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A BIPHASE TURBINE BOTTOMING CYCLE
FOR A DIESEL ENGINE

LANCE HAYS

Prepared for the U.S. Energy Research and
Development Administration under Contract No. E(04-3)-1207

July 31, 1976

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PATENT CLEARED

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ABSTRACT

Application of a two-phase turbine system to waste heat recovery was examined. Bottoming cycle efficiencies ranging from 15-30% were calculated for a 720°F diesel exhaust temperature. A single stage demonstration unit, designed for non-toxic fluids (water and Dow-Therm A) and for atmospheric seals and bearings, had a cycle efficiency of 23%. The net output power was 276 HP at 8,100 rpm, increasing the total shaft power from 1,800 HP for the diesel alone, to 2,076 HP for the combined system. A four stage organic turbine, for the same application, had a rotational speed of 14,700 rpm while a four stage steam turbine had 26,000 rpm. Fabrication drawings were prepared for the turbine and nozzle. The major improvement leading to higher cycle efficiency and lower turbine rpm was found to be the use of a liquid component with lower sensible heat. A possible reduction in capital cost was found to be the use of a contact heat exchanger instead of tube-fin construction. A program resulting in the demonstration of a two-phase bottoming system was planned and the required cost estimated. The program would result in a feasibility test of the nozzle and turbine at the end of the first year, a laboratory performance test of the bottoming system by the end of the second year and a field demonstration test and laboratory endurance test of the bottoming system during the third year.

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I. INTRODUCTION

Effective utilization of waste heat is an energy conservation measure which can have a very large impact on fuel consumption in this country. Conversion of the waste heat into a useful form of energy (such as electricity or mechanical shaft power) has an added advantage in that the basic tooling and plant designs can be preserved. Furthermore, reduction of both air and thermal pollution is a direct consequence of increasing the useful power output for a given amount of fuel.

Many processes and devices have hot exhaust gases that would be suitable for the heat source for energy conversion systems. A particularly attractive class of devices are internal combustion engines, both diesel and spark ignition. These engines account for nearly one third of the country's energy consumption. Moreover, the available energy in the exhaust gases is equal in magnitude to the useful power output for the engine. Existing estimates of bottoming cycle performance have indicated improvements of 15-20%⁽¹⁾ in the fuel economy resulting in a large potential savings if all engines were so equipped. Figure 1 (after Ref. 2) shows an energy diagram for a typical diesel engine with latent heat cooling. The exhaust gas energy is 33% of the heat input compared to 39% of the heat input which appears as useful work output. This diagram also shows a potential recovery of about 20% as the heat value of steam raised by the exhaust gases. For typical steam systems this heat can be converted to shaft power to raise the total engine shaft output by about 5-7½%⁽²⁾.

The subject of this study, and the simplest application of this type, is utilization of the waste heat from a large, stationary diesel engine. Stationary engines don't have the same space and weight limitations, or control and transient operation requirements as vehicular power plants. Therefore, the steady state performance and reliability aspects of bottoming systems can be demonstrated without the complications necessitated by mobile applications.

Fig. 1 - Diagrammatic Representation of Heat Balance with Latent Heat Cooling for Diesel Engine

Current approaches for waste heat conversion include both closed cycle and open cycle heat engines. Only the closed cycle variations will be considered here. The steam Rankine cycle has been used commercially as a bottoming system for gas turbines ^(3,4). Systems of this type have reduced the heat rate from 13,500 B/kWh with a gas turbine alone to 9,600 B/kWh for the combined cycle. Thus, a fuel savings of 30% results from the addition of this bottoming system. Commercial systems have also been placed in operation using steam bottoming cycles to recover waste heat from diesel engines. Figure 2 is a schematic of a typical system described in Ref. 2. As discussed previously, this type of system with 240/280 psig, 600/650°F steam increases the shaft power output by 5½ to 7%.

Another approach to waste heat conversion is the use of a Rankine cycle with an organic working fluid. Ref. 5 discusses the use of an organic vapor turbine system to bottom a diesel engine. A "bread board" system tested resulted in a 12% increase in power without an increase in fuel consumption. An organic vapor expander was also used by Chapman ⁽⁶⁾ to bottom a spark ignition engine. In this case, an orbital vane expander was used as the power conversion component. Advantages resulting from the use of an organic vapor instead of steam include a greater thermodynamic efficiency and a somewhat lower turbine or expander rpm. Disadvantages include expensive fluid and seal requirements, non-availability of commercial components and the high capital cost of the system components, and the use of toxic working fluids (e.g. toluene and 'FS-85).

The subject of this design study is the use of a two-phase power system to convert the waste heat from a diesel engine to useful power. The two-phase power system while similar, has several potential advantages relative to a Rankine cycle bottoming system.

1. The turbine has a lower rpm than a vapor-alone turbine, simplifying transmission problems. For example, the single stage turbine of this design study has a shaft speed of 8,100 rpm compared to 14,700 for a four stage organic vapor turbine for the same application ⁽⁷⁾. A steam turbine with the same number of stages would have a rotation

Fig. 2 - Layout of Complete Engine Heat-Recovery System

speed of about 26,000 rpm. As discussed later the two-phase turbine rpm can be reduced further by increasing the mass ratio of liquid to vapor, but at the expense of pumping losses.

2. The two-phase cycle has a gas to liquid heat exchanger instead of a boiler and maximizes the exhaust gas energy available for conversion. The available energy to the system can be maximized because heat transfer is to the liquid phase. Figure 3 illustrates temperature profiles for typical steam and organic vapor boilers compared to the liquid heat exchanger of the two-phase turbine engine. Since the loss of available energy is proportional to the temperature difference it can be seen that a higher thermodynamic efficiency is possible with the two-phase cycle compared to the vapor cycle.
3. The turbine, seals, bearings and other items associated with the rotating unit are simple, low cost components. This is a consequence of the low rpm and the transformation of the energy from the gas to the high density liquid phase.
4. Variations of the two-phase power cycle exist which may allow the use of a contact heat exchanger for the exhaust gas, reducing the capital cost relative to a tube-fin heat exchanger.

Other possible advantageous features include higher ultimate efficiency, simplified control requirements, and good part load efficiency.

In order to determine the performance and design of a two-phase turbine, cycle calculations were conducted for several two-phase systems. A reference cycle was selected which represented a compromise between efficiency, rpm, and simplicity of construction. The two-phase nozzle and

Fig. 3 - Comparison of Temperature Profiles of Two-Phase Cycle Heat Exchanger and Organic Steam Vapor Generators

turbine were designed for the conditions of the reference cycle and fabrication drawings were prepared. Functional requirements of the major system components were specified and a program resulting in the demonstration of a two-phase bottoming cycle was formulated.

II. TWO-PHASE CYCLES

Several different two-phase bottoming cycles were considered for the design study. In each variation, the common feature is that heat transfer occurs from the exhaust gas to the liquid phase. However, the working fluid may consist of one or more chemical species, staging can be used, and a contact heat exchanger can be used in place of a conventional tube-fin heat exchanger.

In this section, the basic principles of two-phase nozzles and turbines are discussed, followed by a brief description of each cycle.

TWO-PHASE NOZZLES AND TURBINES

The two elements which are responsible for the unique characteristics of the two-phase turbine engine are the two-phase nozzle and the turbine wheel. The nozzle produces efficient conversion of liquid enthalpy to vapor enthalpy to vapor and liquid kinetic energy. This is achieved by mixing a finely dispersed hot liquid of low vapor pressure (such as oil) with a liquid possessing higher vapor pressure (such as water or Freon). The water vaporizes to steam at high pressure. The steam is subsequently expanded to low pressure and high velocity, dragging the oil droplets to a high velocity also. The presence of large amounts of finely dispersed liquid has a similar effect on the nozzle exit velocity as a large increase in molecular weight of the vapor would, i.e. a high mass flow rate and low spouting velocity are obtained. Figure 4 shows the nozzle exit velocity as a function of the ratio of liquid to gas. Starting at a value of about 4,000 ft/s for all gas, the exit velocity is reduced to less than 1,000 ft/s for a mass ratio of 30. Thus, a much lower turbine wheel speed can be

Fig. 4 - Exit Velocity of Two-Phase Nozzles

used for maximum efficiency than for the vapor alone.

The relation of the nozzle and turbine wheel for a single stage, impulse circumferential configuration is shown in Figures 5 and 6. The geometry can be very similar to a simple Pelton Wheel. The only difference is that the initial impingement zone is inclined at a more gradual angle than the Pelton Wheel, to avoid two-phase impact losses; and the surface area can be made less than that of a Pelton Wheel because of the concentration of the liquid phase after impingement. Another geometry which can be used is very similar to a conventional axial flow impulse turbine.

A variation of the above turbine which offers elimination of the primary oil pump and may have a higher efficiency for some applications, is a rotary separator turbine. Here the two-phase mixture is impinged on a rotating separator surface which results in a high velocity liquid annulus. The liquid flows into a radial inflow turbine, which in its simplest case is a rotating pitot tube and diffuser. The liquid is discharged through the base and enough pressure is retained to enable circulation through the heat exchanger to the nozzle. Another version is the use of a simple "U-tube" to recover the liquid energy.

Figures 7 and 8 illustrate this turbine design, consisting of nozzle(s), rotating separator, and liquid turbine. The liquid and gas mixture comprising the working fluid is supplied at high pressure to the nozzle inlets. The mixture expands to low pressure at the nozzle exits, and the resulting high-velocity two-phase jets impinge on the inner surface of the rim of the rotating separator. The liquid becomes concentrated in a layer on the inner surface due to the inertia of the liquid and centrifugal force while the gas flows radially inward through passages and enters the gas discharge pipe through ports in the stationary housing. The rotating separator is supported by bearings mounted in the housing.

Fig. 5 - Schematic of Flow in Two-Phase Impulse Turbine

Fig. 6 - Schematic of Two-Phase Turbine Wheel and Blading

Fig. 7 - Separator-Turbine with Radial Inflow Liquid Turbine

Fig. 8 - Separator-Turbine with U-Tube Liquid Turbine

The rotation of the separator is impeded only by windage and bearing friction losses, which can be very small. Thus, only a very small relative velocity between the impinging jets and the surface, aided by the torque imparted to the rotating separator by the inward flow of the gas through passages, serves to maintain the speed of the liquid layer at a value nearly equal to that of the jets.

The liquid flows from the liquid layer through passages in the rotating separator into an annular chamber which forms an integral part of the separator wheel. As a result another liquid layer is formed, held against the surface by centrifugal force. This layer furnishes the fluid energy source for the turbine rotating concentrically within the separator wheel and having turbine inlet passages immersed in the liquid layer.

The turbine may be of any conventional type where the blades or passages can be arranged to intercept the liquid layer. Figure 7 shows a radial flow type of turbine. The turbine wheel rotates at a lower speed than the separator wheel causing liquid from the layer to enter the inlets, flow radially inward through passages, and flow to the liquid outlet pipe through apertures in the wall of the turbine shaft. The shaft is connected to the load to be driven. The turbine is supported on bearings.

The turbine passage can optionally incorporate a diffuser in which the velocity of the liquid entering the inlet can be partially converted to pressure such that, even allowing for the pressure drop in the radial passages due to centrifugal force, the liquid pressure in the discharge pipe is substantially higher than the pressure in the turbine casing and, in fact, greater than the pressure at the nozzle inlets. Thus, the diffusers can supply the necessary pumping of the liquid, eliminating the need for a separate pump to return the liquid to the nozzles.

For operation with high pressure at the discharge, the leakage of liquid between the shaft and the housing can be reduced by labyrinth seals and drains which return liquid leakage to the bottom of the housing where the liquid from this and other internal leakage sources is picked up by slinger blades and thrown back into the jets. Leakage to the outside of housing is prevented by a shaft seal.

SINGLE COMPONENT - SINGLE STAGE

The single component - single stage system is shown in Figure 9. The waste heat from a diesel engine (or gas turbine) enters the heat exchanger at temperature T_i with an exhaust flow rate of $\dot{m}_e$. The exhaust heat of an amount Q_i is transferred to a single phase liquid in a counter current heat exchanger. The exhaust gas exits at temperature T_e . The heated liquid is flashed in a nozzle and expanded to pressure p_2 from pressure p_1 . The saturated mixture of vapor and liquid with a quality x_2 impinges on a rotary separator. The vapor flows through a radial inflow turbine and the liquid flows through a radial inflow turbine, each on a common shaft, producing a net power output P_s . The vapor exhausts to the condenser where an amount of heat of Q_c is rejected. The condensate is pressurized and merged with the high pressure outlet from the liquid turbine. The liquid flows back to the heater, closing the cycle.

Many variations on this basic scheme are possible. For example, the vapor turbine and/or liquid turbine can be impulse type machines with an additional pump to pressurize the liquid. The main advantage lies in the simpler turbine construction, but of course a disadvantage is the presence of another component and a slight loss in efficiency.

SINGLE COMPONENT - MULTISTAGE

A multistage version of the single component cycle is shown in Figure 10.

Fig. 9 - Single Component, Two-Phase Bottoming Cycle (Single Stage)

Fig. 10 - Single Component - Multistage Bottoming Cycle

In this variation, the exit flow from the first stage is remixed and passed through a regenerative heat exchanger, condensing part of the first stage vapor. The mixture is expanded through the second stage nozzle and turbine (as in the single stage cycle). The condensate and liquid discharge from the final (second) stage is passed through the regenerative heat exchanger increasing the temperature to the minimum dictated by combustor material considerations. The advantage of staging is that lower shaft rpm results and the lower velocities result in more efficient nozzle and turbine operation. Against these gains must be balanced the added complexity and possible interstage losses.

TWO COMPONENT - CONVENTIONAL HEAT EXCHANGER

The use of two components allows more flexibility in the choice of shaft rpm. A single stage two-component cycle is shown in Figure 11.

In this cycle the thermodynamic working fluid is component A. It is supplied to the nozzle as a liquid and flash vaporizes upon contact with the heated liquid, component B. Another variation of the two-component cycle is to mix the condensate (component A) with the turbine discharge (component B) circulating the mixture through the heater. This variation has the same cycle diagram as the single-component variation.

