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ANALYSIS OF A COMBINED REFRIGERATOR-GENERATOR SPACE POWER SYSTEM

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TECHNICAL PAPER proposed for presentation at
Cryogenic Cooler Conference sponsored by U. S. Air Force
Colorado Springs, Colorado, October 16-17, 1973

E-7683

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INTRODUCTION

Separate closed-Brayton-cycle systems for cryogenic cooling and electric power generation are being advanced for use in space. With some operating compromises, these systems could be combined* through a common compressor and a common working gas. A dual-function refrigeration and power system may be advantageous, especially for space use. It would eliminate some components. And, if the operating compromises are not too severe, the dual system might be lighter and more efficient than separate systems.

Typically, cryogenic coolers use helium as the working gas. In contrast, the electric power generators generally use inert gases of higher molecular weight. For a dual function system, there are only two feasible inert gases if cooling is needed below 100° R; namely, helium or neon. The choice of helium would make it possible to achieve very low refrigeration temperatures. However, multi-staged turbomachinery would be needed. The choice of neon would allow the use of simpler, single-stage turbomachinery. But, since the critical temperature for neon is 80° R, as compared to 9° R for helium, the cryo-cooling temperature range would be more restrictive.

The purpose of this study was to take a first-look at the cryo-cooling/electric output capacity and approximate weight of a dual-function space system. For this analysis, neon was assumed as the

*The dual refrigeration and power cycle concept is not new. In 1960, James LaFleur (ref. 1) proposed the dual cycle for use in air-liquification plants. Helium was the working gas and the total output of LaFleur's system was refrigeration. He intended the power-loop, or gas turbine, to be the prime mover for the refrigeration loop. A prototype gas turbine was built (ref. 2) and operated by LaFleur for over 5000 hours. Unfortunately, the cost of building this large tonnage system was not competitive with existing systems.

common working gas with cryo-cooling to 90° R. And, because the analysis was preliminary, the author did not seek a dual system of the least possible weight. Rather, an evaluation was made with a reasonable set of operating conditions that should result in a relatively low weight system.

At the start of the analysis a set of operating conditions, typical of individual power and refrigeration systems, was assumed. The power-loop conditions were the same as those of a 500-to-2000 watt power system (ref. 3) being investigated at NASA-Lewis. As a result of tradeoffs for dual system operation with neon, some of the initially assumed turbomachinery conditions were modified. Then, for the modified set of dual system operating conditions, potential ranges in design-point electric power output and cryogenic cooling capacity were calculated. Some weight estimates were made for the dual system. The sensitivity of the dual system output capacity to the assumed cryo-loop parameters also was studied.

Gross comparisons of performance were made between the dual system and separate power and refrigeration systems. Although the comparisons are approximate, they do allow a first-order judgment on relative performance.

SYSTEM DESCRIPTION

A temperature-entropy diagram for the dual power system assumed in this study is shown in figure 1(a). Both the cryo-loop and power-loop use simple, recuperated, closed Brayton cycles. A single power turbine, a single cryo-turbine and a common compressor are used. The prime energy source for the dual system is assumed to be a radioisotope heat source. Radiators in both loops reject unused cycle heat to space.

Two rotating machinery arrangements were studied: a single-shaft arrangement and a two-shaft arrangement. The schematic diagram of figure 1(b) shows the two-shaft arrangement. The single-shaft arrangement, which is not shown, consisted of a combined assembly of the power turbine, alternator, compressor, and cryo-turbine. Self-acting gas bearings were assumed for both rotating machinery arrangements. The turbines and compressor for each of the arrangements were of single stage, radial-flow design.

PROCEDURE

Because of the large number of independent parameters, a computer program was written to study the thermodynamics of the dual system. The computer program consisted of more than a simple cycle analysis. Turbine and compressor efficiency models that reflect changes in turbomachinery speed, size, and gas conditions were used. Shaft leakage mass-flow rates, shaft losses from bearing friction and alternator windage, and electromagnetic losses were accounted for in the analysis. Options were built into the computer program to study either the one- or two-shaft turbomachinery arrangements. In addition, a subroutine was used that perturbed selected parameters and found those values which resulted in the largest dual power/cooling output capacity.

