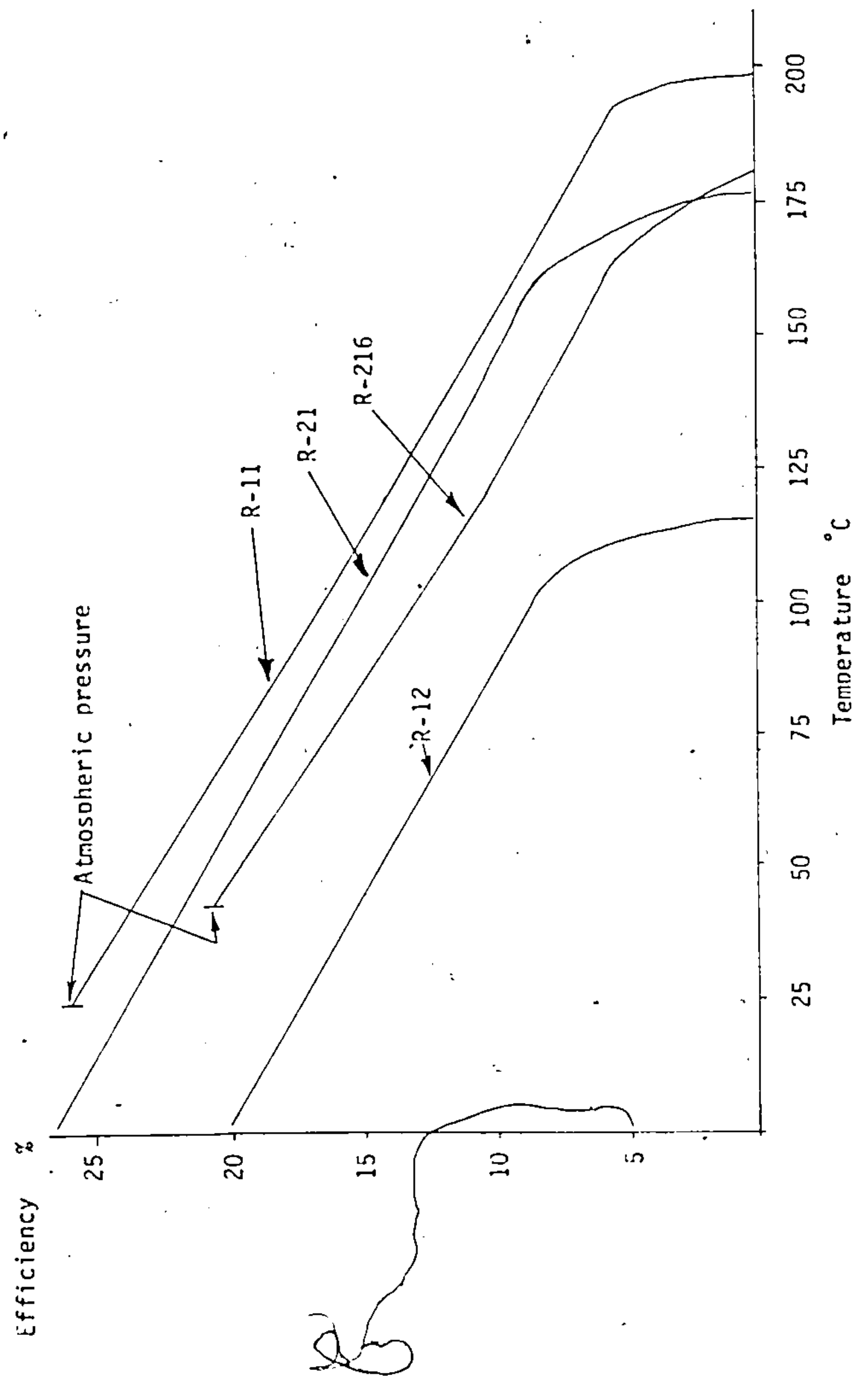

Figure 8. Efficiency vs. Temperature at Point 1, (Several Refrigerants)

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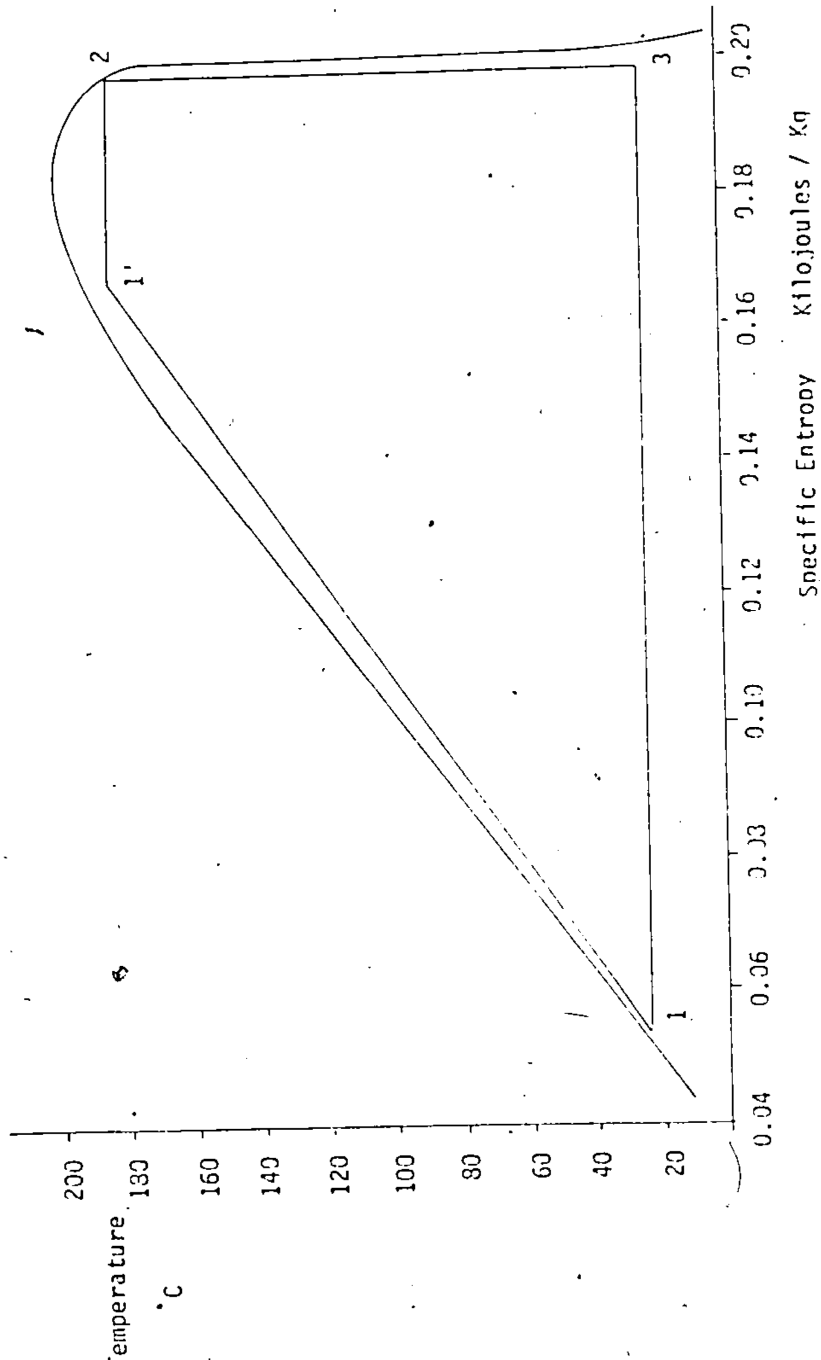