In either case, the vaporizing component gradually vaporizes in the nozzle as the pressure is reduced. Any degree of superheat can be provided by using large ratios of component "B" to "A", however, in the limit of very large mass ratios, vaporization occurs at a nearly constant temperature which produces the same thermodynamic limitation as a Rankine cycle.

The resultant two-phase mixture impinges on the rotary separator. Component A, the vapor, flows through a radial inflow gas turbine. Component B drives either a radial inflow or impulse liquid turbine. The

Fig. 11 - Two-Component, Two-Phase Bottoming Cycle with Tube-Fin Heat Exchanger

vapor, after leaving the gas turbine flows to a regenerator (if superheated) and to the condenser. After rejecting the waste heat from the cycle the condensate is pressurized and returned to the nozzle inlet or to the primary heat exchanger. The liquid "B" leaving the liquid turbine is pressurized either by a pump or by conversion of the liquid kinetic energy to head and subsequently return to the primary heat exchanger.

TWO COMPONENT - CONTACT HEAT EXCHANGER

The final variation to be considered uses two components in a manner to enable the use of a contact heat exchanger for the waste gases. This version is illustrated by the schematic of Figure 12.

The low vapor pressure liquid, component B, is sprayed into a vessel. The resulting droplets contact the hot exhaust gases which flow counter-current to the falling droplets. The heated liquid is collected at the bottom of the vessel and pumped to the nozzle inlet. The cooled exhaust gas is exhausted (possibly through a mist eliminator) to the atmosphere. The remainder of the cycle is identical to the version with a tube-fin heat exchanger. The heat-exchanger construction is simpler with the high-pressure, tube-fin structure being replaced by a low pressure vessel with spray nozzle. In addition, the physical dimensions of the heat exchanger will be smaller because of the large surface area of the droplets and absence of fin losses. The pump power P_{p2} will be about the same required by the diffuser or pump in the cycle of Figure 11, so the efficiency of the two cycles should be similar.

In addition to the basic variations discussed above, other cycle combinations can be used. For example, Figure 12a shows the use of a two-component two-phase cycle to bottom a two-phase cycle. By operating the two-component cycle with a high ratio of "B" to "A" (liquid to gas),

Fig. 12 - Two-Component, Two-Phase Bottoming Cycle with Contact Heat Exchanger

Fig. 12a - Use of Multiple Two-Phase Bottoming Cycles

heat addition occurs at nearly a constant temperature resulting in effective use of the waste heat from the condenser. This waste heat is rejected at a nearly constant temperature as opposed to the exhaust gas waste heat which occurs over a wide temperature range. In general, the choice of cycle will depend upon the heat source temperature, power level and other factors to be discussed later.

TEMPERATURE - ENTROPY DIAGRAM

The temperature - entropy diagram for both single and two component cycles is shown in Figure 13. The temperature variation of the exhaust gas for the diesel engine is illustrated by path A. The temperature decreases from 719°F to 242°F . In a single component water system this exhaust heat is transferred to pressurized water following path E. The water is expanded in the two-phase nozzle following path B, if the process were isentropic, and path C if real flow effects are included. The exit of the nozzle consists of a mixture of steam and water with high kinetic energy. This mixture drives the two-phase turbine. The steam is condensed (path D), pressurized and remixed with the high pressure water leaving the turbine. An alternate method is to extract all of the liquid kinetic energy in the turbine and pressurize that liquid with the condensate pump. As can be seen, the two-phase bottoming cycle maximizes the energy available from the waste heat since the upper temperature of the bottoming cycle continuously increases to match the decrease of the heat source temperature.

In the case of a two-component cycle the thermodynamic working fluid (steam) follows path F during the expansion. The reason the entropy increase is so large is the heat transfer from the all-liquid component into the steam. The net result is that the steam (for the reference cycle) is slightly superheated at the nozzle exit. In the case of the reference cycle, after separation of the steam from the liquid, it is further expanded from the nozzle exit pressure, 14.7 psia, to a lower

value, 1.2 psia, through a radial inflow gas turbine, producing added energy. This portion of the cycle diagram is path G. The steam is then condensed along path H, pressurized and remixed with the liquid from the liquid turbine. The heat input to the steam during the nozzle expansion is derived from the hot liquid which follows path J. This curve shows the entropy decrease per unit mass of component "B". However, since there are "r" lbs. of "B" per lb of "A", the total entropy change is given by path K. As can be seen, the areas under curves K and F are equal equating the heat transfer from the liquid into component B into component A during the expansion process.

TEMPERATURE, °F

1000

900

800

700

600

500

400

300

200

100

0

.2

.4

.6

.8

1.0

1.2

1.4

1.6

1.8

2.0

2.2

ENTROPY, Btu/lb °F

Fig. 13 - Temperature - Entropy Diagram for Single Component and Two-Component Two-Phase Cycles

III. CYCLE PERFORMANCE

Performance of the different cycle variations was determined using the two-phase turbine relations of Appendix A and the cycle equations of Appendix B. These calculations were performed for parametric variation of nozzle efficiency, turbine efficiency, rejection temperature and other variables of the power system. The final cycle selected was analyzed in somewhat greater detail as summarized in Section 4.

SINGLE COMPONENT - SINGLE STAGE

The single component cycle of Figure 9 was analyzed using water as the working fluid. Other possible working fluids considered were Dow-Therm A and E (Dow Chemical), Genetron 133a (Allied Chemical), CP-32 (Monsanto Co.), CP-34 (Monsanto Co.), the Freon family (Du Pont), Trichloroethylene (Dow Chemical), and mercury.

Water obviously has an advantage over the other fluids in terms of cost and the availability of equipment. Consideration of the thermodynamic properties showed most organics to be unsuited for a single component system because of the positive slope of the vapor saturation line. Expansion from pure liquid at a high enough temperature to be of interest for this application (exhaust temperature = 719°F) leads to a superheated vapor rather than a two-phase mixture. This is illustrated in Figure 14 where nozzle expansion for water is compared to a nozzle expansion for CP-32 (pyridine). The starting conditions are 600°F . With an expansion to 200°F , the water has a vapor quality of about 0.35 while the pyridine end point is far to the right of the saturated vapor line in the superheated region.

Mercury, on the other hand, is a very good fluid for a two-phase system. The high ratio of latent heat of vaporization to sensible heat of the liquid ($\approx 4,000^{\circ}\text{F}$) means a large ratio of liquid to vapor results from an

Fig. 14 - Comparison of Two-Phase Nozzle Expansion for Pyridine and Water

expansion. For example, expansion from liquid at 740°F to 240°F results in a vapor quality of only about .10, giving a liquid/vapor ratio of 10. Expansion of water ($L_v/C_p \approx 1,000^\circ\text{F}$) with a similar temperature drop gives a vapor quality of about .50, for a liquid to vapor ratio of only one. Thus, the spouting velocity of the mercury nozzle (aside from the obvious molecular weight differences) would be much less with a higher mass flow rate and the corresponding two-phase turbine rpm would be much less. Of course, mercury is undesirable as a working fluid because of its cost and toxicity. In addition, the vapor pressure is too low for the application of interest. However, future working fluid searches for single components should concentrate on candidates with a high L_v/C_p ratio.

The results of calculations for a single stage two-phase bottoming cycle using water are shown in Figures 15 thru 17. In Figure 15, the net cycle efficiency is plotted versus condensing temperature for several values of maximum cycle temperature. A peak efficiency of about 28% is reached at the maximum cycle temperature of 700°F and the condensing temperature of 90°F. The efficiency is very sensitive to the minimum temperature, decreasing from a value of 28% at 90°F to about 21% at 212°F (rejection temperature). Changing the peak temperature from 700°F to 640°F at a condensing temperature of 90°F, reduces the efficiency from 28% to about 25%. This change is significant because it means dropping the peak pressure from 3100 psia to 2100 psia with attendant lessening of equipment requirements.

These calculations were all performed for a turbine efficiency of .85, a nozzle efficiency of .85, a pump efficiency of .75 and a diffuser efficiency of .85. The mass ratio of course is a dependent variable as the cycle conditions change.

In order to examine the influence of these component efficiencies (cf. Appendix B) on cycle efficiency, a reference cycle was chosen at

Fig. 15 - Efficiency of Single Stage - Single Component Cycle Versus Condensing Temperature and Maximum Cycle Temperature (Water Working Fluid)

Fig. 16 - Influence of Turbine and Nozzle Efficiencies on Cycle Efficiency ($T_1 = 660^\circ\text{F}$, $T_2 = 212^\circ\text{F}$)

Fig. 17 - Influence of Diffuser and Pump Efficiencies on Cycle Efficiency ($T_1 = 660^\circ\text{F}$, $T_2 = 212^\circ\text{F}$)

$T_1 = 660^\circ$ and $T_2 = 212^\circ\text{F}$. The reference cycle efficiency was about 19% at this point with the above component efficiencies. As seen in Figure 16, variations in nozzle efficiency or turbine efficiency about the design value of .85, produce a change of about 1 percentage point in cycle efficiency for every 5 percentage points change in component efficiency. The maximum nozzle efficiency previously measured (.88, Ref. 8), would produce a cycle efficiency of about 19½%. The same maximum nozzle efficiency with the maximum Pelton Wheel efficiency reported ⁽⁹⁾ would give a cycle efficiency of 21%, a 12% relative increase above the design point. On the other hand, a nozzle or turbine efficiency of only .70 would reduce the cycle efficiency to 16%, which would still be a tolerable value.

Figure 17 shows the cycle efficiency to be less dependent on diffuser and pump efficiency. In this chart, the cycle efficiency changes by only about 1/10 percentage point for every 5 percentage points change in pump or diffuser efficiency. This is particularly significant in the case of the diffuser where the ingestion of vapor can cause efficiency losses.

A key problem with a single component water cycle is the high nozzle exit vapor quality, and hence, high spouting velocity. Nozzle spouting velocities range from 2,600 ft/s for 212°F-condensing temperature, to 3,500 ft/s for a 90°F-condensing temperature. These high values negate one of the original advantages of a two-phase cycle - the lower turbine rpm. For a two foot diameter turbine, an rpm range of 12,000 to 17,000 would result. For a Pelton or axial flow turbine the relative liquid impact velocities in the lower range are within turbine current practice (1,300-1,400 ft/s, Ref. 10), but extreme erosion could be expected to occur at the upper limit of 1,750 ft/s. Thus, it would appear that a lower velocity, such as could be obtained by staging or the use of two components, would be desirable.

SINGLE COMPONENT - MULTISTAGE

In order to determine the benefits of staging, the two stage cycle shown in Figure 10 was analyzed and compared to the single stage version. The maximum temperature was chosen as 680°F and the rejection temperature was taken as 90°F. A single stage system analyzed for these conditions gave an efficiency of 27% with a nozzle spouting velocity of 3,240 ft/s. For a two foot wheel, this would result in an rpm of 15,500, clearly too high.

Assuming an equal pressure ratio for each stage of the two stage system, with a 10 psi pressure drop between stages, gives an inlet pressure of 40 psia to the second stage. Table 1 summarizes the results of the nozzle calculations using these operating parameters.

TABLE 1
NOZZLE PARAMETERS OF TWO STAGE, SINGLE COMPONENT SYSTEM

Stage	Inlet Pressure (psia)	Exit Pressure (psia)	Inlet Quality	Exit Quality	Spouting Velocity (ft/s)
1	2710	50	.0	.42	2250
2	40	.7	.27	.33	1915

Using these values, the total cycle efficiency becomes 28% (the slight increase over the single stage is due to the fact that the second stage nozzle is operating at more nearly constant mass ratio). If the pressure ratios were adjusted slightly, to give equal exit velocities, the resulting spouting velocity for the first and second stages would be 2,090 ft/s. For a two foot diameter turbine, an rpm of 9,980 would result. Thus, the beneficial effect of increasing from a single stage to a two stage system would be reduction of the turbine rpm from 15,500 to 9,980 while increasing the cycle efficiency by approximately one percentage point, from 27% to 28%. The disadvantage is the addition

of a second nozzle and turbine. To a first approximation, the rpm is inversely proportional to the square root of the number of stages. Thus, to reduce the rpm from n_1 to n_2 , a total of $(n_1/n_2)^2$ stages would be required.

TWO COMPONENT - SINGLE STAGE

The nozzle exit velocity can also be reduced by the use of a second component "B", having a lower sensible heat than the liquid phase of the thermodynamic working fluid "A". In order to retain the advantages of a gas to liquid heat exchanger, the amount of the second component must be chosen to give the proper temperature variation in the nozzle and heat exchanger (cf. Appendix B). If too high a flow rate of "B" is used, the nozzle and heat exchanger will approach isothermal operation - the same as if "A" were being vaporized in the heat exchanger. This will reduce the cycle efficiency. If too low a flow rate of "B" is used, the liquid to gas mass ratio will be lower than desired - giving a high nozzle spouting velocity (if the heat source were at a constant temperature, then the mass ratio of "B" to "A" could be freely varied to give the exact nozzle spouting velocity and turbine rpm desired).

Two-component two-phase bottoming systems were analyzed using water as the thermodynamic working fluid and Dow-Therm A as the liquid component. Several combinations were examined but the water--Dow-Therm combination had the highest ratio of the latent heat of vaporization to sensible heat of those examined. As will be seen later, the nozzle spouting velocities and turbine rpm are higher than desirable. If a higher molecular weight-thermodynamic working fluid could be found, the turbine rpm could be reduced accordingly.

In order to illustrate the importance of the above parameters, consider a heat source from which it is desired to increase the cycle temperature

from T_2 to T_1 . For an isentropic nozzle expansion we have:

$$\dot{m}_a s''_{a1} + \dot{m}_b s''_{b1} = \dot{m}_a s'_{a2} + \dot{m}_b s''_{b2} \quad (1)$$

where $\dot{m}_a$ = mass flow rate of component A

$\dot{m}_b$ = mass flow rate of component B

s''_{a1} = entropy of component A liquid at the nozzle inlet

s''_{b1} = entropy of component B liquid at the nozzle inlet

s'_{a2} = entropy of component A vapor at the nozzle exit

s''_{b2} = entropy of component B liquid at the nozzle exit

$\dot{m}_b$ is selected so that component A is all vaporized and is at saturated conditions at T_2 . Solving for the mass ratio $r = \dot{m}_b / \dot{m}_a$:

$$r = \frac{s'_{a2} - s''_{a1}}{s''_{b1} - s''_{b2}} = \frac{s'_{a2} - s''_{a2} + s''_{a2} - s''_{a1}}{s''_{b1} - s''_{b2}} \quad (2)$$

or

$$r = \frac{\frac{L_{va2}}{T_2} + C''_{pa} \ln \frac{T_2}{T_1}}{C''_{pb} \ln \frac{T_2}{T_1}} \quad (3)$$

where s''_{a2} = entropy of component A liquid at the nozzle exit

L_{va2} = latent heat of vaporization of A

$C''_{pa,b}$ = specific heat of A,B (B/lb^oF)

Thus, larger L_{va} and smaller C''_{pb} give higher values of mass ratio, which reduces the nozzle velocity.