Perfect gas relations were used in the analysis. A check of the National Bureau of Standards' temperature-entropy chart for neon showed that this assumption was reasonable near 90° R and at moderate (1 or 2 atmospheres) pressures. The neon specific heat at constant pressure in this region was constant and close to the ideal gas value.

The initial set of design conditions is given in table I. They are divided among those for the power loop, the cryo-loop, and those which were common to both loops. The power-loop and common parameter values were those chosen by AiResearch (ref. 4) for the design of the 500-to-2000 watt power system being investigated by NASA-Lewis. These values gave a minimum power system weight. However, the separate power system used a helium-xenon working gas mixture, rather than neon. The 7200 watts of gross thermal power corresponds to the use of 3-MHW (Multi-Hundred-Watt) radioisotope heat sources. This type of heat source is being developed by the AEC (ref. 5). The effective radiator heat-sink temperature of 450° R is a typical design-study value for a cylindrical radiator in low-earth orbit.

In the power loop, a shaft leakage mass-flow rate of 2 percent of the compressor total flow rate was assumed (table I). This leakage flows from the compressor rotor exit through the bearing and alternator cavities and reenters the main flow at the inlet of the power-turbine rotor. In the cycle calculations, none of the shaft losses were assumed to be transferred to the leakage flow.* The leakage was assumed to enter the power-turbine at the compressor-outlet temperature which results in a reduction of the power-turbine-inlet temperature. These leakage flow assumptions are conservative. If part of the shaft losses are transferred to the leakage flow, system performance is improved.

The cryo-loop design parameter values in table I were chosen to be representative of current refrigeration system developments and within the capabilities of the proposed dual system. High values of recuperator heat transfer effectiveness are needed for refrigeration and a value of 0.99 was assumed. The alternator electromagnetic efficiency of 0.98 has been achieved (ref. 6) with high-speed permanent magnet rotors.

During the analysis, the values of power-loop shaft speed and compressor specific speed were studied for dual system operation with neon and were modified from the values listed in table I. Compressor pressure ratio was treated as an independent parameter throughout the analysis. And, although most results are presented for a cryo-cooling

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*Electromagnetic and shaft viscous losses were assumed to be transferred to the neon flow in the alternator heat exchanger (fig. 1(b)).

load of 40 watts and a heat-source gross thermal power of 7200 watts, these parameters were studied over a range of values. Other values were studied for the cryo-loop parameters of cooling temperature, recuperator effectiveness, and pressure drop to determine the sensitivity of the dual system output capacity to the assumed values.

RESULTS

Compatibility of the Brayton power and refrigeration cycles is shown in figure 2. Performance of the power- and cryo-loop is plotted against compressor pressure ratio for the conditions listed in table 1. The power- and cryo-loop performance was determined by dividing the compressor work between the two loops in proportion to mass flow rate. Cycle efficiency is defined as the ratio of shaft power available in the power loop to the net thermal input from the heat source. The available shaft power is the excess of the power turbine work over the compressor work needed in the power loop. Coefficient of performance is defined as the cryo-cooling load divided by the shaft power supplied to the cryo-loop. The supplied shaft power is the compressor work used in the cryo-loop minus the cryo-turbine work. The best power-loop efficiency, for the conditions of table 1, occurred at a compressor pressure ratio of 1.8. And the best cryo-loop performance was at a compressor pressure ratio of 2.1. Good individual cycle performance occurred, however, over a fairly large range in compressor pressure ratio. For example, neither cycle was more than 10 percent from its best performance over a range in compressor pressure ratio from 1.63 to 2.25. In general, however, there will be one pressure ratio for each set of dual cycle conditions that gives the most combined power and cooling capacity.

Turbomachinery Tradeoffs

Figure 3 presents a comparison of performance between the single- and two-shaft turbomachinery arrangements for the cycle conditions of table 1. For the two-shaft turbomachinery, the cryo-turboalternator shaft speed also is constant at 52 000 rpm. Net electric power output* is plotted against compressor pressure ratio. The solid curve shows the capacity of the two-shaft arrangement. And the dashed curves show single-shaft turbomachinery outputs for three values - 0, 0.5, and 1 percent - of cryo-shaft leakage.

In the two-shaft turbomachinery arrangement, there is no shaft leakage flow in the cryo-turboalternator as in the power loop machinery. However, there are cryo-turboalternator shaft viscous and electromagnetic losses. These losses were assumed to be transferred to the neon at the cryo-turbine exit. This assumption lowered the overall cryo-turbine efficiency by raising the exit temperature.