Figure 9. T-s Diagram

(Two Process Heat Addition, R-11)

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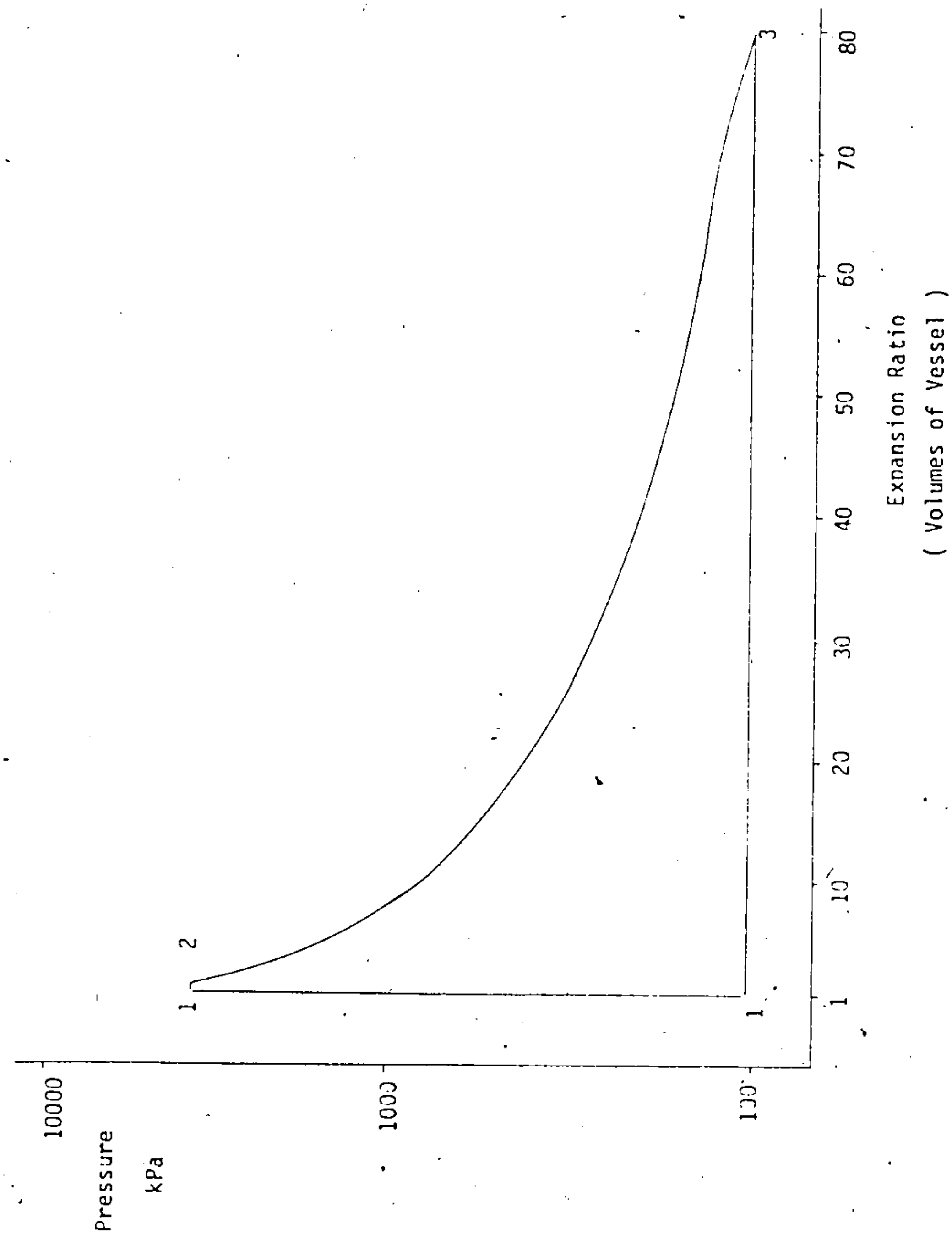

Figure 10. P-v Diagram

(Two Process Heat Addition, R-11)

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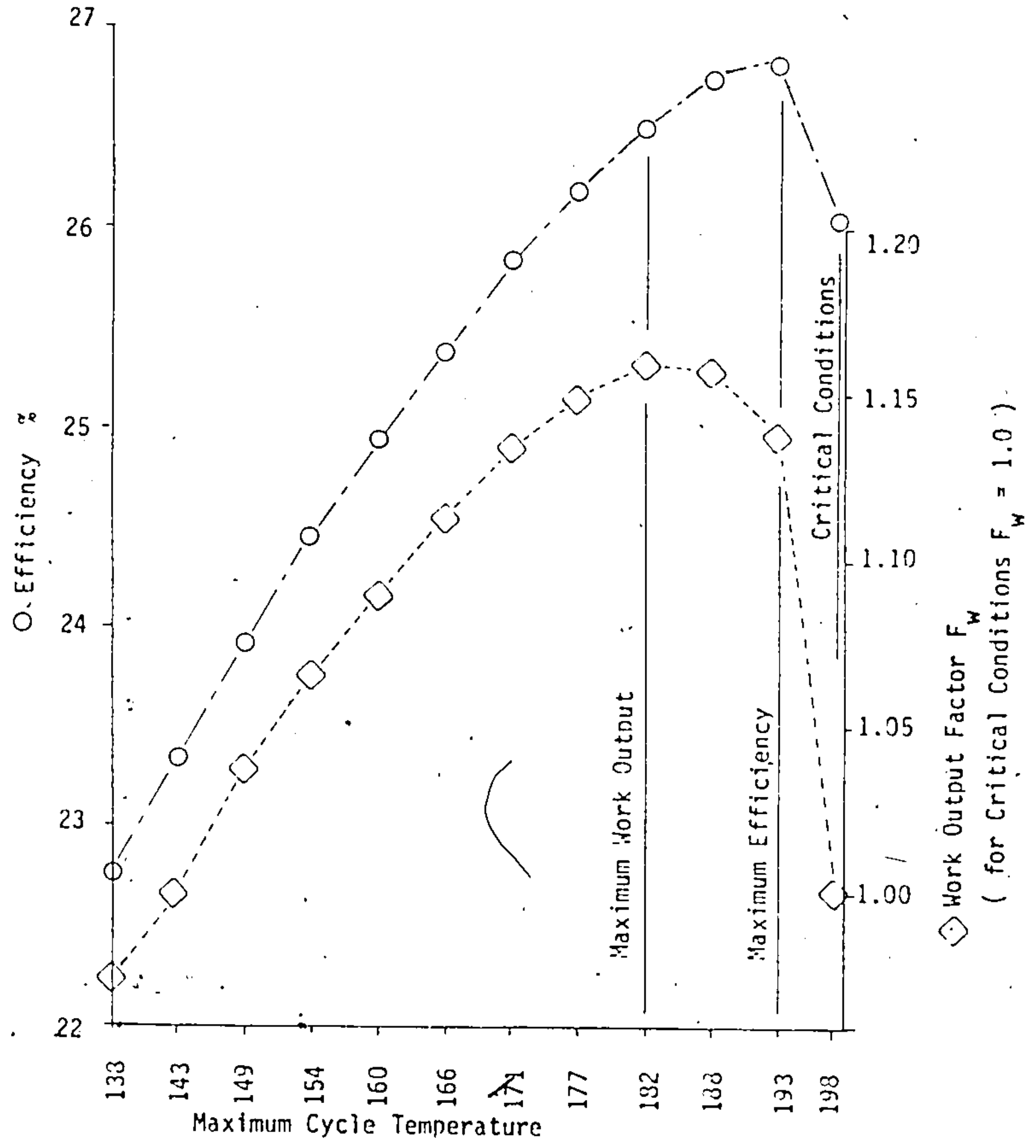