The two component cycle using water and Dow-Therm was analyzed for variation of the following parameters:

- Peak Cycle Temperature 500^oF to 700^oF
- Rejection Temperature 90^oF to 212^oF
- Diesel Exhaust Temperature 520^oF to 720^oF
- Diesel Exhaust Flowrate 15,300 lb/h to 27,300 lb/h
- Working Fluids (C_p variation) steam & Freon thru polyphenyl ether

f. Liquid/Gas Mass Ratio	3.5 to 23.0
g. Output rpm	5,000 to 12,000

In the case of the mass ratio and output rpm, independent variation was not always possible because of the interrelation of variables.

The effect of the peak cycle temperature is shown in Figure 18. Operation with a condensing pressure of .7 psia (90°F) and a nozzle exit liquid temperature of 212°F , gave a peak cycle efficiency of 30% at a peak temperature of 700°F . Reduction of the temperature from 700°F to 500°F (reduction of the peak pressure from 3,090 psia to 680 psia) reduced the cycle efficiency by only 6 percentage points (to 24%). Of course, the reduction is much greater in terms of the available energy if the peak exhaust temperature is constant at 719°F . The significance of this result is that the part load efficiency can be kept at a high value as the diesel power (exhaust temperature) drops.

The influence of liquid temperature on cycle efficiency for a constant back pressure (.7 psia) and constant peak cycle temperature (660°F) is shown in Figure 19. Reduction of the temperature from 240°F to 90°F results in a reduction of the efficiency from 29% to 25%. This change results from reducing the mass ratio from 6.5 to 3.9. At the higher mass ratio, more energy is transferred into the liquid phase resulting in the higher efficiency.

The nozzle back pressure (condensing temperature) has perhaps the greatest effect on cycle efficiency and turbine rpm. Figure 20 shows the cycle efficiency to increase from 18% to 29% as the back pressure is decreased from 14.7 psia to .7 psia. For a constant liquid outlet temperature this results in a simultaneous increase in mass ratio, r , and nozzle spouting velocity. The mass ratio increases from 4.7 to 6.5 while the spouting velocity increases from 1,660 ft/s to 1,920 ft/s. In addition to the disadvantage of higher turbine rpm, decreasing the

Fig. 18- Efficiency Versus Peak Cycle Temperature for Two-Component Bottoming Cycle

Fig. 19 - Efficiency Versus Nozzle Liquid Exit Temperature for Two-Component Bottoming Cycle

Fig. 20 - Efficiency Versus Nozzle Exit Pressure (Heat Rejection Temperature) for Two-Component Two-Phase Bottoming Cycle

back pressure results in a large increase in nozzle exit area which tends to reduce both the nozzle and the turbine efficiency. Thus, the gains reflected in Figure 20 would be somewhat reduced due to decreasing nozzle and turbine efficiency.

The effect of the liquid working fluid can be seen in Figure 21 through the influence of the sensible heat change. The specific heats of different fluids are indicated on the figure for comparison purposes. As can be seen, a gradual decrease in cycle efficiency occurs as the heat capacity is decreased. For example, decreasing the sensible heat from .6 B/lb[°]F to .2 B/lb[°]F reduces the cycle efficiency from 29% to 27½%. However, the mass ratio increases from 3.9 to 11.7 and the nozzle spouting velocity is reduced from 2,350 ft/s to 1,520 ft/s. This example best illustrates the transformation possible if the mass ratio of liquid to gas can be increased within the thermodynamic constraints available. The reference cycle finally adopted, is operated at a relatively high nozzle exit pressure to limit the nozzle spouting velocity. However, if a suitable fluid with a significantly lower C_p than that of Dow-Therm A can be used, considerable efficiency improvement should be obtainable.

The above results were generated for constant values of nozzle efficiency, turbine efficiency and pump efficiency. The influence of these parameters on cycle efficiency are shown in Figures 22 through 26. In general, the nozzle and turbine efficiencies have a very strong influence on cycle efficiency, whereas, the pump or diffuser efficiency has a less pronounced effect. For example, increasing the turbine and nozzle efficiencies from the design value of .85 to .90 would increase the cycle efficiency from 28% to 31%. Decreasing them both to .70 would reduce the cycle efficiency from 28% to a little over 20%, which is still a usable value. On the other hand, if the pump efficiency were decreased from .85 to .65, the cycle efficiency would only drop by less than one percentage point (from 28.4% to 27.5%). Thus, a considerable range of useful performance exists whether

Fig. 21 - Cycle Efficiency Versus Sensible Heat of Liquid Component for Two-Component Two-Phase Bottoming Cycle

Fig. 22 - Influence of Nozzle, Turbine and Pump Efficiencies on Cycle Efficiency

Figs. 23,24 :- Influence of Nozzle, Turbine and Pump Efficiencies on Cycle Efficiency.

Figs. 25,26 - Influence of Nozzle, Turbine and Pump Efficiencies on Cycle Efficiency

the nozzle, turbine and diffuser attain the highest values tested or calculated (~ 0.9) or the lowest values likely (~ 0.7).

To summarize, the two-component cycle appears to offer substantially lower turbine rpm and somewhat higher efficiencies than a single component system. It was, therefore, chosen to provide the basis for the reference cycle design. Other variations, such as compound cycles or the contact heat exchanger cycle, can offer substantially the same performance as the above variations while possibly achieving other advantages. However, the basic criteria for choice of the reference cycle type and parameters was felt to be simplicity, cost and 'state of the art' components, so a single-stage two-component system was examined.

IV. REFERENCE CYCLE

The reference cycle chosen for this design study uses the exhaust from an 1800 HP Colt-Pielstick PC-2 diesel engine to provide heat to the two-phase turbine engine. Table 2 (from Ref. 7) summarizes the operating characteristics of the engine over the range of interest. This engine was chosen for the design study because the design power (1800 HP) is large enough to provide a reasonable bottoming cycle efficiency and the engine exhaust temperature (719°F) is representative of many engines of interest for waste heat recovery. The minimum exhaust temperature (525°F) was treated in the cycle analysis and the cycle efficiency remained in excess of 20%.

The peak efficiency ($\sim 30\%$) for a bottoming cycle could be obtained by expansion of the two-phase mixture to the lowest possible exhaust pressure (.7 psia for 90°F rejection temperature). Use of a liquid component with a low sensible heat would enable operation at this exhaust pressure with a high mass ratio and, therefore, low spouting velocity. However, the lowest sensible heat ($\sim .37 \text{ B/lb}^{\circ}\text{F}$) was found for a 'Therminol' fluid which may have toxicity problems. It was therefore decided to use Dow-Therm A which although having a higher sensible heat ($\sim .51$), is a common heat transfer fluid with no toxicity problems for which commercial equipment is available. With this fluid, the spouting velocity for peak efficiency (29%) was very high.

In order to limit the rpm to reasonable values (~ 8000) and to simplify the seals and bearings problems, it was decided to operate the initial system with a nozzle back pressure of one atmosphere. This still produced an acceptably high efficiency (23%) with a single stage system without any sub-atmospheric seals. In order to increase the efficiency beyond the normal range for this back pressure (17-18%), the steam was further expanded (to 1.2 psia) after separation.

TABLE 2

OPERATING PARAMETERS OF 38TD8-1/8 DIESEL ENGINE

% of Continuous Rated Power	110	100	75	50	25
BMEP, psi	139.8	127	95.3	63.5	31.8
Input (HHV)	7117	7132	7311	7861	9534
Input (LHV)	6687	6702	6871	7386	8936
Exh. Sensible, Btu/bhphr	2200	2260	2460	2860	3660
Exh. Latent, Btu/bhphr	430	430	440	475	598
Lube Oil, Btu/bhphr	700	702	728	808	1075
AC & Turbo H ₂ O, Btu/bhphr	560	512	430	380	349
Cooling H ₂ O, Btu/bhphr	460	461	478	530	706
Rad. & Unacc., Btu/bhphr	222	222	230	263	602
Useful Work, Btu/bhphr	2545	2545	2545	2545	2545
Exhaust Temp., °F	720	719	701	654	525
Air Consumption, lb/bhphr	14.2	14.8	16.3	20.5	33.5
Fuel Consumption, lb/bhphr	0.368	0.369	0.378	0.406	0.492
Exhaust Gas, lb/bhphr	14.57	15.17	16.68	20.91	33.99
η (HHV)	0.358	0.357	0.348	0.324	0.267
η (LHV)	0.381	0.380	0.370	0.345	0.285

The final state points and flow conditions for the reference cycle are shown in Table 3. The temperatures and pressures are well within the state of the art for steam and Dow-Therm systems. The net power output of 276 HP, when added to the diesel output of 1800 HP gives an increase of 15%, or on a per unit fuel basis, the same power for 15% less fuel. For the 1800 HP size, a 40% basic diesel efficiency, and a fuel price of \$2.00/million B, the annual fuel savings would be \$24,000 (for an 80% capacity factor), giving a 3 year payback for about \$300/kw capital cost of the bottoming system. This cost appears to be a reasonable goal, considering the simplicity.

FLOW CONDITIONS

The flow conditions were chosen to provide a slight degree of superheat at the exit of the nozzle to avoid the Wilson Line during subsequent expansion of the steam in the radial inflow turbine. For inlet conditions of 660°F and 2,365 psia and nozzle exit conditions of 230°F and 14.7 psia, the following values result:

$$s''_{a_1} = .8987 \text{ B/lb}^\circ\text{F}$$

$$s''_{ax} = 1.7695 \text{ B/lb}^\circ\text{F}$$

$$C''_{pb} = .51 \text{ B/lb}^\circ\text{F}$$

$$h''_{a_1} = 714.2 \text{ B/lb}$$

$$h'_{ax} = 1159.3 \text{ B/lb}$$

Substitution in Eq. (3) gives:

$$r = 3.50$$

Substitution of the enthalpy values in Eq. (28) of Appendix B gives:

$$V_{2i} = 1,078 \text{ ft/s}$$

A nozzle efficiency of .85 yields:

$$V_2 = 1,824 \text{ ft/s}$$

Entry at 15°, with a radius change of .96 from the nozzle center line to the separator drum, gives a separator liquid velocity of:

$$V_s = (.96)(\cos 15^\circ)(1,824) = 1,691 \text{ ft/s}$$

TABLE 3
STATE POINTS FOR REFERENCE BOTTOMING CYCLE

<u>State Point</u>	<u>Temperature (°F)</u>	<u>Pressure (psia)</u>	<u>Flowrate (lb/s)</u>	<u>Velocity (ft/s)</u>	<u>Liquid/ Liquid/Gas</u>
1	660	2365	2.97	30	L
2	230	14.7	2.97	1824	L+G
3	230	14.7	2.31	1691	L
4	107	1.2	.66	50	L+G*
5	230	14.7	2.31	30	L
6	100	1.2	.66	30	L
7	190	2465	2.97	30	L
8	719	15.0	7.58	50	G
9	242	14.7	7.58	50	G

*Vapor quality = .96

POWER OUTPUT - 276 HP @ 8075 RPM

CYCLE EFFICIENCY - 23%

NET POWER INCREASE - 15%
ABOVE DIESEL OUTPUT

For a two foot diameter the separator rpm is:

$$W_s = 16,150 \text{ rpm}$$

Use of a "U-tube" turbine at an rpm one half of the separator rpm gives a turbine rpm of:

$$W_t = 8,075 \text{ rpm}$$

The liquid turbine tip speed is 845.5 ft/s. The torque on the turbine due to momentum change is:

$$L''_{tm} = 2\dot{m}_b (V_s - V_t) r_t \quad (4)$$

$$L''_{tm} = 121.3 \text{ lbft}$$

The drag on the liquid turbine is given by:

$$F''_{td} = \frac{C_d \rho_b (V_s - V_t)^2}{2} A_i \quad (5)$$

or

$$F''_{td} = C_d \frac{\dot{m}_b (V_s - V_t)}{2} \quad (6)$$

where C_d = the drag coefficient for the liquid pickup

A_i = the inlet area

A value of $C_d = .1$ appears attainable for an inlet with base ventilation⁽¹¹⁾. Substitution in Eq. (6) gives:

$$F''_{td} = 3.03 \text{ lb}$$

Since a 1 foot radius is found at this location the drag torque on the inlet is:

$$L''_{td} = 3.03 \text{ lbft}$$

The total torque on the liquid turbine is:

$$L''_t = 124.3 \text{ lbft}$$

The power is therefore:

$$P''_t = \frac{(124.3)(845.5)}{(550)} = 191 \text{ HP}$$

The power loss due to the drag of the inlet is:

$$P_{s1} = \frac{(3.03)(1,691)}{(550)} = 9.3 \text{ HP}$$

The windage loss of the separator was estimated considering the rim to be a flat plate with the width equal to the separator width and the length equal to the circumference.