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*Dual system control power needs were assumed to be 40 watts plus 2 percent of the alternator gross output. Net system electrical output then is defined as the gross power at the alternator terminals minus the assumed control power needs.

In the single-shaft turbomachinery arrangement, there was no thermal penalty in cryo-turbine efficiency; however, there would be an additional leakage flow. This shaft leakage would flow from the cryo-turbine rotor inlet through the cryo-bearing and reenter the main flow at the compressor inlet. Because of the low temperature and restricted flow passages, there may be some shaft cooling due to the Joule-Thompson effect. In the cycle analysis, it was assumed that any Joule-Thompson cooling would be balanced by the bearing friction loss. And the cryo-leakage flow was assumed to enter the compressor at the cryo-turbine inlet temperature, reducing the compressor-inlet temperature.

For the single-shaft turbomachinery arrangement, net output was quite sensitive to the amount of shaft leakage (fig. 3). Since cooling load is constant at 40 watts, the largest dual system capacity occurs at the pressure ratio which results in the highest net electric output. Comparing largest capacities in figure 3 shows that 1 percent leakage flow reduced total capacity by about 29 percent from the idealized zero leakage case. There are two opposing effects of cryo-shaft leakage. One effect is the compressor-inlet temperature reduction. This improves the thermal efficiency. The opposing effect is that the leakage requires more compressor mass-flow to achieve the same amount of cooling. The penalty of additional compressor work through the increased flow was much greater than the benefit of reduced compressor-inlet temperature. If shaft leakage could be kept to about 0.4 percent, both machinery arrangements would give the same total capacity. However, the author believes that no less than 1-percent leakage flow could be achieved. Comparing largest capacities shows that the two-shaft arrangement gave about 18 percent more capacity than the single-shaft arrangement with 1 percent shaft leakage. Therefore, two-shaft turbomachinery was selected as the preferred arrangement.

Before continuing, it is interesting to note the performance differences between the zero-leakage single-shaft machinery and the two-shaft machinery. The zero-leakage curve is ideal, while the two-shaft machinery curve has the penalty of the cryo-turboalternator losses. Comparison then shows that the cryo-shaft losses resulted in about a 9-percent loss in total capacity.

Selection of power-loop shaft speed and compressor specific speed was made from the results shown in figure 4. The selections were made by trading-off output capacity and design pressure level. The machinery parameters affect the system's design pressure levels and that in turn influences heat exchanger and system weight. In figure 4(a), net electric power output is plotted as a function of compressor pressure ratio and compressor specific speed. In figure 4(b), net electric power output is plotted as a function of compressor pressure ratio and power-loop shaft speed. In both parts of the figure, values of compressor-inlet pressure are cross-plotted on the results. In the results of figure 4 and the remaining figures, the cryo-turboalternator shaft speed was optimized within the computer program to provide the best possible combined output.

For space power systems, it is desirable to design for a low specific weight (pounds/watt) system. Therefore, for any given working gas, a high pressure level is desired to reduce heat-transfer-component weights, but not so high as to seriously reduce output capacity. The compressor-inlet pressures shown in figure 4 for the dual system with neon are low.* And, fairly large reductions in output capacity may be needed to get low specific weights.

Compressor-inlet pressure increases with decreasing compressor pressure ratio, decreasing specific speed, and increasing power-loop shaft speed. The rate of increase in compressor-inlet pressure with decreasing pressure ratio is such that compressor-outlet pressure increases. For the fixed conditions of figure 4(a), the largest possible output capacity occurred at a compressor pressure ratio of 1.78 and a compressor specific speed of 0.80. Further increases in specific speed at any pressure ratio resulted in lower outputs. For the range of specific speeds in figure 4(a) and at any constant value of pressure ratio, the rate of decrease in output capacity was nearly linear with the rate of increase in compressor-inlet pressure. As an arbitrary compromise, a compressor specific speed of 0.54 was chosen for the dual system. Comparing largest output capacities, the selection of 0.54 rather than 0.80 results in a 13-percent decrease in capacity and an increase of compressor-inlet pressure by a factor of 2.1.