Figure 11. Efficiency & Work Output vs. Maximum Cycle Temperature

(Two Process Heat Addition, R-11)

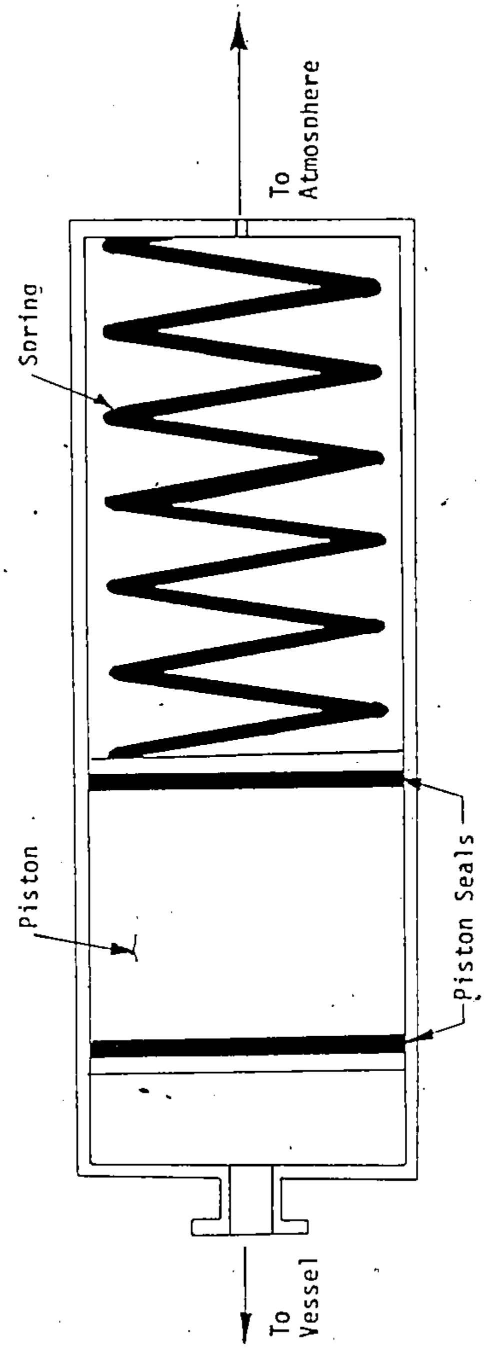

Figure 12. Modification for Two Process Heat Addition

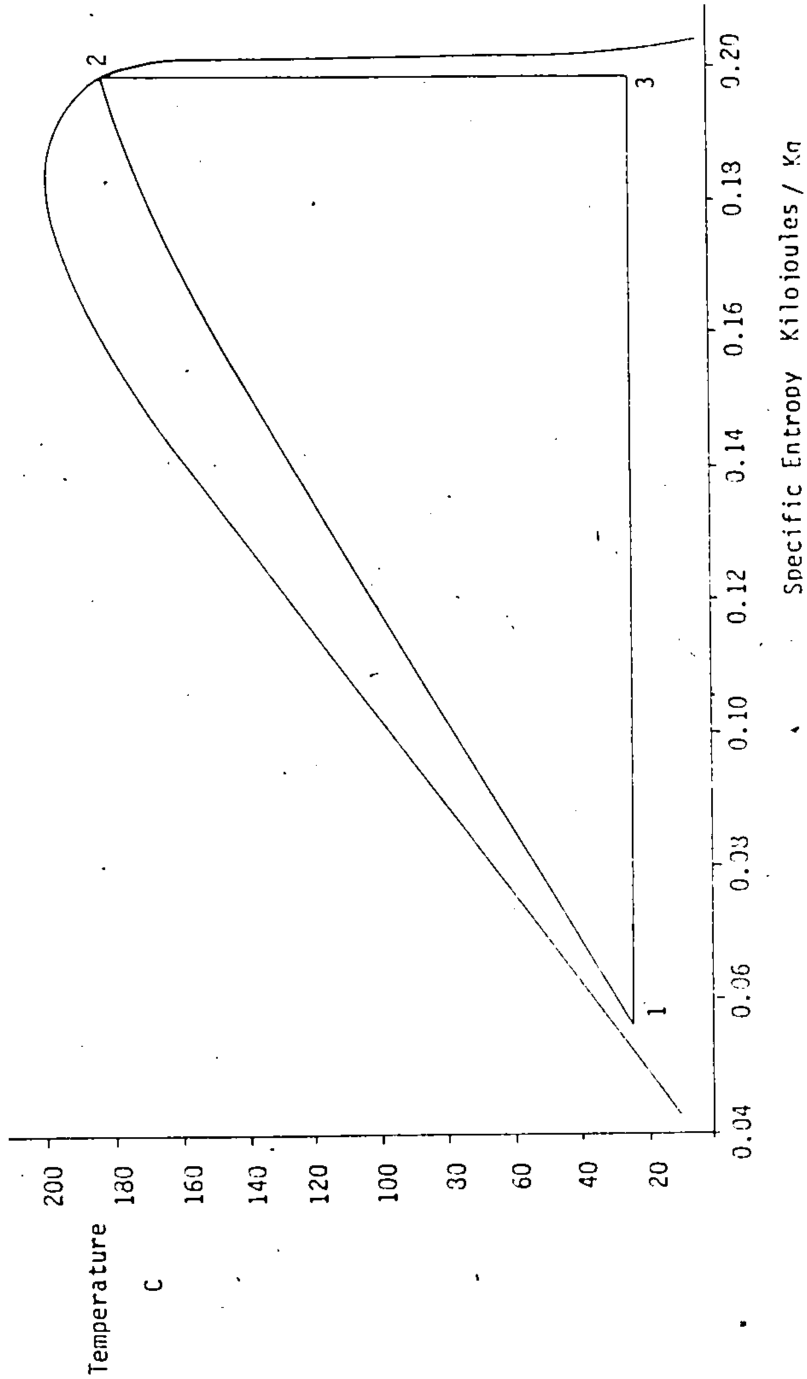

Figure 13. T-s Diagram

(Increased Volume Vessel, R-11)