The Reynolds Number is:

$$Re_x = \frac{V_s L_s}{\nu'_a} \quad (7)$$

For steam at 230°F and 14.7 psia, $\nu'_a = 2.3 \times 10^{-4} \text{ ft/s}$

$$Re_x = \frac{(1,691)(6.28)}{(2.3 \times 10^{-4})} = 4.55 \times 10^7$$

The friction factor is (12):

$$f_{pm} = .0024$$

The drag on the separator is:

$$F_s = f_{pm} \frac{\rho'_a V_s^2}{2} L_s w_s \quad (8)$$

where w_s = separator width

$$F_s = 4.04 \text{ lb}$$

The power loss due to windage is:

$$P_{s2} = \frac{(4.04)(1,691)}{(550)} = 12.4 \text{ HP}$$

Allowing 1.3 HP loss in the bearings and seals, gives a total separator loss of:

$$P_o = 23 \text{ HP}$$

The net power from the radial inflow turbine obtained by expansion to 1.2 psia is estimated in Appendix D to be:

$$P'_t = 137 \text{ HP}$$

The pumping power required to return the liquid through the heat exchanger to the nozzle is:

$$P_p = \frac{(Q_a + Q_b) \Delta p}{\eta_p} \quad (9)$$

where Q_a = volume flowrate of A

Q_b = volume flowrate of B

Δp = pressure rise = 2,450 psi

η_p = pump efficiency = .75

$$P_p = 28.9 \text{ HP}$$

The net power is given by:

$$P_n = P'_t + P''_t - P_s - P_p \quad (10)$$

Substitution of the above values gives:

$$P_n = 137 + 191 - 23 - 29 = 276 \text{ HP}$$

The heat rejection is:

$$Q_c = \dot{m}_a L_{va} x_{va} \quad (11)$$

where x_{va} = the exit quality from the steam turbine

Substitution of the appropriate values gives:

$$Q_c = 654.4 \text{ B/s}$$

The cycle efficiency is:

$$\eta_c = \frac{P_n}{P_n + Q_c} \quad (12)$$

or

$$\eta_c = .23$$

OPERATION

The schematic diagram of Figure 27 shows a configuration of the reference cycle where the exhaust from the diesel is passed through a jacket to pre-heat the inventory. Startup is then achieved by a ramp injection of the hot charge downstream of a check valve. This type of startup technique was successfully used in a similar two-phase cycle⁽¹³⁾ where the power was extracted in a magneto-hydrodynamic generator instead of a turbine. In that case, steady state operation with a flow of 150 lb/s of Nak was achieved in 2-3 seconds.

Fig. 27 - Schematic of Reference System

V. TURBINE DESIGN

The design configuration of the two-phase turbine resulting from the analysis for the reference bottoming application summarized in Section IV, is given in Figures 28 and 29. The basic procedures used in the mechanical design are summarized in Appendix C. The steam turbine design is described in Appendix D.

The turbine has a single stage expansion of the two-phase mixture in the 12 inch nozzles. The mixture impinges on the inside surface of the free wheeling drum, forming a liquid layer traveling at 1,691 ft/s. The layer drives a liquid turbine to 845 ft/s (8,075 rpm) for the "U-tube" version shown in Figure 28. The discharge from the U-tube is collected in an annular tube and exits from the hot well of the turbine. The separator drum free wheels on the turbine shaft and is held in place by clips and retainers as shown. The separator drum and turbine are overhung from the 35 mm shaft. The bearings are standard grease packed double seal bearings. A simple mechanical sliding seal is used to isolate the low temperature (230°F) steam at atmospheric pressure from ambient.

The separated steam flows radially inward from the drum through a two-stage turbine to the exhaust pressure of 1.2 psia. The radial inflow turbine is keyed to the same shaft as the liquid turbine and thus has the same rpm, 8,075. As summarized in Appendix D, a total of 137 HP is added by the steam turbine to the liquid turbine output of 191 HP. Expansion of the total two-phase mixture to the lower pressure would result in a greater fraction of the power being available in the liquid phase. The turbine blading is conventional radial design and could probably be cast in quantity for \$75-100.

Figure 29 shows the nozzle detail and the detail of a counter-balanced radial inflow liquid turbine. The nozzle is built up of two conical sections and a throat, in accordance with past design practice⁽¹⁴⁾.

Fig. 28 - Diesel Bottoming Test Turbine

BPHASE ENGINEERING INC.	
DIESEL BOTTOMING TEST TURBINE	
DATE 8-21-76	DWG NO 76-101-1
BY L. HARRIS	FRAGILL 101

Fig. 29 - Nozzle and Liquid Turbine Design Details

Recent cost estimates have been obtained for hydroforming these nozzles for a price of about \$150 in quantity⁽¹⁵⁾. The liquid inlet design on the liquid turbine is similar to configurations used for high pressure, pitot pumps⁽¹⁶⁾.

Thus, the turbine and nozzle components appear to be basically low cost units.

The materials are entirely conventional alloys except for the drum which is titanium for this design. Table 4 summarizes the materials for major components of the assembly.

To summarize, the nozzle and turbine assembly is a simple, low cost design which can be mass produced for much less than a multi-stage axial flow gas turbine with high speed bearings and seals. Increases in the mass ratio, made possible through using a liquid with lower sensible heat would enable a lower separator drum rpm and the use of low carbon steel. Staging the nozzle expansion would have a similar influence but with an increase in the total number of components.

TABLE 4
MATERIALS LIST FOR MAJOR TURBINE COMPONENTS

Dash No.	Part	Material		
-1	Nozzle (2)	ASTM A148-58	GRADE 150-125	(Steel Casting)
-2	Casing, Front	ASTM A105-55T		(Steel Forging)
-3	Casing, Side	ASTM A105-55T		(Steel Forging)
-4	Casing, Back	ASTM A105-55T		(Steel Forging)
-5	Liquid Collector	ASTM A106		(Steel Pipe)
-6	Steam Turbine	ASTM A148-58	GRADE 150-125	(Steel Casting)
-7	Separator Drum	AMS 4969 (Ti - 5Al - 1.5 Fe - 1.4 Cr - 1.2 Mu)		
-8	Liquid Turbine	ASTM A148-58	GRADE 150-125	(Steel Casting with Stellite tip)
-9	Bearing Housing	ASTM A106-55T		(Steel Pipe)
-10	End Plate	ASTM A105-55T		(Steel Forging)
-11	Shaft	ASTM A105-55T		(Steel Forging)
-12	Base	ASTM A105-55T		(Steel Forging)
-13	Outlet Pipe	ASTM A106		(Steel Pipe)

VI. SYSTEM SPECIFICATIONS

Components of the two-phase turbine cycle chosen for this application are all well within the state of the art. Common fluids were chosen for the first system (water and Dow-Therm A). The temperatures for the pumps and valves are in the range of 90-250°F. The peak temperature for the exhaust gas-liquid heat exchanger is that of the diesel exhaust, 719°F. The pressures required are high (2,500 psia, maximum) but are within normal practice for both steam and hydraulic systems. The only "exotic" materials required are Stellite tip on the inlet of the liquid turbine to minimize erosion and, for the initial system, a titanium drum for the separator. Use of a liquid with a lower specific heat than Dow-Therm would reduce the separator and turbine rpm, simplifying the bearing and seal requirements further. However, the values for the reference cycle (16,150 and 8,075 rpm) are within the range of commercially available seals and bearings. In order to enable preliminary assessments to be made of the system cost and complexity, specifications for the major components are given below:

PRIMARY HEAT EXCHANGER

Design	- tube-fin construction, heliarc welded joints
Material	- ASTM A105-55T, A106-55T (carbon steel)
Inlet liquid temperature (tube side)	190°F
Inlet liquid pressure	2,465 psia
Inlet gas temperature (fin side)	719°F
Outlet liquid temperature	660°F
Outlet liquid pressure	2,365 psia
Outlet gas temperature	242°F
Gas pressure drop	10" H ₂ O
Liquid flowrate	2.97 lb/s
Gas flowrate	7.58 lb/s
Heat transfer rate	654 B/s (net)

The type of construction to be used can be very similar to the unit shown in Fig. 30. Some decrease in construction costs relative to a vapor generator should be realized because of the smaller flow area (hence, tube size) required by the liquid.

CONDENSER

Design - tube-shell construction	
Material - ASTM A105-55T, ASTM A106-55T (carbon steel)	
Inlet temperature	107°F
Inlet pressure	1.2 psia
Outlet temperature	100°F
Outlet pressure	1.1 psia
Flowrate	.66 lb/s, steam
Heat transfer rate	504 B/s

Figure 31 is a isometric drawing of a condenser used in a diesel bottoming application. A smaller surface area would be possible for the two-phase bottoming cycle because of the higher condensing coefficients of the steam (relative to organic vapor).

START AND DUMP TANK

Design - welded construction	
Material - ASTM A105-55T	
Pressure (maximum)	2,500 psia
Temperature (maximum)	500°F
Volume	4 ft ³

PUMP

Pressure rise	2,500 psia
Flowrate	3 lb/s, water - Dow-Therm A mixture
Efficiency	.75

Fig. 30 - Isometric Illustration of a Typical
Exhaust Gas Heat Exchanger
(After Ref. 7)

Fig. 31 - Isometric Illustration of a Typical Condenser
(After Ref. 7)

CONTROL VALVES AND START VALVE

Type - high pressure globe

Material - ASTM A217 Grade WC6 or ASTM A182 Grade F11

Pressure 2,500 psia

Temperature (maximum) 660°F

Size 1 "

CHECK VALVE

Type - swing or lift

Material - ASTM A335-55T Grade P11 or ASTM 217 Grade WC6

Pressure 2,500 psia

Temperature (maximum) 660°F

Size 1 "

PIPING, HIGH TEMPERATURE

Seamless Chrome - Moly Steel, ASTM A369

Pressure 2,500 psia

Temperature (maximum) 660°F

Diameter 1 ", 1/2"

PIPING, LOW TEMPERATURE

Seamless Carbon Steel, ASTM A106

Pressure 2,500 psia

Temperature (maximum) 250°F

Diameter 1 ", 1/2", 1'

BEARINGS

Bore - 35 mm

Grease Packed

e.g. Fafnir Medium 2MM307 WI Single and 2 MM307 DB Double

SEALS

Carbon Face Seal, Bellows Loading

(Design Equivalent to Chicago Rawhide Manf. Co. No. B33-90 66 T1)

VII. PROGRAM PLAN

A program culminating in the demonstration of a two-phase bottoming system was developed for the reference design cycle. The main objective of the plan is to demonstrate the technology of two-phase turbine bottoming systems in sufficient depth to provide a firm basis for commercial applications. The time required, 3 years, and the estimated cost, 1.3 million dollars, are very reasonable values for demonstration of a new technology which has the potentially large industrial impact of the two-phase turbine.

The first year's program is summarized in Figure 32. The basic objective of the first year is to demonstrate performance of the nozzle and turbine for the actual flow conditions of the bottoming system. A simple blowdown test will be conducted with steam alone and with a mixture of steam and a second fluid. The major portion of the blowdown test facility will be available from capital equipment and government furnished equipment. The system is sized to provide 50 HP output power for a 10 minute duration. The testing of both steam alone and steam and oil, will provide sufficient data for either single component or two-component systems. The end of the first year provides a convenient, low cost (\$230K), decision point for continuing the effort. The level of performance attainable will have been demonstrated for the fluids and temperatures of interest. Performance and economic analyses based on the actual test performance will have been completed.

The second year's program will demonstrate performance of the complete two-phase bottoming system using a hot gas generator to simulate the exhaust from a diesel engine. The estimated schedule is summarized in Figure 33. The power output from the system will be in excess of 200 HP on a continuous basis. The bottoming system will be designed for eventual long term operation to verify reliability.

MONTHS AFTER INCEPTION OF CONTRACT

Fig. 32 - Schedule for Demonstration of a Biphas Bottoming Cycle - Year 1

MONTHS AFTER INCEPTION OF CONTRACT

Fig. 33 - Schedule for Demonstration of a Biphasic Bottoming Cycle - Year 2

The third year's program, shown in Figure 34, will consist of endurance testing of the hot gas laboratory system and field testing of the final two-phase bottoming system. Thus, reliability of the rotating unit and performance of the complete system will both be demonstrated. The power level of the final field test unit will depend upon the available diesel engine for the demonstration. The program plan assumes availability of a large industrial diesel or spark ignition engine in the Southern California area. Likely candidates that have such engines are the Southern California Gas Company, Lomita Gas Company, the seven major refineries, Southern California Edison and the Los Angeles Department of Water and Power.

A cost breakdown for the 1.3 million dollar demonstration program is shown in Tables 5, 6 and 7. The costs are based on 1976 dollars with an expected annual salary increase of 10% and an overhead increase of five percentage points per year.

MONTHS AFTER INCEPTION OF CONTRACT

Fig. 34 - Schedule for Demonstration of a Biphas Bottoming Cycle - Year 3

TABLE 5

COST SUMMARY OF FIRST YEAR'S PROGRAM

DETAIL DESCRIPTION OF COST ELEMENTS					
1. DIRECT MATERIAL (Itemize on Exhibit A)			EST COST (\$)	TOTAL EST COST ¹	REFERENCE ²
a. PURCHASED PARTS					
b. SUBCONTRACTED ITEMS --Nozzle, turbine/separator			36,000		
c. OTHER--(1) RAW MATERIAL					
(2) YOUR STANDARD COMMERCIAL ITEMS					
(3) INTERDIVISIONAL TRANSFERS (At other than cost)					
TOTAL DIRECT MATERIAL				36,000	
2. MATERIAL OVERHEAD ³ (Rate %X\$ base=)					
3. DIRECT LABOR (Specify)		ESTIMATED HOURS	RATE/HOUR	EST COST (\$)	
Principal Scientist		900	17.17	15,453	
Technical Specialist		1800	11.67	21,006	
Technician		1800	8.40	15,120	
Design/Draftsman		300	7.88	2,364	
Secretarial		900	7.35	6,615	
TOTAL DIRECT LABOR				60,558	
4. LABOR OVERHEAD (Specify Department or Cost Center) ⁴		O.H. RATE	X BASE =	EST COST (\$)	
		.75		45,419	
TOTAL LABOR OVERHEAD				45,419	
5. SPECIAL TESTING (Including field work at Government installations)			EST COST (\$)		
TOTAL SPECIAL TESTING				40,600	
6. SPECIAL EQUIPMENT (If direct charge) (Itemize on Exhibit A)					
7. TRAVEL (If direct charge) (Give details on attached Schedule)			EST COST (\$)		
a. TRANSPORTATION			2,000		
b. PER DIEM OR SUBSISTENCE					
TOTAL TRAVEL				2,000	
8. CONSULTANTS (Identify--purpose--rate)			EST COST (\$)		
TOTAL CONSULTANTS					
9. OTHER DIRECT COSTS (Itemize on Exhibit A)				13,000	
TOTAL DIRECT COST AND OVERHEAD				107,577	
11. GENERAL AND ADMINISTRATIVE EXPENSE (Rate % of cost element Nos.) ⁵				21,733	
12. ROYALTIES ⁶					
13. TOTAL ESTIMATED COST				219,310	
14. FEE OR PROFIT 7%				15,451	
15. TOTAL ESTIMATED COST AND FEE OR PROFIT				234,761	