Power-loop shaft speed (fig. 4(b)) was studied only at values higher than the initially assumed speed of 52 000 rpm. Lower speeds would result in somewhat greater output capacities. However, the lower speeds, at any value of pressure ratio, also would decrease compressor-inlet pressure. For the range of power-loop shaft speeds in figure 4(b) and at constant pressure ratio, the rate of decrease in output capacity was also nearly linear with the rate of increase in compressor-inlet pressure. In this case, a power-loop shaft speed of 72 000 rpm was chosen for the dual system. In comparison to the 52 000 rpm results and at largest capacities, the selected speed caused about a 12 percent decrease in capacity with an increase in compressor-inlet pressure by a factor of about 1.8.

In a more detailed study, the choices in specific speed and shaft speed would be re-examined and perhaps traded-off quantitatively with changes in dual system specific weight.

Dual System Output Capacity

Figure 5 presents the dual system performance. Part (a) of the figure shows net electric power output plotted against compressor pressure ratio for a range of constant cryo-cooling loads. Resulting

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*Relatively low system pressures are beneficial to the range of refrigeration temperatures that can be obtained. The normal boiling temperature for neon is 49° R. Hence, cryo-temperatures lower than the 90° R considered in this study would be possible.

values of compressor-inlet pressure are cross-plotted on the figure. Part (b) of the figure shows the dual system prime radiator area plotted against compressor pressure ratio and as a function of the cryo-cooling load.

The largest output capacity, figure 5(a), with no cooling load was about 1980 watts of electricity and occurred at a compressor pressure ratio of about 1.7. As cooling load increased, the pressure ratio for largest output capacity increased. At a cooling load of 40 watts, a pressure ratio of about 1.8 resulted in the most output. There, 1280 watts of net electrical output were available. At 100 watts of cooling, a pressure ratio of about 1.9 resulted in 360 watts of electricity. And, in general at largest capacities, the rate of change in electric output with cooling load was nearly constant at about -15 watts electric per watt of cooling.

The lowest specific weight solution for these power levels should occur at or below the compressor pressure ratios which result in the largest output capacities. Compressor-inlet pressure at the largest capacity conditions was near 6 psia. This pressure increased rapidly at lower pressure ratios. The increased pressure by itself will decrease heat exchanger size. However, the decrease in efficiency with decreasing pressure ratio will tend to increase heat exchanger size.

The trends of changes in radiator size are shown in figure 5(b). Prime radiator area was calculated neglecting the temperature drop between the local gas temperature and the radiating surface. Prime area is ideal but still useful for showing trends. Radiator area needs increase with decreasing pressure ratio and increasing cryo-cooling load. The dashed line shows the area change for those pressure ratios which gave the largest output capacities. Detailed radiator calculations at a constant cooling load should show a least possible radiator weight at some low value of compressor pressure ratio because of the opposing effects of radiator area and system pressure.

Figure 6 shows the effects of changes in the number of Multi-Hundred-Watt heat sources on the output capacity of the dual system at the selected conditions. Net electric power output is plotted against cryogenic-cooling load. The 7200 watts of heat-source gross thermal power corresponds to the use of 3-MHW heat sources, while 4800 would be 2 MHW heat sources, and 2400 would be a single heat source. Two output capacities are shown for each case. The solid lines show results for the largest output capacities. The dashed lines show results for a constant compressor pressure ratio of 1.5.

The largest cooling loads were 123 watts with 3-MHW sources, 80 watts with 2, and 36 watts with 1. The largest electric outputs were 1980 watts with 3 sources, 1280 with 2, and 610 with 1. And in general, the output capacities were nearly directly proportional to the number of MHW heat sources. The rate of change in electric output with increasing cooling load was the same for 3-, 2-, and 1-MHW heat source.

Cryo-Loop Sensitivity Effects

The separate effects of changes in the assumed cryo-loop parameters on the dual system output capacity are summarized in table 2. Results are for those compressor pressure ratios which gave the largest output capacities. The slopes of the curves of electric power output with cooling load and the largest possible cooling loads are presented. Effects of an 18° R increase in the cryo-turbine inlet temperature are shown in part (a). Two decreases in cryo-recuperator effectiveness are shown in part (b). And, two larger pressure drop allowances are shown in part (c). The slopes, or rates of change in electric output with cooling load, given in table 2 are valid above cooling loads of about 20 watts. Below 20 watts, the slopes were larger.