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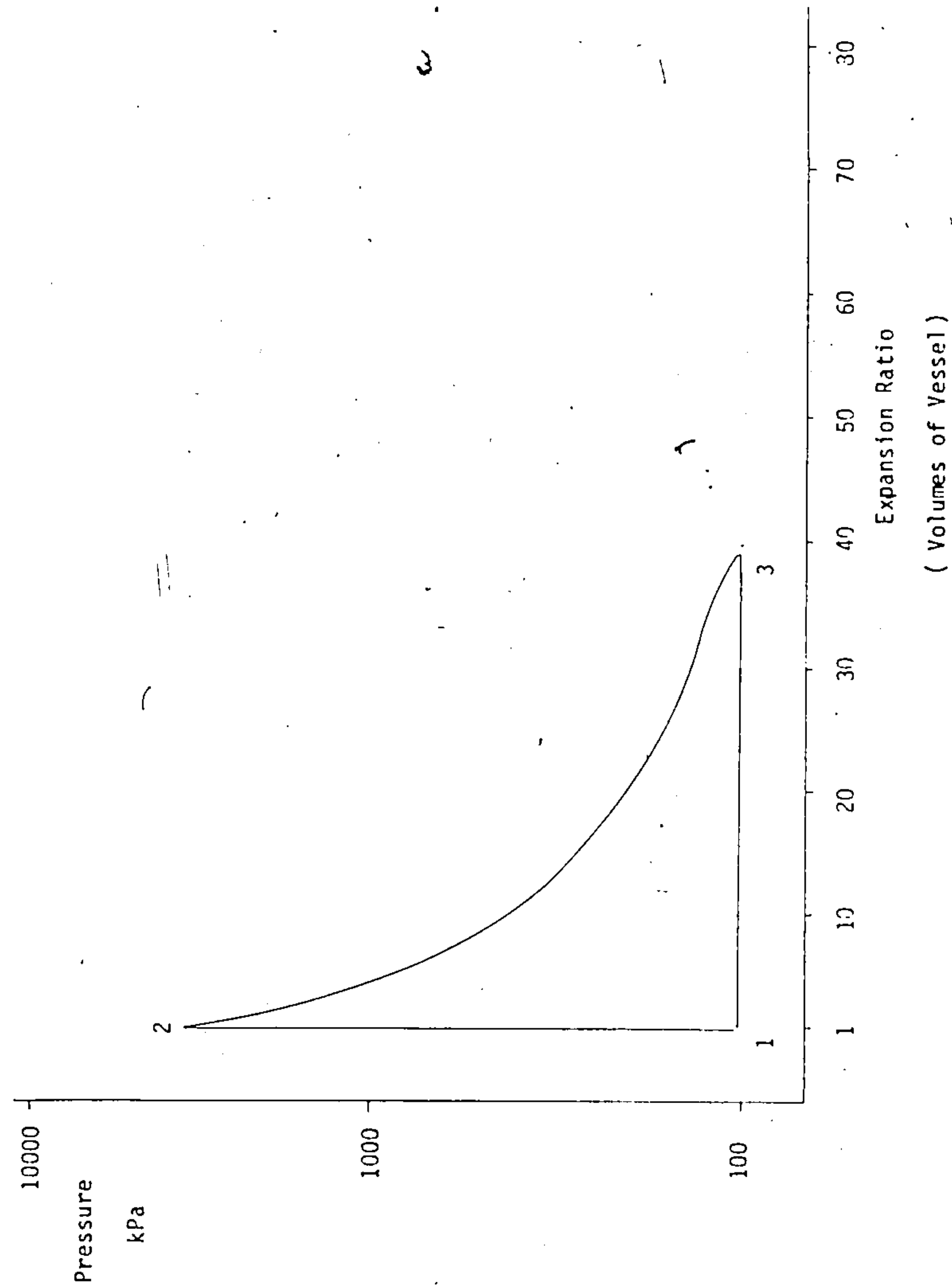

Figure 14. P-v Diagram

(Increased Volume Vessel, R-11)