TABLE 5 - continued

EXHIBIT A—SUPPORTING SCHEDULE (Specify. If more space is needed, use reverse)		
COST EL NO.	ITEM DESCRIPTION (See footnote 5)	EST COST (\$)
6	Oil Accumulator Tank	3,000
6	Start/Dump Tank	5,000
6	Chilled Water System	7,000
6	Working Fluid	2,000
6	Condensate Pump	2,500
6	Throttle Valves (3)	1,800
6	Control Valves (6) sol. op.	2,400
6	Instrumentation:	
	Pressure Gauges (10)	500
	Flowmeters (4)	2,000
	Temp Readouts & TC's (3)	1,800
	Recorders (2), Counter (1), Digital Display (1)	7,000
	Load Cells (2)	1,400
	Tachometers (2)	1,200
6	Dynamometer (50 HP)	1,500
6	Safety Equipment	1,500
9	Test Loop Support System:	
	Piping Fabrication	3,000
	Water System Fabrication	1,000
	Steam System Fabrication	4,000
	Insulation Fabrication	2,000
	Control Panel Fabrication	3,000
I. HAS ANY EXECUTIVE AGENCY OF THE UNITED STATES GOVERNMENT PERFORMED ANY REVIEW OF YOUR ACCOUNTS OR RECORDS IN CONNECTION WITH ANY OTHER GOVERNMENT PRIME CONTRACT OR SUBCONTRACT WITHIN THE PAST TWELVE MONTHS? ☐ YES ☐ NO (If yes, identify below.)		
NAME AND ADDRESS OF REVIEWING OFFICE AND INDIVIDUAL		TELEPHONE NUMBER/EXTENSION
II. WILL YOU REQUIRE THE USE OF ANY GOVERNMENT PROPERTY IN THE PERFORMANCE OF THIS PROPOSED CONTRACT? ☐ YES ☐ NO (If yes, identify on reverse or separate page)		
III. DO YOU REQUIRE GOVERNMENT CONTRACT FINANCING TO PERFORM THIS PROPOSED CONTRACT? ☐ YES ☐ NO (If yes, identify.): ☐ ADVANCE PAYMENTS ☐ PROGRESS PAYMENTS OR ☐ GUARANTEED LOANS		
IV. DO YOU NOW HOLD ANY CONTRACT (Or, do you have any independently financed (IR&D) projects) FOR THE SAME OR SIMILAR WORK CALLED FOR BY THIS PROPOSED CONTRACT? ☐ YES ☐ NO (If yes, identify.):		
V. DOES THIS COST SUMMARY CONFORM WITH THE COST PRINCIPLES SET FORTH IN AGENCY REGULATIONS? ☐ YES ☐ NO (If no, explain on reverse or separate page)		

TABLE 6

COST SUMMARY OF SECOND YEAR'S PROGRAM

DETAIL DESCRIPTION OF COST ELEMENTS				
1. DIRECT MATERIAL (Itemize on Exhibit A)	EST COST (\$)		TOTAL EST COST	REFERENCE
a. PURCHASED PARTS				
b. SUBCONTRACTED ITEMS Nozzles (2), Separator/Turbine	58,000			
c. OTHER—(1) RAW MATERIAL				
(2) YOUR STANDARD COMMERCIAL ITEMS				
(3) INTERDIVISIONAL TRANSFERS (At other than cost)				
TOTAL DIRECT MATERIAL			58,000	
2. MATERIAL OVERHEAD ¹ (Rate % X \$ base =)				
3. DIRECT LABOR (Specify)	ESTIMATED HOURS	RATE/HOUR	EST COST (\$)	
Principal Scientist	1800	18.80	33,840	
Technical Specialist	2550	12.78	32,589	
Technician	2100	9.20	19,320	
Design/Draftsman	1200	8.63	10,356	
Secretarial	900	8.05	7,245	
TOTAL DIRECT LABOR			103,350	
4. LABOR OVERHEAD (Specify Department or Cost Center) ²	O.H. RATE	X BASE =	EST COST (\$)	
	.80		82,680	
TOTAL LABOR OVERHEAD			82,680	
5. SPECIAL TESTING (Including field work at Government installations)	EST COST (\$)			
TOTAL SPECIAL TESTING				
6. SPECIAL EQUIPMENT (If direct charge) (Itemize on Exhibit A)			96,400	
7. TRAVEL (If direct charge) (Give details on attached Schedule)	EST COST (\$)			
a. TRANSPORTATION	4,000			
b. PER DIEM OR SUBSISTENCE				
TOTAL TRAVEL			4,000	
8. CONSULTANTS (Identify—purpose—rate)	EST COST (\$)			
TOTAL CONSULTANTS				
9. OTHER DIRECT COSTS (Itemize on Exhibit A)			46,000	
10. TOTAL DIRECT COST AND OVERHEAD			390,430	
11. GENERAL AND ADMINISTRATIVE EXPENSE (Rate % of cost element Nos.) ³			42,947	
12. ROYALTIES ⁴				
13. TOTAL ESTIMATED COST			433,377	
14. FEE OR PROFIT 7%			30,336	
15. TOTAL ESTIMATED COST AND FEE OR PROFIT			463,713	

TABLE 6 - continued

EXHIBIT A—SUPPORTING SCHEDULE (Specify. If more space is needed, use reverse)		
COST EL NO.	ITEM DESCRIPTION (See footnote 5)	EST COST (\$)
6	Procurements:	
	Working Fluids	5,000
	Gas-fired Burner	13,000
	Heat Exchanger	8,000
	Start/Dump Tank	3,000
	Regenerator	5,000
	Condenser	6,000
	Pump	2,000
	Throttle Valves (5)	4,000
	Control Valves (10)	5,000
	Instrumentation:	
	Gauges (15)	1,000
	Flowmeters (6)	3,000
	Temperature Readouts (6)	3,600
	Recorders (2), Counters (2), Digital Display (2)	10,000
	Load Cells (4)	2,800
	Dynamometer (600 HP)	12,000
	Data Handling Hardware	5,000
	Computer Processing	8,000
9	Test Loop Support Systems Fabrication	
	Hot Gas Generator	14,000
	Start and Dump System	6,000
	Liquid Feed System	4,000
	Vapor Feed/Condensation System	7,000
	Control Panel	15,000
I. HAS ANY EXECUTIVE AGENCY OF THE UNITED STATES GOVERNMENT PERFORMED ANY REVIEW OF YOUR ACCOUNTS OR RECORDS IN CONNECTION WITH ANY OTHER GOVERNMENT PRIME CONTRACT OR SUBCONTRACT WITHIN THE PAST TWELVE MONTHS?		
☐ YES ☐ NO (If yes, identify below.)		
NAME AND ADDRESS OF REVIEWING OFFICE AND INDIVIDUAL		TELEPHONE NUMBER/EXTENSION
II. WILL YOU REQUIRE THE USE OF ANY GOVERNMENT PROPERTY IN THE PERFORMANCE OF THIS PROPOSED CONTRACT?		
☐ YES ☐ NO (If yes, identify on reverse or separate page)		
III. DO YOU REQUIRE GOVERNMENT CONTRACT FINANCING TO PERFORM THIS PROPOSED CONTRACT?		
☐ YES ☐ NO (If yes, identify.): ☐ ADVANCE PAYMENTS ☐ PROGRESS PAYMENTS OR ☐ GUARANTEED LOANS		
IV. DO YOU NOW HOLD ANY CONTRACT (Or, do you have any independently financed (IR&D) projects) FOR THE SAME OR SIMILAR WORK CALLED FOR BY THIS PROPOSED CONTRACT?		
☐ YES ☐ NO (If yes, identify.):		
V. DOES THIS COST SUMMARY CONFORM WITH THE COST PRINCIPLES SET FORTH IN AGENCY REGULATIONS?		
☐ YES ☐ NO (If no, explain on reverse or separate page)		

TABLE 7 - continued

EXHIBIT A—SUPPORTING SCHEDULE (Specify. If more space is needed, use reverse)		
COST EL NO.	ITEM DESCRIPTION (See footnote 5)	EST COST (\$)
6	Procurements:	
	Working Fluids	5,000
	Heat Exchanger	8,000
	Start/Dump Tank	3,000
	Regenerator	5,000
	Condenser	6,000
	Condensate Pump	2,000
	Alternator, Switch Gear, Controls	20,000
	Throttle Valves (4)	3,200
	Control Valves (8)	4,000
	Instrumentation:	
	Pressure Transducers (10)	5,000
	Flowmeters (6)	3,000
	Temperature Readouts (6)	3,600
	Recorders (1), Counters (1), Digital Readout (1)	5,000
	Data Handling Hardware	20,000
9	Field Test System	
	Startup & Dump System	6,000
	Liquid Feed System	4,000
	Vapor Feed System	7,000
	Control & Instrumentation Module	25,000
9	Heat Exchanger Installation	10,000
9	Interface Equipment Installation	6,000
9	IPI Installation	4,000
9	Control & Instrumentation Module Installation	12,000
<p>I. HAS ANY EXECUTIVE AGENCY OF THE UNITED STATES GOVERNMENT PERFORMED ANY REVIEW OF YOUR ACCOUNTS OR RECORDS IN CONNECTION WITH ANY OTHER GOVERNMENT PRIME CONTRACT OR SUBCONTRACT WITHIN THE PAST TWELVE MONTHS?</p> <p>☐ YES ☐ NO (If yes, identify below.)</p>		
NAME AND ADDRESS OF REVIEWING OFFICE AND INDIVIDUAL		TELEPHONE NUMBER/EXTENSION
<p>II. WILL YOU REQUIRE THE USE OF ANY GOVERNMENT PROPERTY IN THE PERFORMANCE OF THIS PROPOSED CONTRACT?</p> <p>☐ YES ☐ NO (If yes, identify on reverse or separate page)</p>		
<p>III. DO YOU REQUIRE GOVERNMENT CONTRACT FINANCING TO PERFORM THIS PROPOSED CONTRACT?</p> <p>☐ YES ☐ NO (If yes, identify.): ☐ ADVANCE PAYMENTS ☐ PROGRESS PAYMENTS OR ☐ GUARANTEED LOANS</p>		
<p>IV. DO YOU NOW HOLD ANY CONTRACT (Or, do you have any independently financed (IR&D) projects) FOR THE SAME OR SIMILAR WORK CALLED FOR BY THIS PROPOSED CONTRACT?</p> <p>☐ YES ☐ NO (If yes, identify.):</p>		
<p>V. DOES THIS COST SUMMARY CONFORM WITH THE COST PRINCIPLES SET FORTH IN AGENCY REGULATION#5?</p> <p>☐ YES ☐ NO (If no, explain on reverse or separate page)</p>		

VIII. CONCLUSIONS

Cycle calculations for the two-phase turbine bottoming system showed the efficiency to be superior to, or competitive with, organic vapor turbine systems, depending on the assumptions of the analysis. The *two-phase turbine* appears to be much simpler than an *organic or steam vapor turbine* because of the following reasons:

- (1) One stage versus four stages
- (2) 8,100 rpm versus 14,700 to 26,000 rpm
- (3) Simple construction with very few close tolerances

Other advantages for the designed *two-phase system* relative to an *organic Rankine cycle* are:

- (1) No regenerator
- (2) No toxic fluids (the working fluids of candidate organic systems are toluene and FS-85, both of which are toxic)
- (3) Smaller, simpler condenser (steam condensing instead of organic vapor)

The main disadvantages appear to be:

- (1) The required pump is larger than for an organic vapor system
- (2) The state of development is low relative to organic or steam Rankine cycles

Because of the advantages cited it appears that the two-phase bottoming system has the potential of early commercialization. This seems to be particularly true if the cost savings and reduction in fouling that would result from the use of a contact heat exchanger, could be realized. A demonstration program of the type outlined should be sufficient for industry to gain the confidence level necessary for widespread application of this technology.

Research topics which could materially aid in improving performance and reducing cost are:

- (1) Development of low vapor pressure, low sensible heat fluids
- (2) Low loss inlet optimization for the liquid turbine
- (3) Investigation of other moving surface geometries for liquid-gas separation
- (4) Improved atomization techniques for the two-phase nozzle

Institution of these improvements should result in increases in system efficiency, from the 23-25% range to the 30% range, and in reduction of turbine rpm, from the 8,000 to 10,000 range to the 3,000 to 4,000 range. However, as summarized above, the two-phase turbine bottoming system appears to offer enough advantages in its present form to warrant the pursuit of this technology.