Effects of these parameter changes are somewhat academic, but are presented to give the reader an idea of the sensitivity of output capacity to the selected values. The increase in cooling temperature, part (a), gave a larger output capacity. The rate of change of electric output with cooling load decreased by about 24 percent. And, the largest cooling load increased by about one-third. Output capacity was quite sensitive to the cryo-loop recuperator effectiveness, part (b). A 0.005 drop in effectiveness resulted in a 20 percent decrease in the largest cooling load. And, a 0.01 drop gave a 37 percent decrease. In contrast, output capacity was not as sensitive to the cryo-loop pressure drop, part (c). The increase from 2 to 4 percent pressure drop resulted in a 3 percent decrease in the largest cooling load. The change from 2 to 6 gave a 7 percent decrease.

Dual System Weight Estimates

Dual system weight estimates were made for the selected conditions with a cryo-cooling load of 40 watts. Detailed computer programs were used to size the system recuperators and space radiators. Other system component sizes were estimated from similar equipment. Compressor pressure ratios of 1.5, 1.6, and 1.83 in figure 5 were chosen for the calculations. The effects of pressure ratio on the weight of the recuperators and radiators are shown in table 3. For all but the cryo-loop recuperator, the heat exchanger component weights decreased with decreasing pressure ratio. That is, the pressure effect was greater than the effect of the increased heat load. The reverse was true for the cryo-loop recuperator. The total weights for the recuperators and radiators were 1130 pounds at a pressure ratio of 1.83, 710 pounds at 1.6, and 650 pounds at 1.5. Hence, the least total system weight should occur near a compressor pressure ratio of 1.5. The remaining system components were estimated to weigh about 350 pounds. Therefore, an estimated total dual system weight is about 1000 pounds However, the author believes this weight is conservative.

The recuperator and radiator calculations were made with the same assumptions and configurations as in the separate power system study (ref. 3). The recuperators had a plate-fin core with triangular end sections leading to the headers. Axial conduction effects are important in these designs. And the use of perforated-plate recuperators as in reference 6 should result in weight savings because the perforated-plate approach nearly eliminates axial conduction. The power and cryo-loop radiators were assumed to be separate, with aluminum tube-fin construction. They were cylindrical with a five-foot diameter. Enough armor was provided to give a 0.99 probability of no punctures in five years. The radiator weights were also 10-percent above the least possible weights to favor a smaller required radiator surface area. Some radiator size and weight reductions would occur if the two radiators were combined to share common fins.

The author estimates that with perforated-plate recuperators and a single radiator with shared fins, the weight of these components could be reduced by about 200 pounds. Further design-point optimizations, seeking a minimum weight system, might reduce the total weight by another 100 pounds. Therefore, a more optimistic total dual system weight would be on the order of 700 pounds.

Comparisons with Separate Refrigeration and Power Systems

Performance estimates for separate power and cooling systems were made for the reference output capacity of the dual system; namely, 40 watts of cooling and about 1200 watts of net electric power. The separate power system was sized, based on current estimates for the system of reference 3, to provide 1200 watts plus the electric power needs of the separate cooling system. For the separate cooling system, both a Brayton turborefrigerator and a Vuilleumier (VM) cooler were considered. The VM cooler was assumed to use isotope heating for its hot displacer. Although no cooler now provides 40 watts of cooling, the size was extrapolated from smaller units.

If a separate VM cooler were used, it would need about 3200 watts of isotope heat and about 120 watts of electric power. The VM cooler and its heat source would weigh about 400 pounds. The separate Brayton power system sized for 1320 watts of output would need about 4500 watts of isotope power and would weight about 450 pounds. Therefore, these separate systems would need a total of about 7700 watts of thermal input and would weigh a total of about 850 pounds. Compared to the dual system with 7200 watts of thermal input and a weight of about 700 pounds, there was no first-order difference in performance.

If a separate Brayton turborefrigerator were used, it would weigh only about 100 pounds, but would need about 4000 watts of electric power input to the compressor motor. At a net efficiency of 0.30, the Brayton power system would need about 17 000 watts of isotope power to supply a total of 5200 watts of electricity. The total weight for these separate systems would be at least 1400 pounds. It appears then that the performance of a dual-function Brayton system is better than that of separate Brayton power and refrigeration systems.