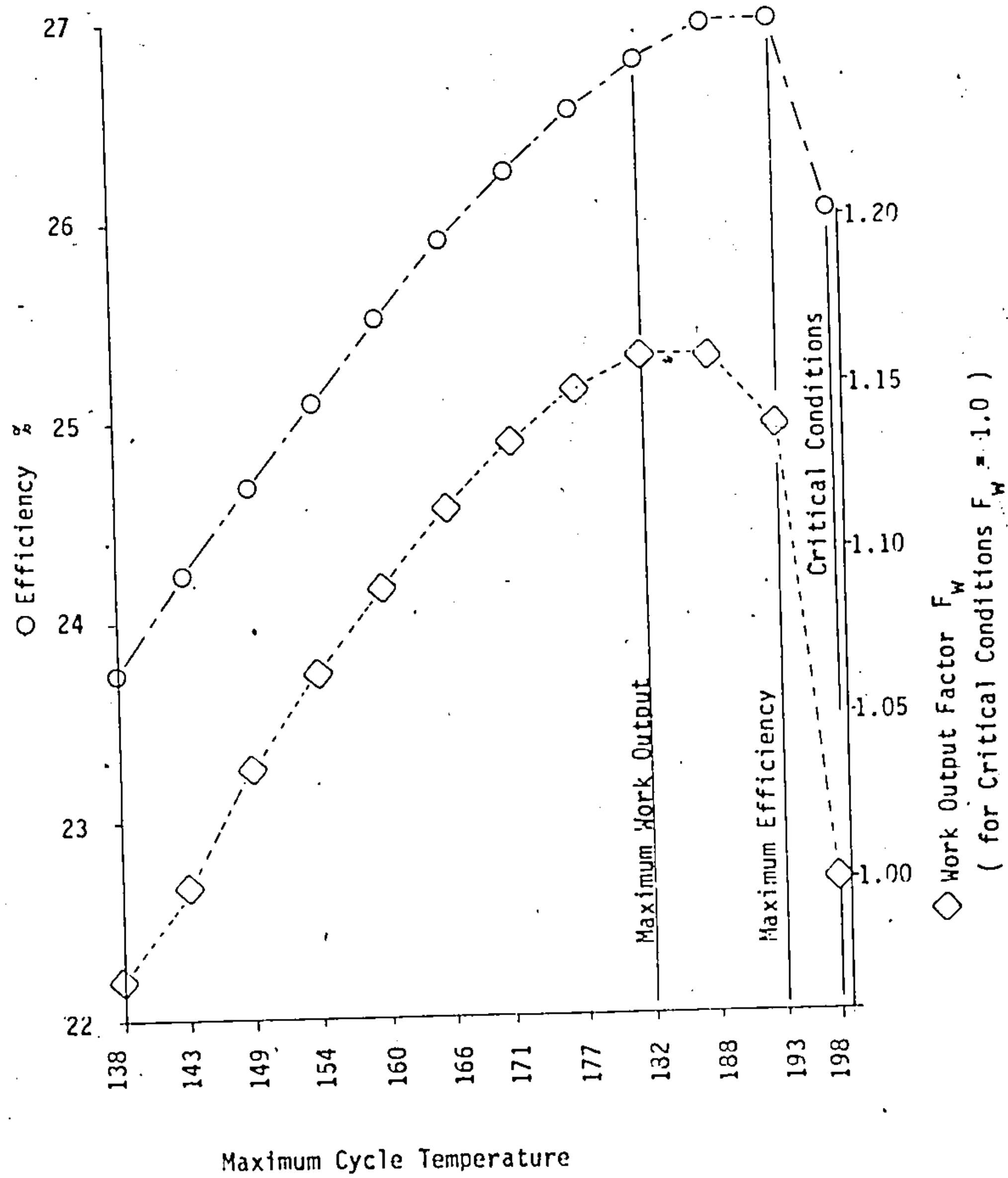

Figure 15. Efficiency & Work Output vs. Maximum Cycle Temperature

(Increased Volume Vessel, R-11)

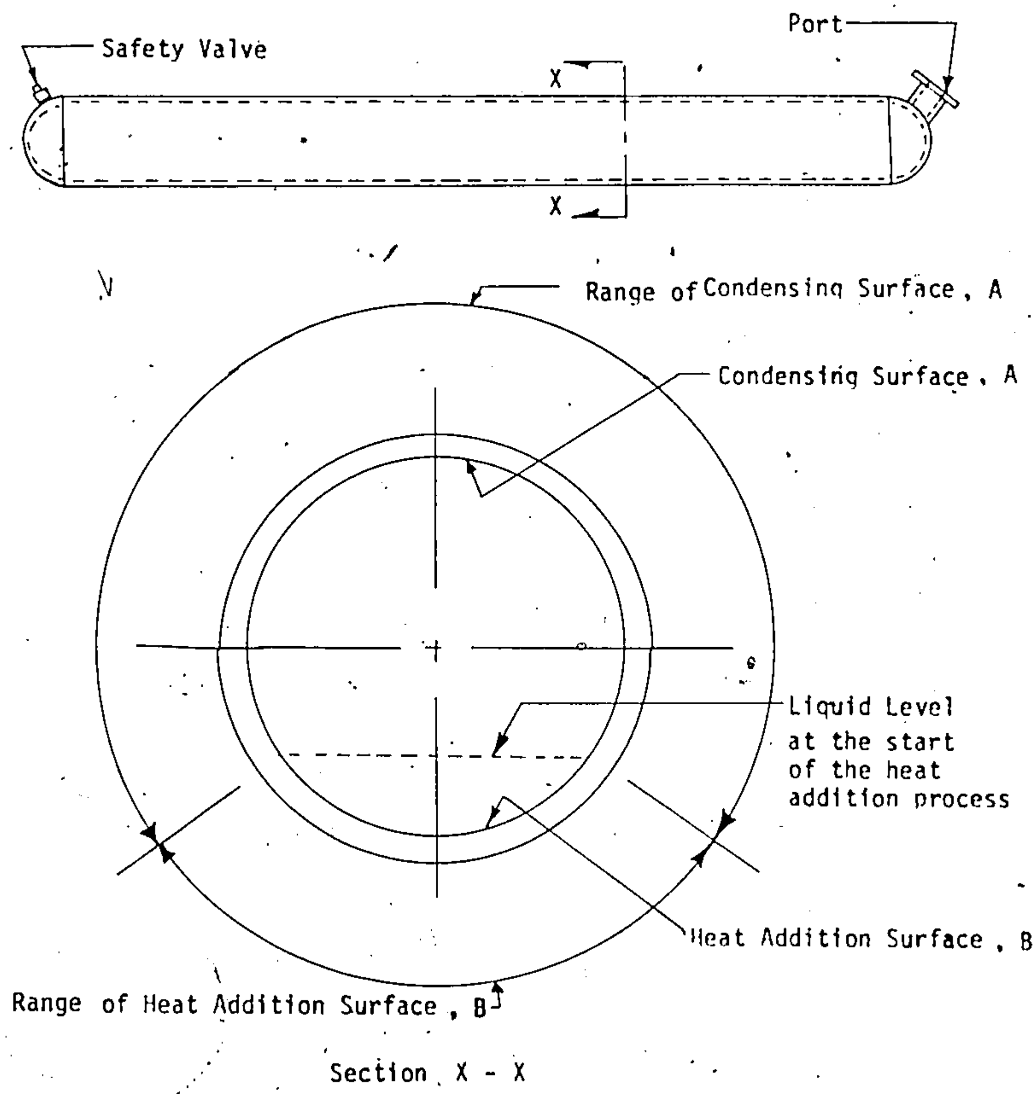

Figure 16. Alternate Vessel Design

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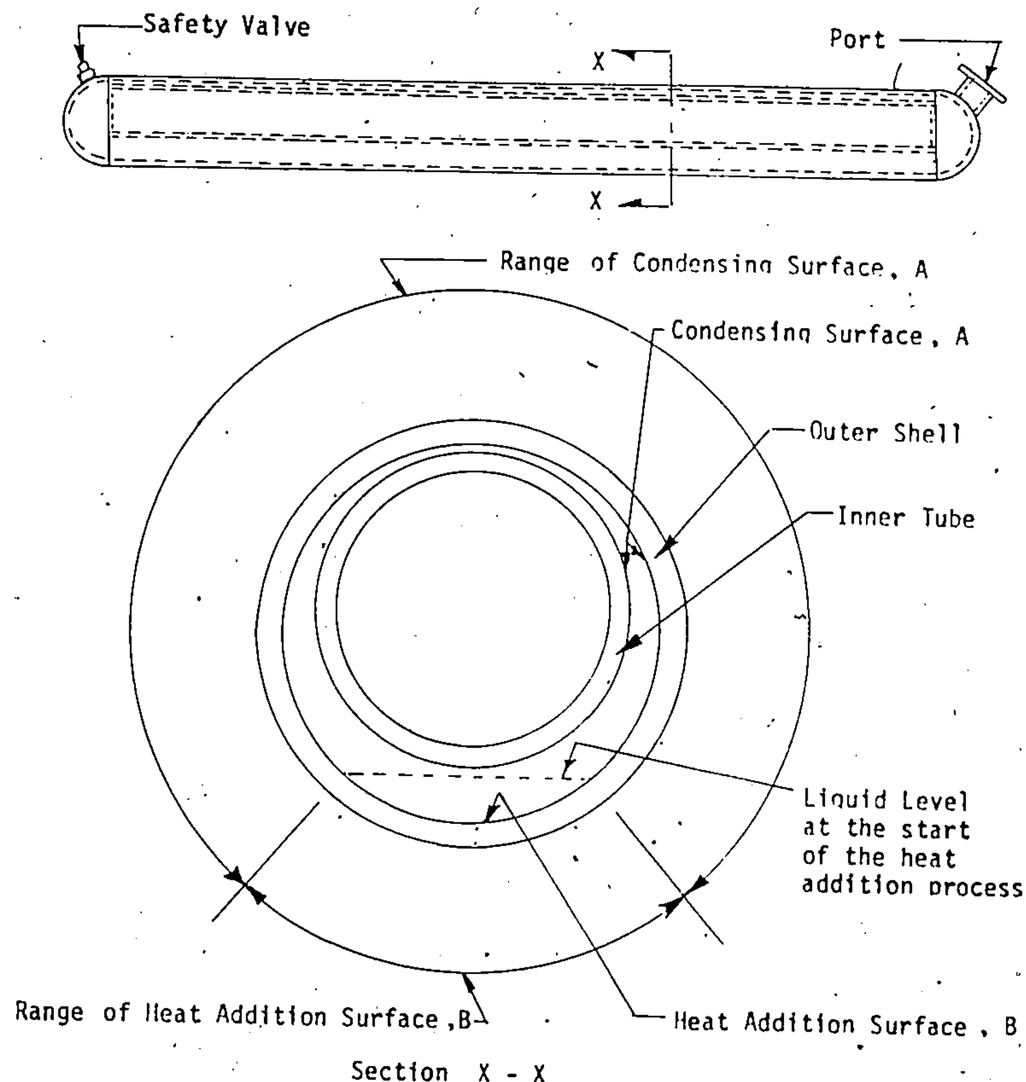