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APPENDIX A
EXPANDER RELATIONS

Nomenclature

A_2	nozzle exit area
k_a	ratio of nozzle ring area to wheel area
$\dot{m}_g$	gas flow rate
$\dot{m}_L$	liquid flow rate
$\dot{m}_t$	total flow rate $\dot{m}_g + \dot{m}_L$
p_1	nozzle inlet pressure
p_2	nozzle exit pressure
p_8	diffuser exit pressure
P_d	net output power
P_p	liquid pump input power
P_t	turbine power
R_m	mixture ratio $\dot{m}_L/\dot{m}_g$
R_w	turbine radius
V_g	velocity that would be attained by the gas alone flowing from p_1 to p_2
V_L	velocity that would be attained by the liquid alone flowing from p_1 to p_2
V_s	separator drum velocity
V_w	turbine wheel velocity
V_{wO}	turbine velocity at peak turbine efficiency
V_2	nozzle exit velocity
η_d	diffuser efficiency
η_{exp}	expander efficiency
η_n	nozzle efficiency
η_p	liquid pump efficiency

η_s separator efficiency
 η_t turbine efficiency
 ρ_{g_1} gas density at nozzle inlet
 ρ_{g_2} gas density at nozzle exit
 ρ_L liquid density
 θ nozzle angle relative to plane of rotation

Two-Phase Nozzle

As shown previously, the kinetic energy attained in isentropic two-phase nozzle flow at mixture ratios high enough to give negligible temperature drop, is the sum of the kinetic energies that would be attained by the gas and liquid phases alone. The liquid energy would be:

$$\dot{m}_L V_L^2 / 2 = \dot{m}_L (p_1 - p_2) / \rho_L \quad (1)$$

and the gas energy would be:

$$\dot{m}_g V_g^2 / 2 = \dot{m}_g p_1 \log(p_1 / p_2) / \rho_{g1} \quad (2)$$

The two-phase jet energy is the sum of these two energies times the nozzle efficiency η_n , which is determined from existing algorithms of Biphasic Engines, Inc.

$$\dot{m}_t V_t^2 / 2 = \eta_n (\dot{m}_g V_g^2 + \dot{m}_L V_L^2) / 2 \quad (3)$$

The ratio of two-phase jet velocity V_t to liquid-alone velocity V_L is given by:

$$(V_t / V_L)^2 = (\eta_n \dot{m}_L / \dot{m}_t) \left[(V_g / V_L)^2 / R_m + 1 \right] \quad (4)$$

where R_m is the mass ratio of liquid to gas

Turbine Expander

Wheel Velocity

An impulse turbine has peak efficiency when the blade speed is $V_w = V_t / 2$. From Eq. (4), neglecting the small difference between $\dot{m}_t$ and $\dot{m}_L$ at high mixture ratios, the wheel speed is given by:

$$(V_w/V_L)^2 = (\eta_n/4) \left[(V_g/V_L)^2/R_m + 1 \right] \quad (5)$$

Thus, the mixture ratio R_m required for a specified ratio of wheel velocity V_w to liquid velocity V_L is given by:

$$(V_g/V_L)^2/R_m = (4/\eta_n)(V_w/V_L)^2 - 1 \quad (6)$$

Efficiency

The turbine output power is:

$$P_t = \eta_t \dot{m}_t V_2^2/2 \quad (7)$$

where η_t is calculated from existing algorithms of Biphasic Engines, Inc.

The liquid pump input power is:

$$P_p = \dot{m}_L (p_1 - p_2)/\eta_p \rho_L = \dot{m}_L V_L^2/2\eta_p \quad (8)$$

The net output, or drive, power is:

$$P_d = P_t - P_p \quad (9)$$

Substituting V_2 from Eq. (4), the net power is:

$$P_d = \dot{m}_L V_L^2 \left\{ \eta_t \eta_n \left[(V_g/V_L)^2/R_m + 1 \right] - (1/\eta_p) \right\} / 2 \quad (10)$$

The available input power to the expander is the gas energy $\dot{m}_g V_g^2/2$ from Eq. (2). The expander efficiency is:

$$\eta_{exp} = P_d / \dot{m}_g V_g^2/2 \quad (11)$$

Substituting P_d from Eq. (10) and replacing the expression $R_m (V_g/V_L)^2$ by the expression in terms of V_w/V_L from Eq. (6), the turbopump expander efficiency as a function of the ratio of wheel velocity V_w to liquid-alone velocity is:

$$\eta_{exp} = \frac{4\eta_t (V_w/V_L)^2 - 1/\eta_p}{4(V_w/V_L)^2/\eta_n - 1} \quad (12)$$

This expression gives the efficiency only at a turbine speed V_w equal to half the jet speed V_2 . As V_w and V_2 are reduced, a point is reached where further reduction in V_w can more efficiently be accomplished by reducing V_w and leaving V_2 fixed; the added turbine loss is less than the increased liquid pumping loss. As V_w is reduced below $V_2/2$, the torque increases approximately by the factor $(2 - V_w/V_{w_0})$ where $V_{w_0} = V_2/2$. Therefore, the efficiency decreases by the factor:

$$\eta_{\text{exp}} = \left\{ \eta_{\text{exp}} \right\}_{V_w = V_{w_0}} (2 - V_w/V_{w_0})(V_w/V_{w_0}) \quad (13)$$

The attainable expander efficiency is the larger of the two values given by Eqs. (12) and (13).

Rotopitot Expander

Wheel Velocity

The separator efficiency η_s is defined as the ratio of the liquid energy in the separator to the two-phase jet energy and can be calculated from previously determined two-phase flow relations. Thus, the separator velocity V_s is given by:

$$\dot{m}_L V_s^2/2 = \eta_s \dot{m}_t V_2^2/2 \quad (14)$$

Substituting V_2 from Eq. (4), the ratio of separator velocity V_s to liquid-alone velocity V_L is given by:

$$(V_s/V_L)^2 = \eta_n \eta_s (V_g/V_L)^2/R_m + 1 \quad (15)$$

The pressure at the exit of the diffuser is:

$$p_8 = p_2 + \eta_d \rho_L (V_s - V_w)^2/2 \quad (16)$$

The pressure at the liquid outlet, which must be equal to p_1 , is p_8 minus the centrifugal pressure.

$$p_1 = p_8 - \rho_L V_w^2 / 2. \quad (17)$$

Combining Eqs. (16) and (17) and replacing $(p_1 - p_2)/\rho_L$ by $V_L^2/2$, the ratio of wheel velocity to separator drum velocity is given by:

$$(V_L/V_S)^2 = \eta_d (1 - 2V_w/V_S) - (1 - \eta_d)(V_w/V_S)^2 \quad (18)$$

which can be solved for V_w using the quadratic formula.

Efficiency

The liquid flow $\dot{m}_L$ of velocity V_S is brought to rest in passing radially inward through the turbine. Therefore, the torque on the turbine is $\dot{m}_L V_S R_w$, and the power output, equal to torque times angular velocity, is:

$$P_d = \dot{m}_L V_S V_w \quad (19)$$

The expander efficiency is:

$$\begin{aligned} \eta_{\text{exp}} &= P_d / \dot{m}_g V_g^2 / 2 \\ &= 2R_m V_S V_w / V_g^2 \\ &= 2R_m (V_S/V_L)^2 (V_w/V_S) (V_L/V_g)^2 \end{aligned} \quad (20)$$

Substituting the expression for $R_m (V_L/V_g)^2$ in terms of $(V_S/V_L)^2$ from Eq. (15) and the expression for $(V_S/V_L)^2$ in terms of V_w/V_S from Eq. (18), the rototot expander efficiency is:

$$\eta_{\text{exp}} = \left\{ \eta_d + \left[(1/\eta_n \eta_S) - \eta_d \right] / (V_w/V_S)^2 + (1 - \eta_d)(V_w/V_S)/2 \right\}^{-1} \quad (21)$$

The dependence of η_{exp} on V_w/V_S can be found by choosing a value of V_w/V_S (between 0 and 1/2), finding the efficiency from Eq.(21), the

ratio V_L/V_S from Eq. (18), and then determining V_W/V_L from:

$$V_W/V_L = \frac{V_W/V_S}{V_L/V_S} \quad (22)$$

Flow Rate and Power Limitations

If the nozzle ring is confined to a fraction k_a of the turbine wheel area and the nozzles are tilted at an angle θ relative to the plane of rotation, then the allowable nozzle exit area is:

$$A_2 = \pi R_w^2 k_a \sin \theta \quad (23)$$

The fraction of the nozzle exit area used by the liquid can be neglected, at least for the low pressure stages where flow area limitations are a problem. Hence, the gas flow rate is:

$$\dot{m}_g = \rho_{g2} V_2 A_2 \quad (24)$$

The output power is:

$$P_d = \eta_{\text{exp}} \dot{m}_g V_g^2 / 2 \quad (25)$$

Substituting V_g from Eq. (2), $\dot{m}_g$ from Eq. (24), and A_2 from Eq. (23), the area-limited output power is:

$$P_d = \eta_{\text{exp}} k_a \sin \theta \pi R_w^2 V_2^2 p_2 \log(p_1/p_2) \quad (26)$$

APPENDIX B

ANALYSIS OF TWO-PHASE CYCLES

Two-Phase Bottoming Cycles

The single component-single stage system is shown in Figure 4. The waste heat from a diesel engine (or gas turbine) enters the heat exchanger at temperature T_i with an exhaust flow rate of $\dot{m}_e$. The exhaust heat of an amount Q_i is transferred to a single phase liquid in a counter current heat exchanger. The exhaust gas exits at temperature T_e . The heated liquid is flashed in a nozzle and expanded to pressure p_2 from pressure p_1 . The saturated mixture of vapor and liquid with a quality x_2 impinges on a rotary separator. The vapor flows through a radial inflow turbine and the liquid flows through a radial inflow turbine, each on a common shaft, producing a net power output P_s . The vapor exhausts to the condenser where a heat of Q_c is rejected. The condensate is pressurized and merged with the high pressure outlet from the liquid turbine. The liquid flows back to the heater, closing the cycle.

A multistage version of the single component cycle is shown in Figure 5.

In this version, the exit flow from the first stage is remixed and passed through a regenerative heat exchanger, condensing part of the first stage vapor. The mixture is expanded through the second stage nozzle and turbine (as in the single stage cycle). The condensate and liquid discharge from the final (second) stage is passed through the regenerative heat exchanger, increasing the temperature

to the minimum dictated by combustor material considerations. The advantage of staging is that lower shaft rpm results and the lower velocities result in more efficient nozzle and turbine operation. Against these gains must be balanced the added complexity and possible inter-stage losses.

The use of two components allows more flexibility in the choice of shaft rpm. A single stage two component cycle is shown in Figure 6.

In this cycle the thermodynamic working fluid is component A. It is supplied to the nozzle as a liquid and flash vaporizes upon contact with the heated liquid, component B. Another variation of the two-component cycle is to mix the condensate (component A) with the turbine discharge (component B) circulating the mixture through the heater. This variation has the same cycle diagram as the single-component variation.

The final variation considered uses two components in a manner to enable the use of a contact heat exchanger for the waste gases. This version is illustrated by the schematic of Figure 7.

Component B is sprayed into the contact heat exchanger vessel and flows counter to the direction of the exhaust gases, picking up heat in the process. The hot liquid is collected in the bottom of the vessel and is pressurized by a pump, and injected into the nozzle. The thermodynamic working fluid A is vaporized upon contact with the component B and the mixture is expanded through the nozzle. Component B is separated in the rotary separator turbine driving the hydraulic turbine. Sufficient pressure is left in B to return it to the vessel and spray it into the contact heat exchanger. The vapor exhaust from the rotary separator is condensed, pressurized and returned to the nozzle.

Single Component - Single Stage Analysis

Consider Figure 4, the heat rejected is given by:

$$Q_C = x_{v7} \dot{m}_T L_{v7} + \{(1-\phi_S)(1-x_{v7}) + x_{v7}\} \{\dot{m}_T C''_{p7} \Delta T_{sc}\} \quad (1)$$

where Q_C = cycle heat rejection

x_{v7} = vapor quality at condenser inlet

L_{v7} = latent heat at condenser inlet

ϕ_S = separation efficiency = $\dot{m}_5 / (1-x_{v4}) (\dot{m}_T)$

$\dot{m}_T$ = total mass flow rate

C''_{p7} = liquid specific heat at condenser inlet

ΔT_{sc} = subcooling

The maximum power output from the turbine is:

$$P_t = \Psi_v \eta'_t x_{v4} \dot{m}_T (V'_2)^2 / 2g_c J + \phi_S \eta''_t (1-x_{v4}) \dot{m}_T (V''_2)^2 / 2g_c J \quad (2)$$

where Ψ_v = fraction of gas energy extracted in turbine, 0 or 1

η'_t = efficiency of gas turbine

x_{v4} = vapor quality at 4

V'_2 = vapor exit velocity from nozzle

g_c = 32.16 ft/s²

J = 778 B/lb·ft

η''_t = liquid turbine efficiency

The liquid pumping work is given by:

$$P_{p1} = \frac{\phi_S (1-x_{v4}) \dot{m}_T (p_6 - p_4) (144)}{\eta_d \rho''_4 J} \quad (3)$$

where η_d = diffuser efficiency

p_6 = heater inlet pressure

p_4 = separator casing pressure

ρ''_4 = liquid density at liquid turbine inlet temperature

This power must be subtracted from the liquid turbine power if it is recovered in the diffuser of a radial inflow turbine. The pumping power required for the condensate is:

$$P_{p2} = \frac{\{(1-\phi_s)(1-x_{v4}) + x_{v4}\}(p_9 - p_8)(144)\dot{m}_T}{\eta_p \rho_8'' J} \quad (4)$$

The net power available from the turbine is:

$$P_n = \frac{\psi \eta_t' x_{v4} \dot{m}_T (V')^2}{2g_c J} + \frac{\eta_t'' \phi_s (1-x_{v4}) \dot{m}_T}{g_c J} \left[\frac{(V'')^2}{2} - \frac{(144)(p_6 - p_4)g_c}{\eta_d \rho_4''} \right] - \frac{(144)\{(1-\phi_s)(1-x_{v4}) + x_{v4}\}(p_9 - p_8)\dot{m}_T}{\eta_p \rho_8'' J} \quad (5)$$

The cycle efficiency is given by:

$$\eta_c = \frac{P_n}{P_n + Q_c} \quad (6)$$

If an exhaust gas to working fluid temperature difference of ΔT_{he} is maintained, then the heat input is given by:

$$Q_{i0} = \eta_{he} \dot{m}_e C_{pe} (T_i - T_f) \quad (7)$$

and

$$T_i = T_i - \Delta T_{he}$$

$$T_f = T_o + \Delta T_{he}$$

where η_{he} = heat exchanger efficiency

$\dot{m}_e$ = exhaust gas mass flow rate, lb/s

C_{pe} = exhaust gas specific heat, B/lb^oF

T_i = exhaust gas initial temperature, ^oF

T_f = exhaust gas final temperature, ^oF

The engine efficiency is then:

$$\eta_e = \eta_{he} \eta_c \quad (8)$$

If the exhaust gases are cooled below the dew point, then

$$Q_{i1} = \eta_{he} \dot{m}_e \{ C_{pe} (T_i - T_f) + x_w L_{vw} \} \quad (9)$$

The nozzle exit velocities V_2' and V_2'' can be estimated as:

$$V_2' \approx V_2'' = \sqrt{\eta_n} V_{2i} \quad (10)$$

where V_{2i} is the isentropic expansion velocity for the saturated liquid from T_1 to T_2

η_n = nozzle energy efficiency (0.7 to 0.88)

The vapor quality at the nozzle exit is determined by the approximation

$$X_{v2} \approx X_{v2i} + (1 - \eta_n) \frac{V_{2i}^2}{2g_c J C_p} \frac{1}{L_{v2}} \quad (11)$$

where X_{v2i} = the isentropic expansion exit quality

L_{v2} = latent heat at Station 2, nozzle exit

The rpm is given by:

$$\omega = P_n / L \quad (12)$$

$$L = \frac{\dot{m}_T V_{2i} r_t}{g_c} \quad (13)$$

Single Component - Multistage

In the two-stage system shown in Figure 5, the exit flow from the first stage is passed through a regenerative heat exchanger to reduce the quality to the second stage in order to reduce the spouting velocity and to increase the heat exchanger inlet temperature to a value above the dewpoint of the exhaust gases.