Figure 17. Alternate Vessel Design

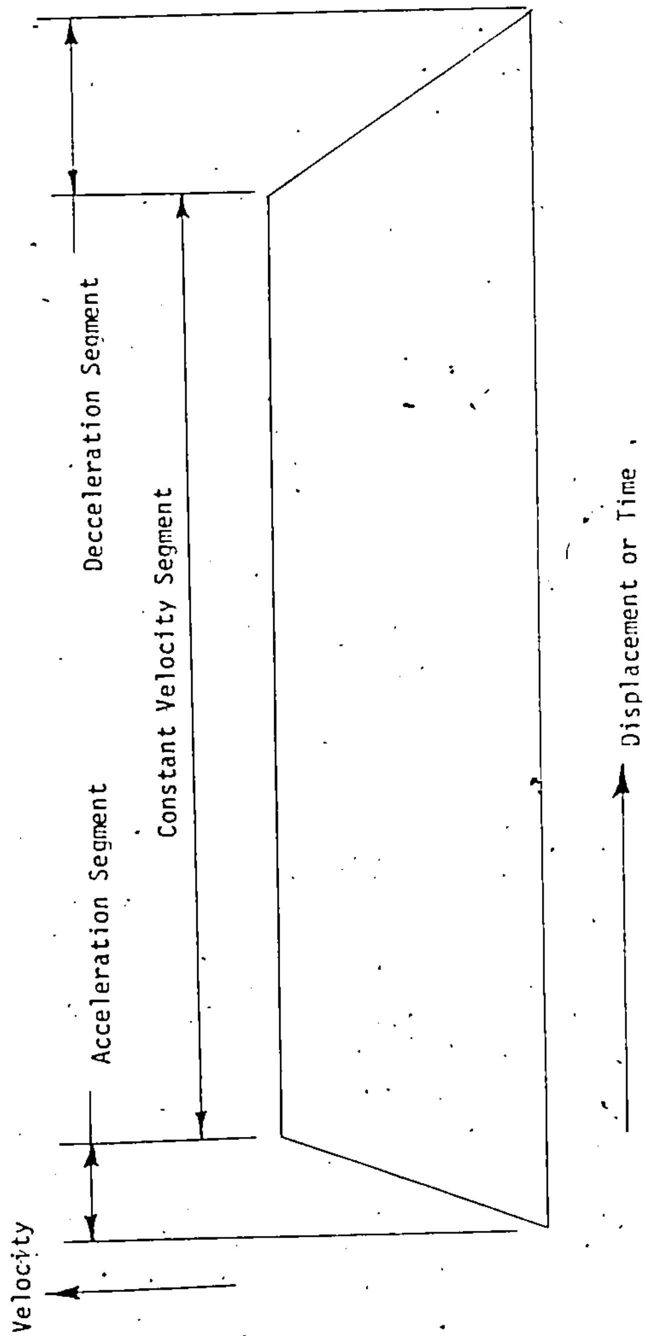

Figure 18. Velocity Diagram for the Piston

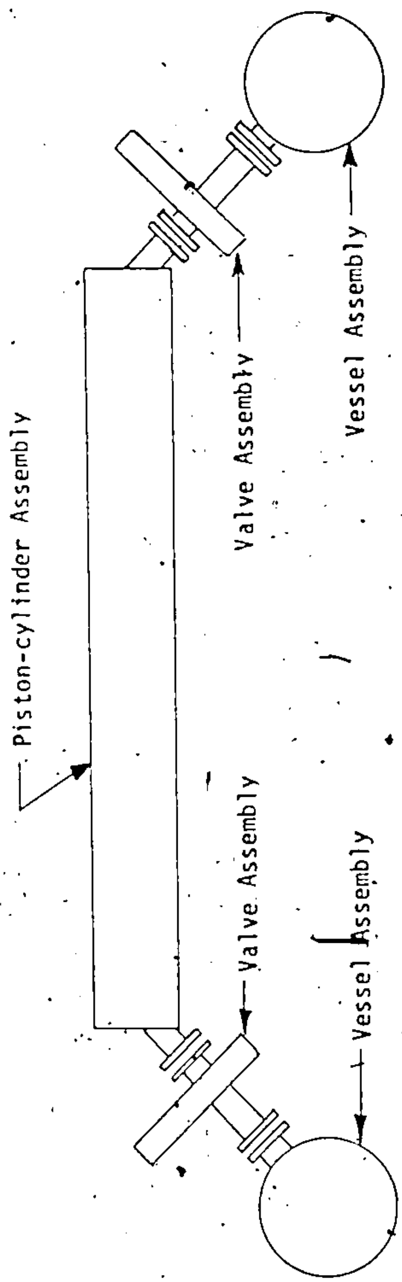

Figure 19. Alternate F.P.G. Design

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APPENDIX "A"

Program for Calculating

the Cycle of a

Free Piston Generator

with Isentropic Expansion

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INTRODUCTION

This program is designed to give an analysis of the cycle of the new machine. The processes used are a constant volume heat addition which is followed by an isentropic expansion and a constant pressure heat rejection. The cycle has been programmed to be operated on a small programmable hand calculator (TI-PROGRAMMABLE-59).

The cycle used is as shown in Figure A. Starting at point 2, the program will first calculate the quality of the fluid after expansion to point 3, using an isentropic process. From this information, it will then find the expansion ratio of the fluid, as well as, the enthalpy at this point. This will enable the machine to calculate the work output of the process as a change in the enthalpy of the fluid. Next it will calculate the quality of the fluid at point 1, assuming that the volume at this point is the same as at point 2. This will enable the calculator to find the entropy and enthalpy at point 1. With this information it can find the heat that must be added and the heat that must be rejected. Finally the efficiency of a cycle can be calculated. The information that is required for the calculations is the conditions of the fluid at point 2 and the data for the liquid and vapour phases (saturated) at the limit of expansion (line 1 to 3).

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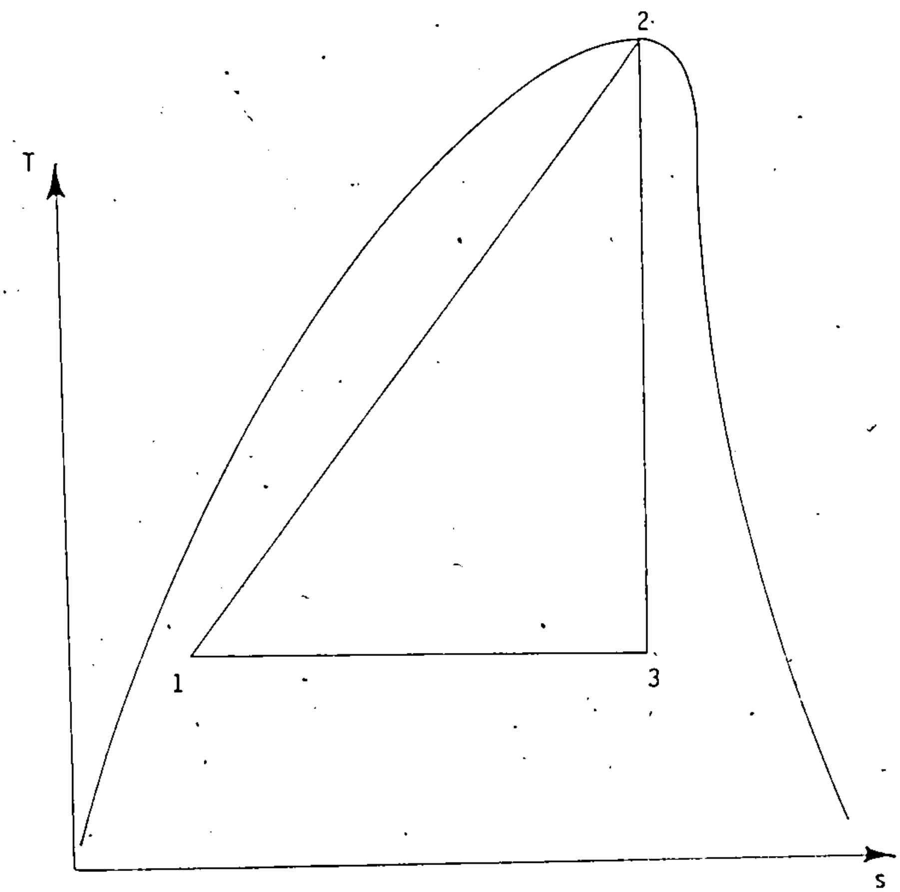

Figure A

T-s Diagram

Isentropic Free Piston Generator Cycle

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PROCEEDURE

STEP	ENTER		PRESS	DISPLAY
1	s_2	Entropy at point 2	B	---
2	v_{g2}	Specific Volume (gas) at point 2	B	---
3	h_2	Enthalpy at point 2	B	s_2
4	v_{g13}	Specific Volume (gas) for line 1-3	A	---
5	ρ_{f13}	Density (liquid) for line 1-3	A	---
6	h_{f13}	Enthalpy (liquid) for line 1-3	A	---
7	h_{g13}	Enthalpy (gas) for line 1-3	A	---
8	s_{f13}	Entropy (liquid) for line 1-3	A	---
9	s_{g13}	Entropy (gas) for line 1-3	A	v_{g13}
10	---		C	Efficiency
11	---		R/S	Quality at 3
12	---		R/S	Enthalpy at 3
13	---		R/S	Expansion Ratio
14	---		R/S	Quality at 1
15	---		R/S	Entropy at 1
16	---		R/S	Enthalpy at 1
17	---		R/S	Work Done
18	---		R/S	Heat Rejected
19	---		R/S	Heat Added
20	---		R/S	Efficiency

21 Repeat - steps 1 to 3 and steps 10 to 20 and/or
- steps 4 to 9 and steps 10 to 20 as required.

FLOW CHART

Enter Data for point 2	000–010	2nd LBL B 2nd EXC 00 2nd EXC 01 2nd EXC 02 RCL 02 R/S
Enter Data for line 1-3	011–027	2nd LBL A 2nd EXC 03 2nd EXC 04 2nd EXC 05 2nd EXC 06 2nd EXC 07 2nd EXC 08 RCL 08 R/S
Calculate Quality at 3	028–049	2nd LBL C RCL 03 - RCL 04 = STO 19 RCL 02 - RCL 04 = ÷ RCL 19 = STO 09
Calculate Enthalpy at 3	050–064	RCL 05 - RCL 06 = X RCL 09 + RCL 06 = STO 10
Calculate Expansion Ratio	065–087	RCL 08 X RCL 09 + RCL 07 1/x X (1- RCL 09) = ÷ RCL 01 = STO 11
Calculate Quality at 1	088–106	RCL 01 - RCL 07 1/x = ÷ (RCL 08 - RCL 07 1/x) = STO 12

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Calculate Entropy at 1	107–115	X RCL 19 + RCL 04 = STO 13
Calculate Enthalpy at 1	116–130	RCL 05 - RCL 06 = X RCL 12 + RCL 06 = STO 14
Calculate Work Done	131–138	RCL 00 - RCL 10 = STO 15
Calculate Heat Rejected	139–146	RCL 10 - RCL 14 = STO 16
Calculate Heat Added and Efficiency	147–161	RCL 15 ÷ (RCL 00 - RCL 14) STO 17 = STO 18
Data Out	162	
Efficiency		R/S
Quality at 3		RCL 09 R/S
Enthalpy at 3		RCL 10 R/S

Expansion Ratio	RCL 11 R/S
Quality at 1	RCL 12 R/S
Entropy at 1	RCL 13 R/S
Enthalpy at 1	RCL 14 R/S
Work Done	RCL 15 R/S
Heat Rejected	RCL 16 R/S
Heat Added	RCL 17 R/S
Efficiency	RCL 18 R/S

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ISENTROPIC FREE PISTON GENERATOR

CODING

LOCATION	CODE	KEY	COMMENTS
000	76	2^{nd} LBL	
001	12	B	Data Entry for point 2
002	48	2^{nd} EXC	
003	00	00	h_2 Enthalpy at point 2
004	48	2^{nd} EXC	
005	01	01	v_{g2} Specific Volume (gas) at point 2
006	48	2^{nd} EXC	
007	02	02	s_2 Entropy at point 2
008	43	RCL	
009	02	02	s_2 Entropy at point 2
010	91	R/S	
011	76	2^{nd} LBL	
012	11	A	Data Entry for line 1-3
013	48	2^{nd} EXC	
014	03	03	s_{g13} Entropy (gas) for line 1-3
015	48	2^{nd} EXC	
016	04	04	s_{f13} Entropy (liquid) for line 1-3
017	48	2^{nd} EXC	
018	05	05	h_{g13} Enthalpy (gas) for line 1-3
019	48	2^{nd} EXC	
020	06	06	h_{f13} Enthalpy (liquid) for line 1-3