The heat rejected from the condenser is:

$$Q_c = x_{v9} \dot{m}_T L_{v9} + \{ (1 - \phi_{s2}) (1 - x_{v9}) + x_{v9} \} C_{p9} \Delta T_{sc} \dot{m}_T \quad (14)$$

The power output from the first stage turbine is:

$$P_{t1} = \frac{\eta_{v1} \eta_{t1} x_{v4} \dot{m}_T (V_1')^2}{2g_c J} + \frac{\phi_{s1} \eta_{t1} (1 - x_{v4}) \dot{m}_T (V_1'')^2}{2g_c J} \quad (15)$$

The change in temperature through the regenerator in the flow to the heater is given by:

$$\dot{m}_T(x_{V5} - x_{V6})L_{V5} = \dot{m}_T C''_{P11} (T_{12} - T_{11}) \quad (16)$$

where T_{12} = minimum metal surface temperature for exhaust gas
($\approx 217^\circ\text{F}$ from Ref. 7)

or

$$x_{V6} = \frac{L_{V5} x_{V5} - C''_{P11} (T_{12} - T_{11})}{L_{V5}} \quad (17)$$

The power output from the second stage is:

$$P_{t2} = \frac{\psi_v \eta_{t2} x_{V7} \dot{m}_T (V_7')^2}{2g_c J} + \frac{\eta_{t2} \phi_{s2} (1 - x_{V7}) \dot{m}_T}{g_c J} \left[\frac{(V_7'')^2}{2} - \frac{(144g_c)(p_{11} - p_9)}{\eta_d \rho_4'} \right] - \frac{(144)\{(1 - \phi_{s2})(1 - x_{V7}) + x_{V7}\}(p_{11} - p_9)(\dot{m}_T)}{\eta_p \rho_{11}'' J} \quad (18)$$

where the nozzle exit parameters x_V , V_7' , and V_7'' are found by considering an isentropic expansion of the saturated liquid-vapor mixture from quality x_{V6} at temperature T_6 to a pressure of p_7 ; and subsequently applying corrections for the actual nozzle efficiency, η_{n2} .

The cycle efficiency is subsequently given by:

$$\eta_c = \frac{P_{t1} + P_{t2}}{P_{t1} + P_{t2} + Q_c} \quad (19)$$

Methods of determining engine efficiency and other parameters follow those outlined in the single component single stage analysis.

Two-Component - Conventional Heat Exchanger

The most efficient two-component system is achieved by circulating both components through the heater. Considering the cycle diagram of Figure 4 with components A and B, the heat rejected in the condenser is given by:

$$\begin{aligned}
 Q_c &\approx \frac{\dot{m}_T L_{va}}{(1+r)} \left[1 + \frac{C_1}{1-C_1} \frac{L_{vb}}{L_{va}} + \frac{C''_{pa} \Delta T_{sc}}{L_{va}} + \frac{C_1}{1-C_1} \frac{C''_{pb} \Delta T_{sc}}{L_{va}} \right. \\
 &\quad \left. + \frac{(1-\phi_s)(1+r)C''_{pb} \Delta T_{sc}}{L_{va}} \right] \\
 &\approx \frac{\dot{m}_T}{(1+r)} \left\{ L_{va} + \frac{C_1}{1-C_1} L_{vb} + C''_{pa} \Delta T_{sc} + \left[\frac{C_1}{1-C_1} + (1-\phi_s)(1+r) \right] C''_{pb} \Delta T_{sc} \right\} \quad (20)
 \end{aligned}$$

where $L_{va,b}$ = latent heat of vaporization of A,B (B/lb)

r = mass flow ratio of B to A

C_1 = vapor fraction of B in gas = $\frac{W_{b7} P_{b7}}{W_7 P_7}$

W_{b7} = molecular weight of B at T_7 (lb/lb mole)

P_b = vapor pressure of B at T (psia)

W_7 = molecular weight of gas at T_7 (lb/lb mole)

P_7 = condenser inlet pressure (psia)

$C''_{pa,b}$ = specific heat of A,B (B/lb^oF)

ϕ_s = separator efficiency for liquid = $\frac{\dot{m}_{bs}}{\dot{m}_t} \frac{(1+r)}{r}$

ΔT_{sc} = subcooling = $T_7 - T_8$, (^oF)

The exit velocity from the nozzle is given by:

$$V_2 = \sqrt{V_2} = \sqrt{\eta_n} V_{2i} \quad (21)$$

where V_{2i} = the isentropic exit velocity

The power output is given by:

$$P_w = \frac{\Psi \eta_t \dot{m}_t (V'_t)^2}{2g_c J(1+r)} + \frac{\eta_t \phi \dot{m}_t r}{g_c J(1+r)} \left[\frac{(V'')^2}{2} - \frac{(144)(p_6 - p_4)g_c}{\eta_d \rho''_{b4}} \right] \\ - \frac{(144)(1-\phi_s) \dot{m}_t r (p_9 - p_8)}{\eta_p \rho''_{b8} J(1+r)} - \frac{(144) \dot{m}_t (p_9 - p_8)}{\eta_p \rho''_{a8} J(1+r)} \quad (22)$$

In order to determine the mass ratio of component B to component A the flow in the nozzle must be considered. If A and B enter the nozzle in the liquid state and if A is entirely vaporized until only B remains in the liquid state then for an isentropic process:

$$\dot{m}_a s''_{a1} + \dot{m}_b s''_{b1} = \phi_x \dot{m}_a s'_{ax} + (1-\phi_x) \dot{m}_a s''_{ax} + \dot{m}_b s''_{bx} \quad (23)$$

rearranging gives:

$$\phi_x s'_{ax} - \phi_x s''_{ax} = s''_{a1} + r s''_{b1} - s''_{ax} - r s''_{bx} \quad (24)$$

or

$$\phi_x = \frac{r(s''_{b1} - s''_{bx})}{\Delta s_{ax}} + \frac{s''_{a1} - s''_{ax}}{\Delta s_{ax}} \quad (25)$$

If $\phi = 1$

$$r = \frac{\Delta s_{ax} - (s''_{a1} - s''_{ax})}{s''_{b1} - s''_{bx}} \quad (26)$$

where ϕ_x = the mass fraction of A vaporized by station x

$\dot{m}_a$ = mass flow rate of component A

$\dot{m}_b$ = mass flow rate of component B

s''_{a1} = entropy of liquid A at the conditions of station 1 (inlet).

Δs_{ax} = entropy of vaporization of component A at station x

The velocity at x for an isentropic process is given by:

$$\dot{m}_a h''_{a1} + \dot{m}_b h''_{b1} = \phi_x \dot{m}_a h'_{ax} + (1-\phi_x) \dot{m}_a h''_{ax} + \dot{m}_b h''_{bx} + (\dot{m}_a + \dot{m}_b) V_{xi}^2 / 2 \quad (27)$$

Solving for V_{xi}^2 :

$$V_{xi}^2 = \frac{2}{1+r} \left[h''_{a1} - (1-\phi_x)h''_{ax} - \phi_x h'_{ax} + r h''_{b1} - r h''_{bx} \right] \quad (28)$$

$$= \frac{2}{1+r} \left[h''_{a1} - h''_{ax} + \phi_x h''_{ax} - \phi_x h'_{ax} + r(h''_{b1} - h''_{bx}) \right] \quad (29)$$

$$= \frac{2}{1+r} \left[(C''_{pa} + r C''_{pb})(T_1 - T_x) - \phi_x L_{va} + \frac{p_1 - p_x}{\rho''_{ax}} \left(1 + r \frac{\rho''_{ax}}{\rho''_{bx}} \right) \right] \quad (30)$$

And

$$A_{xi} = \frac{\dot{m}_a \phi_x}{\rho''_{ax} V_{xi}} + \frac{\dot{m}_a (1-\phi_x)}{\rho''_{ax} V_{xi}} + \frac{\dot{m}_b}{\rho''_{bx} V_{xi}} \quad (31)$$

$$A_x = \frac{A_{xi}}{\sqrt{\eta_n}} \quad (32)$$

where $V_{xi} = V'_{xi} = V''_{xi}$ = velocity of two-phase mixture after an isentropic expansion from station 1 to x

h''_{a1} = enthalpy of component A liquid at station 1

C''_{pa} = specific heat of component A liquid

T = temperature

L_{va} = latent heat of vaporization of component A

p = pressure

ρ''_{ax} = density of component A liquid at station x

A_{xi} = area at station x for an isentropic expansion from 1 to x

A_x = area for a real expansion from 1 to x

$\eta_n = V_x^2 / V_{xi}^2$

Two Components With Contact Heat Exchanger

The contact heat exchanger variation of Figure 4 has the primary pump as an added component, but replacement of the primary tube and fin heat exchanger by a simple contact unit should result in a significant savings in cost.

The heat transferred from the condenser is:

$$Q_c = \frac{\dot{m}_t}{1+r} \left\{ L_{va} + \left(\frac{C_1}{1-C_1} \right) L_{vb} + C_{pa}'' \Delta T_{sc} + \left[\left(\frac{C_1}{1-C_1} \right) + (1-\phi_s)(1+r) \right] C_{pb}'' \Delta T_{sc} \right\} \quad (33)$$

where the nomenclature was defined previously.

The turbine output power now becomes:

$$P_n = \frac{\psi \eta_t' \dot{m}_t (V_1')^2}{2g_c J(1+r)} + \frac{\eta_t'' \phi_s \dot{m}_t r}{g_c J(1+r)} \left[\frac{(V'')^2}{2} - \frac{(144)(\Delta p_1) g_c}{\eta_d \rho_{b4}''} \right] \\ - \frac{(144)(1-\phi_s) \dot{m}_t r (p_9 - p_8)}{\eta_{p1} \rho_{b8}'' J(1+r)} - \frac{(144) \dot{m}_t (p_9 - p_8)}{\eta_{p1} \rho_{a8}'' J(1+r)} \\ - \frac{(144)(\phi_s) \dot{m}_t r (p_1 - p_6)}{\eta_{p2} \rho_{b1}'' J(1+r)} \quad (34)$$

The same expression for mass ratio holds, but in general the temperature of the thermodynamic working fluid at the nozzle inlet will be equal to the condenser temperature instead of the peak cycle temperature.

Single Component Single Stage Calculations

The engine chosen for consideration is the 38•TD8-1/8. The exhaust temperature and flow rate at full power are:

$$T_i = 719^\circ \text{F}$$

$$\dot{m}_e / P_d \cong 15.17 \text{ lb/bHP}\cdot\text{h}$$

$$P_d = 1980 \text{ bHP}$$

$$\dot{m}_e = 3.004 \times 10^4 \text{ lb/h} = 8.344 \text{ lb/s}$$

If we chose a heat exchanger ΔT of 25°F (pgs. 3-26 of ref. 7), then the peak temperature of the bottoming cycle is:

$$T = 696^{\circ}\text{F}$$

In order to limit the pressure and provide a consistent calculational basis, let $T_1 = 680^{\circ}\text{F}$. For saturated liquid, $p_1 = 2708.1$ psia.

The other conditions at state point 1 are:

$$v''_1 = .0305 \text{ ft}^3/\text{lb}$$

$$v'_1 = .1115 \text{ ft}^3/\text{lb}$$

$$h''_1 = 757.3 \text{ B/lb}$$

$$h'_1 = 1067.2 \text{ B/lb}$$

$$L_{v1} = 309.9 \text{ B/lb}$$

$$s''_1 = .9351 \text{ B/lb}^{\circ}\text{F}$$

$$s'_1 = 1.2071 \text{ B/lb}^{\circ}\text{F}$$

$$\Delta s_v = .2719 \text{ B/lb}^{\circ}\text{F}$$

The first case considered will be a condensing temperature of 212°F .

In this case:

$$T_2 = 212^{\circ}\text{F}$$

$$p_2 = 14.7 \text{ psia}$$

$$v''_2 = .01672$$

$$v'_2 = 26.80$$

$$h''_2 = 180.07$$

$$h'_2 = 1150.4$$

$$L_{v2} = 970.3$$

$$s''_2 = .3120$$

$$s'_2 = 1.7566$$

$$\Delta s_{v2} = 1.4446$$

Isentropic expansion would yield:

$$\Delta h_i = h_1'' - h_2'' - T_2 (s_1'' - s_2'') \quad (35)$$

$$x_2 = \frac{T_2 (s_1'' - s_2'')}{L_{v2}} \quad (36)$$

$$\begin{aligned} \Delta h &= (757.3 - 180.07) - 672 (.9351 - .3120) \\ &= 577.23 - 418.723 = 158.507 \text{ B/lb} = 158.507 \text{ B/s/lb/s} \end{aligned}$$

$$x_2 = (672)(.9351 - .3120)/(970.3) = .432$$

The heat rejected is:

$$\begin{aligned} Q_c &= (.432)(1)(970.3) + [(.99)(1) + (.432)(1)] (1.0)(10) \\ &= 419.7 + 14.22 = 433.39 \text{ B/s} \end{aligned}$$

For a nozzle efficiency of 0.85, the exit velocity is:

$$\begin{aligned} V_2^2 / 2g_c J &= \eta_n \Delta h_i = (158.507)(.85) \\ V_2 &= \sqrt{(64.4)(778)(158.57)(.85)} = 2598.68 \text{ ft/s} \end{aligned}$$

For a turbine efficiency of .85 and a diffuser efficiency of .85 and $\Psi_v = 1$:

$$\begin{aligned} P_n &= \frac{(1)(.85)(.432)(1)(2598.68)}{(64.4)(778)} + \frac{(.85)(.99)(.568)(1)}{(32.2)(778)} \left[\frac{(2598.68)}{2} - \right. \\ &\quad \left. \frac{(144)(2693.4 + 40)(32.2)}{(.85)(59.81)} \right] - \frac{(144)(.432 + .01)(1)(2693.4 + 40)}{(.75)(59.81)(778)} \\ &= 49.49 + 59.667 - 4.883 = 104.27 \text{ B/s} \end{aligned}$$

The cycle efficiency is therefore:

$$\eta_c = \frac{104.27}{104.27 + 433.39} = .1939$$

APPENDIX C
SUMMARY OF TURBINE DESIGN
METHODS AND PARAMETERS

Mechanical Design Procedure

1. Make use of standard and conventional parts and materials.
2. Estimate rotor component weights.
3. Calculate rotor shaft size by:
 - A. Minimum static deflection with load (.001")
 - B. Bearing size for rotor speed and load
 - C. Material selection.
4. Calculate shaft stresses based on size, material, and drive horsepower.
5. Estimate first critical speed.
6. Adjust any of the above parameters as required.
7. Determine component balance required to minimize total load.
8. Calculate housing stresses and determine minimum thicknesses and bolt size.
9. Calculate inlet line size and outlet line size required for standard flow.
10. Design support members for maximum rigidity.