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LOCATION	CODE	KEY	COMMENTS
021	48	2^{nd} EXC	
022	07	07	ρ_{f13} Density (liquid) for line 1-3
023	48	2^{nd} EXC	
024	08	08	v_{g13} Specific Volume (gas) for line 1-3
025	43	RCL	
026	08	08	v_{g13} Specific Volume (gas) for line 1-3
027	91	R/S	
028	76	2^{nd} LBL	
029	13	C	Start of Calculations
030	43	RCL	
031	03	03	s_{g13} Entropy (gas) for line 1-3
032	75	-	
033	43	RCL	
034	04	04	s_{f13} Entropy (liquid) for line 1-3
035	95	=	
036	42	STO	
037	19	19	Temporary Storage
038	43	RCL	
039	02	02	s_2 Entropy at point 2
040	75	-	
041	43	RCL	
042	04	04	s_{f13} Entropy (liquid) for line 1-3
043	95	=	
044	55	÷	

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LOCATION	CODE	KEY	COMMENTS
045	43	RCL.	
046	19	19	Temporary Storage
047	95	=	
048	42	STO	
049	09	09	Quality at 3
050	43	RCL	
051	05	05	h_{g13} Enthalpy (gas) for line 1-3
052	75	-	
053	43	RCL	
054	06	06	h_{f13} Enthalpy (liquid) for line 1-3
055	95	=	
056	65	X	
057	43	RCL	
058	09	09	Quality at 3
059	85	+	
060	43	RCL	
061	06	06	h_{f13} Enthalpy (liquid) for line 1-3
062	95	=	
063	42	STO	
064	10	10	h_3 Enthalpy at 3
065	43	RCL	
066	08	08	v_{g13} Specific Volume (gas) for line 1-3
067	65	X	
068	43	RCL	

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LOCATION	CODE	KEY	COMMENTS
069	09	09	Quality at 3
070	85	+	
071	43	RCL	
072	07	07	ρ_{f13} Density (liquid) for line 1-3
073	35	1/x	
074	65	X	
075	53	(	
076	01	1	
077	75	-	
078	43	RCL	
079	09	09	Quality at 3
080	54	)	
081	95	=	
082	55	÷	
083	43	RCL	
084	01	01	v_{g2} Specific Volume (gas) at point 2
085	95	=	
086	42	STO	
087	11	11	Expansion Ratio
088	43	RCL	
089	01	01	v_{g2} Specific Volume (gas) at point 2
090	75	-	
091	43	RCL	
092	07	07	ρ_{f13} Density (liquid) for line 1-3

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LOCATION	CODE	KEY	COMMENTS
093	35	1/x	
094	95	=	
095	55	÷	
096	53	(	
097	43	RCL	
098	08	08	v_{g13} Specific Volume (gas) for line 1-3
099	75	-	
100	43	RCL	
101	07	07	ρ_{f13} Density (liquid) for line 1-3
102	35	1/x	
103	54	)	
104	95	=	
105	42	STO	
106	12	12	Quality at 1
107	65	X	
108	43	RCL	
109	19	19	Temporary Storage
110	85	+	
111	43	RCL	
112	04	04	s_{f13} Entropy (liquid) for line 1-3
113	95	=	
114	42	STO	
115	13	13	s_1 Entropy at 1
116	43	RCL	

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LOCATION	CODE	KEY	COMMENTS
117	05	05	h_{g13} Enthalpy (gas) for line 1-3
118	75	-	
119	43	RCL	
120	06	06	h_{f13} Enthalpy (liquid) for line 1-3
121	95	=	
122	65	X	
123	43	RCL	
124	12	12	Quality at 1
125	85	+	
126	43	RCL	
127	06	06	h_{f13} Enthalpy (liquid) for line 1-3
128	95	=	
129	42	STO	
130	14	14	h_1 Enthalpy at 1
131	43	RCL	
132	00	00	h_2 Enthalpy at point 2
133	75	-	
134	43	RCL	
135	10	10	h_3 Enthalpy at point 3
136	95	=	
137	42	STO	
138	15	15	Work Done
139	43	RCL	
140	10	10	h_3 Enthalpy at point 3

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LOCATION	CODE	KEY	COMMENTS
141	75	-	
142	43	RCL	
143	14	14	h_1 Enthalpy at 1
144	95	=	
145	42	STO	
146	16	16	Heat Rejected
147	43	RCL	
148	15	15	Work Done
149	55	÷	
150	53	(	
151	43	RCL	
152	00	00	h_2 Enthalpy at point 2
153	75	-	
154	43	RCL	
155	14	14	h_1 Enthalpy at 1
156	54	)	
157	42	STO	
158	17	17	Heat Added
159	95	=	
160	42	STO	
161	18	18	Efficiency
162	91	R/S	
163	43	RCL	
164	09	09	Quality at 3

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LOCATION	CODE	KEY	COMMENTS
165	91	R/S	
166	43	RCL	
167	10	10	h_3 Enthalpy at 3
168	91	R/S	
169	43	RCL	
170	11	11	Expansion Ratio
171	91	R/S	
172	43	RCL	
173	12	12	Quality at 1
174	91	R/S	
175	43	RCL	
176	13	13	s_1 Entropy at 1
177	91	R/S	
178	43	RCL	
179	14	14	h_1 Enthalpy at 1
180	91	R/S	
181	43	RCL	
182	15	15	Work Done
183	91	R/S	
184	43	RCL	
185	16	16	Heat Rejected
186	91	R/S	
187	43	RCL	
188	17	17	Heat Added

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LOCATION	CODE	KEY	COMMENTS
189	91	R/S	
190	43	RCL	
191	18	18	Efficiency
192	91	R/S	

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MEMORY ALLOCATION

MEMORY	SYMBOL	COMMENTS
00	h_2	Enthalpy at point 2
01	v_{g2}	Specific Volume at point 2
02	s_2	Entropy at point 2
03	s_{g13}	Entropy (gas) for line 1-3
04	s_{f13}	Entropy (liquid) for line 1-3
05	h_{g13}	Enthalpy (gas) for line 1-3
06	h_{f13}	Enthalpy (liquid) for line 1-3
07	ρ_{f13}	Density (liquid) for line 1-3
08	v_{g13}	Specific Volume (gas) for line 1-3
09		Quality at point 3
10	h_3	Enthalpy at point 3
11		Expansion Ratio
12		Quality at point 1
13	s_1	Entropy at point 1
14	h_1	Enthalpy at point 1
15		Work Done
16		Heat Rejected
17		Heat Added
18		Efficiency
19		Temporary Storage