Turbine Parameters*

1. Two inlet nozzles - .939" exit diameter
2. Steam mass flowrate = .659 lb/s
3. DTA mass flowrate = 2.16 lb/s
4. Separator rpm = 16,322
5. Turbine rpm = 8,161
6. Separator speed = 1,600 ft/s

*The final turbine parameters changed somewhat due to iterations, but not significantly enough to affect the mechanical design.

7. Turbine power = 160 HP
8. Steam turbine power = 137.4 HP
9. Pump power = 38.7 HP @ 75% efficiency
10. Nozzle exit velocity = 1,841 ft/s (@ 15°)
11. Separator (nozzle exit) pressure = 14.7 psia
12. Separator (nozzle exit) temperature = 212°F
13. Turbine discharge pressure = 1.2 psia (.7 psia minimum)
14. Separator drum inner diameter = 24"
15. Nozzle length = 12"
Throat diameter = .90"
Exit diameter = .939"
16. Separator drum material - titanium

Numerical Calculations

1. Estimated rotor weights:

Steam turbine - 20 lbs
 Separator disc - 20 lbs
 Liquid turbine - 5 lbs
 Shaft - 8.5 lbs

2. Rotor unbalance load:

Vapor turbine: 8,161 rpm
 .04 ounce inches maximum

$$\text{Force} = \frac{\text{Ounces} \times \text{inches}}{6172} \times (.1045 \times \text{rpm})^2$$

$$\text{Force} = \frac{.04}{6172} (728,702)$$

$$\text{Force} = 2.72 \text{ lbs}$$

Separator disc: 16,322 rpm
 .016 ounce inches maximum

$$\text{Force} = \frac{.016}{6172} (2,914,809)$$

$$\text{Force} = 7.5 \text{ lbs}$$

Liquid turbine and shaft = Negligible

$$\text{Total Rotor Load} = 65.72 \text{ lbs}$$

3. Rotor shaft size:

$$\text{Deflection} = \frac{Wc^2}{3EI} \times (c + l)$$

$$.001 = \frac{(65.72)(4)^2}{3(30 \times 10^6)l} (4 + 7)$$

$$I = .129 = .049(d^4)$$

$$d = 1.344"$$

where W = Rotor weight - lbs

$$E = 30 \times 10^6$$

$$I = .049(d^4)$$

c = Rotor overhang - inches

l = Bearing spacing - inches

d = Shaft diameter - inches

Bearing Size = 35 mm. bore (1.3780" shaft size)

Bearing Type (test purposes only) - Extra Precision ABEC - 7

*Typical Bearing Type = Fafnir Medium 2MM307 WI single and
2MM307 DB double*

$$\text{Radial Load} = R_e = X_1 R + Y_1 T \text{ Equation 1} = 65.72$$

$$= X_2 R + Y_2 T \text{ Equation 2} = 52.23$$

$$\text{where } X_1 = 1.0$$

$$Y_1 = 0$$

$$X_2 = .45$$

$$Y_2 = 1.8$$

$$R = 65.72$$

$$\text{Life Calculation: } L_{10} = 1,500 \left| \frac{C_u}{R_e} \right|^3$$

$$\text{where } C_u = i^{0.7}(C_n) = 2,505.6$$

$$\text{and } C_n = n_f \times C_b$$

$$C_n = .16 \times 7,250$$

$$C_n = 1,160$$

$$\text{Therefore, } L_{10} = 82 \times 10^6$$

Standard grease packed double seal bearings rated at 250°F are therefore suitable.

4. Rotor shaft size by stress:

ASME Power Code Equation: =

$$d^3 = \frac{16}{S_s} \sqrt{(K_b M_b)^2 + (K_t M_t)^2}$$

where d = Shaft diameter - inches

S_s = Maximum shear stress - torsion, bending and axial, Code for standard commercial with keyway = 6,000 psi

K_b = Combined shock and fatigue - bending

M_b = Bending moment - in/lb

K_t = Combined shock and fatigue - torsional
Code - 1.5 for minor shock

M_t = Torsional moment - in/lb

Neglecting bending stresses: @ 300 HP

$$d^3 = \frac{16}{S_s} K_t M_t$$

$$\begin{aligned} \text{where } M_t &= \frac{HP(33,000)(12)}{2\pi \text{ rpm}} \\ &= 2,318 \text{ in/lb} \end{aligned}$$

$$d^3 = \frac{16}{6,000} 1.5(2,318)$$

$$d^3 = 2.95''$$

$$\text{Therefore, } d = 1.37''$$

The 35 mm. bearing size (1.3780) is correct.

APPENDIX D

STEAM TURBINE ANALYSIS AND DESIGN

The two-phase nozzle accelerates a steam/Dow-Therm A mixture. In the separator, the two fluids are separated and each is allowed to exchange momentum in a turbine wheel.

In the separator process, the liquid Dow-Therm A is centrifuged to the inside of the rotating separator rim. The steam, at essentially rim speed, exists below the liquid. If the pressure gradient is sufficient to initiate flow radially inward, the flow can be directed through properly shaped vanes and energy can be extracted. This would then be a radial inflow turbine configuration, and is selected here because it adapts so well to the rest of the geometry. Actually, a small performance penalty will be paid because the jet-speed ratio (U/C_o) is somewhat low, even if two stages are used.

Operating Conditions

The conditions for which the turbine was designed are listed in Table D-1.

Analysis

In contradistinction to the ordinary radial inflow turbine where the working fluid is accelerated in a regular nozzle located in the plane of the wheel and directed to provide a radial inflow component to the velocity, here the fluid is separated on the rotating drum and is directed with a slight axial component. Therefore, the analysis begins with the already accelerated steam jet assuming a realistic value of velocity (tangential component = 1,800 ft/sec, radial component = 250 ft/sec). The steam is also assumed to be saturated vapor at 14.7 psia. The

TABLE D-1
OPERATING CONDITIONS

Nozzle Exit Diameter, in.	0.939
Steam Mass Flowrate, lb/sec	0.659
Dow-Therm Mass Flowrate, lb/sec	2.16
Separator Speed, rpm	16,322
Turbine Speed, rpm	8,161
Nozzle Exit Velocity, ft/sec	1,841
Nozzle Angle, degrees	12
Separator Diameter, in.	24.0
Nozzle Exit Pressure, psia	14.7
Turbine Discharge Pressure, psia	1.2 maximum 0.7 minimum

separator and nozzle diameters set the allowable inlet diameter to the turbine at 22.0 inches, making the tip speed 783 ft/sec at 8,161 rpm. This is the starting point of the analysis.

The variables which operate in the trade-off analysis include:

- Diameter at inlet and outlet of each stage
- Intermediate stage pressures
- Blade angles
- Absolute velocities (mach numbers)
- Radial acceleration permissible

The objective, of course, is to minimize losses and maximize power output. In terms of the turbine analysis, the absolute leaving velocity should have the smallest possible tangential (whirl) component.

A preliminary trade-off was made to determine some of the variables. Diameters were selected and the radial acceleration was set at 20 percent. It would be best to have no acceleration but blade angles will be very shallow. Throughflow velocity was chosen to give reasonable blade height. A parametric analysis of intermediate stage pressures was performed.

Table D-2 lists the constraints imposed on the analysis. Figure D-1 shows the nomenclature used. Table D-3 is a summary of the calculations and Figures D-2 through D-5 show the results. From these data the selected configuration is as summarized in Table D-4.

TABLE D-2
CONSTRAINTS AND EQUATIONS

$C_{u2} = 1,800 \text{ ft/sec}$	Assumed, set by drum speed
$C_{r2} = 250 \text{ ft/sec}$	Assumed, reasonable blade height
$w_{3i}^2 = w_2^2 + 2gJ(h_2 - h_{3i})$	First Law
$w_3 = 0.9w_{3i}$	Assumed, Losses
Stage Work = $\frac{C_{u2} U_2 - C_{u3} U_3}{gJ}$	Euler Equation
$C_{r3} = 1.2 C_{r2} = 300 \text{ ft/sec}$	Reasonable acceleration
$C_{u4} = C_{u3} (D_4 / D_3)$	Vortex Law
$C_{r5} = 1.2 C_{r4} = 360 \text{ ft/sec}$	Reasonable acceleration
$C_{5i}^2 = C_4^2 + 2gJ(h_4 - h_{5i})$	First Law
$C_5 = 0.9 C_{5i}$	Assumed, Losses
$h_6 = h_4 - C_o^2 / 2gJ$	First Law
$C_{r7} = 1.2 C_{r6} = 432 \text{ ft/sec}$	Reasonable acceleration
Stage Work = $\frac{C_{u6} U_6 - C_{u7} U_7}{gJ}$	Euler Equation
Total Work = Sums of Stage Work	
Power = Total Work x Flowrate	

Fig. D-1 - Typical Diagrams

TYPICAL INLET TO ROTOR OUTLET
FROM STATOR VELOCITY DIAGRAM

TYPICAL OUTLET FROM ROTOR INLET
TO STATOR VELOCITY DIAGRAM

TABLE D-3
TYPICAL CALCULATIONS

Point 2

$$\begin{aligned} p_2 &= 14.7 \text{ psia} \\ h_2 &= 1,150.5 \text{ B/lb} \\ C_2 &= 1,817.3 \text{ ft/sec} \\ U_2 &= 783.4 \text{ ft/sec} \\ \omega_2 &= 1,046.9 \text{ ft/sec} \\ C_{u2} &= 1,800 \text{ ft/sec} \\ C_{r2} &= 250 \text{ ft/sec} \end{aligned}$$

Point 3

$p_3 =$	14.7 psia	12.0	8.0	6.0	4.0	1.2
$h_{3i} =$	1,150.5 B/lb	1,136	1,108	1,089	1,063	992
$\omega_3 =$	- 942 ft/sec	-1,215	-1,616	-1,845	-2,112	-2,706
$U_3 =$	712 ft/sec	712	712	712	712	712
$C_{r3} =$	300 ft/sec	300	300	300	300	300
$C_{u3} =$	- 181 ft/sec	- 465	- 876	-1,108	-1,379	-1,977
Stage Work =	61.4 B/lb	69.5	81.2	87.8	95.5	112.5

Point 4

$C_{u4} =$	- 181 ft/sec	- 470	- 885	-1,119	-1,393	-1,997
$C_{r4} =$	300 ft/sec	300	300	300	300	300

TABLE D-3 - continued

Point 5

			<u>p₃ = 12.0</u>		<u>p₃ = 8.0</u>		
p ₅	=	10 psia	6	1.2	6	4	1.2
h _{5i}	=	1,132 B/lb	1,097	999	1,115	1,088	1,015
C ₅	=	883 ft/sec	1,490	2,485	1,342	1,750	2,557
C _{r5}	=	360 ft/sec	360	360	360	360	360
C _{u5}	=	806 ft/sec	1,446	2,459	1,293	1,712	2,531
C _{u6}	=	815 ft/sec	1,462	2,487	1,307	1,732	2,560
U ₆	=	627 ft/sec	627	627	627	627	627
ω _{u6}	=	188 ft/sec	835	1,860	681	1,105	1,933
C _{r6}	=	360 ft/sec	360	360	360	360	360
ω ₆	=	406 ft/sec	909	1,895	770	1,162	1,966
C ₆	=	891 ft/sec	1,506	2,513	1,356	1,770	2,585
h ₆	=	1,135 B/lb	1,106	1,027	1,116	1,090	1,021
p ₇	=	1.2 psia	1.2	1.2	1.2	1.2	1.2
h _{7i}	=	1,003 B/lb	1,006	1,027	1,017	1,018	1,021
ω ₇	=	-2,338 ft/sec	-2,171	-1,706	-2,121	-2,012	-1,769
U ₇	=	556 ft/sec	556	556	556	556	556
C _{r7}	=	432 ft/sec	432	432	432	432	432
C _{u7}	=	-1,742 ft/sec	-1,572	-1,094	-1,520	-1,409	-1,160
Stage Work = 59.0 B/lb			71.4	86.5	66.4	74.6	89.3
Power	=	120 HP	132	146	137	145	159

Fig. D-2

Fig. D-3

Fig. D-4

TABLE D-4
CONFIGURATION DATA

FIRST STAGE	Inlet	$p = 14.7$ psia
		$D = 22.0$ in.
		$b = 0.185$ in.
		$\beta = 13.8$ degrees
	Exit	$p = 8.0$ psia
		$D = 20.0$ in.
STATOR	Inlet	$b = 0.296$ in.
		$\beta = 10.7$ degrees
		$p = 8.0$ psia
		$D = 19.8$ in.
	Exit	$b = 0.300$ in.
		$\alpha = 18.7$ degrees
SECOND STAGE	Inlet	$p = 4.0$ psia
		$D = 17.8$ in.
		$b = 0.514$ in.
		$\alpha = 11.9$ degrees
	Exit	$p = 4.0$ psia
		$D = 17.6$ in.
	Inlet	$b = 0.520$ in.
		$\beta = 18.1$ degrees
		$p = 1.2$ psia
		$D = 15.6$ in.
	Exit	$b = 1.49$ in.
		$\beta = 12.49$ degrees

Fig. D-5 - Results of Trade-off Study