CONCLUDING REMARKS

A preliminary analysis was made of a dual-function space power system. This system combined the closed Brayton power and refrigeration cycles through a common compressor. Although the analysis was done for a space system, the concept might be useful in other applications. The study did not seek the best possible dual-function space system, but rather made an evaluation using reasonable design conditions. Neon was assumed as the working gas with cryo-cooling to 90° R.

It was found that a two-shaft arrangement of the power- and refrigeration-loop rotating machinery offered better output capacities than a single-shaft arrangement. With a 7200-watt radioisotope heat source, the largest cooling capacity provided by the selected dual system was about 120 watts. The largest electric output was about 2000 watts. Combined outputs, at a constant heat source capacity, were available at less than the largest cooling or electric outputs. For example, the dual system could provide 40 watts of cooling and about 1200 watts of net electric power. At this output capacity and for the selected conditions, the dual system was conservatively estimated to weigh about 1000 pounds. The author believes that with improvements in components and further design-point optimizations, that the dual system weight might be reduced to about 700 pounds for the same output capacity. With a 4800-watt heat source, the dual system provided up to 80 watts of cooling or up to about 1300 watts, electric. With a 2400-watt source, up to 36 watts of cooling or up to about 600 watts, electric were provided.

Gross comparisons of performance were made between the dual system and separate power and refrigeration systems for the assumed cooling temperature of 90° R. These comparisons showed no first-order difference between dual system performance and that of a separate Brayton power system and a Vuilleumier cooler that uses isotope heating. The comparisons did indicate that the performance of the dual system was better than that of separate Brayton power and refrigeration systems.

In summary then, this analysis has shown the potential performance of a dual refrigeration and power system for spacecraft. Operating compromises do not appear to be prohibitive. A wide range in electric and cryogenic cooling capacities are available. If the cryo-cooling temperature of 90° R used in this analysis is not low enough for some applications, other dual cycle variations could be investigated.

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TABLE 1. - INITIAL DUAL SYSTEM PARAMETER VALUES

POWER LOOP:		
Turbine-inlet temperature	2060	°R
Heat-source gross thermal power	7200	W
Heat-source heat loss	10	%
Recuperator heat-transfer effectiveness	0.975	
Pressure losses	2	%
Alternator electromagnetic efficiency	0.92	
Shaft leakage mass-flow rate	2	%
Shaft speed	52000	rpm
COMMON:		
Compressor-inlet temperature	536	°R
Compressor specific speed	0.445	
Compressor pressure ratio	1.5	
Effective radiator heat-sink temperature	450	°R
CRYO-LOOP:		
Turbine-inlet temperature	90	°R
Cryo-cooling load	40	W
Recuperator heat-transfer effectiveness	0.99	
Pressure losses	2	%
Alternator electromagnetic efficiency	0.98	

TABLE 2. - SEPARATE EFFECTS OF CRYO-LOOP PARAMETERS ON LARGEST OUTPUT CAPACITIES. POWER-LOOP SHAFT SPEED, 72 000 RPM; COMPRESSOR SPECIFIC SPEED, 0.54; OTHER PARAMETER VALUES, TABLE 1

		Rate of change in electric output with increasing cooling load, $W_{(e)}/W_{(t)}$	Largest cooling load (no electric output), W
Cryo-turbine inlet temperature, °R	108	-11.8	163
	90*	-15.5	123
(A.) Sensitivity to cooling temperature			
Cryo-recuperator effectiveness	0.99*	-15.5	123
	0.985	-18.6	99
	0.98	-22.2	77
(B.) Sensitivity to recuperator heat-transfer effectiveness			
Cryo-loop pressure drop, %	2*	-15.5	123
	4	-16.0	119
	6	-16.6	114
(C.) Sensitivity to total pressure drop			

*Indicates assumed system conditions.

TABLE 3. - DUAL SYSTEM COMPONENT WEIGHTS. POWER-LOOP SHAFT SPEED, 72000 RPM; COMPRESSOR SPECIFIC SPEED, 0.54; OTHER PARAMETER VALUES, TABLE 1

	Compressor pressure ratio 1.50	1.60	1.83
POWER LOOP:			
Recuperator, lbs	120	156	309
Radiator, lbs	268	317	607
CRYO LOOP:			
Recuperator, lbs	189	158	125
Radiator, lbs	76	79	85
TOTALS:	653	710	1126

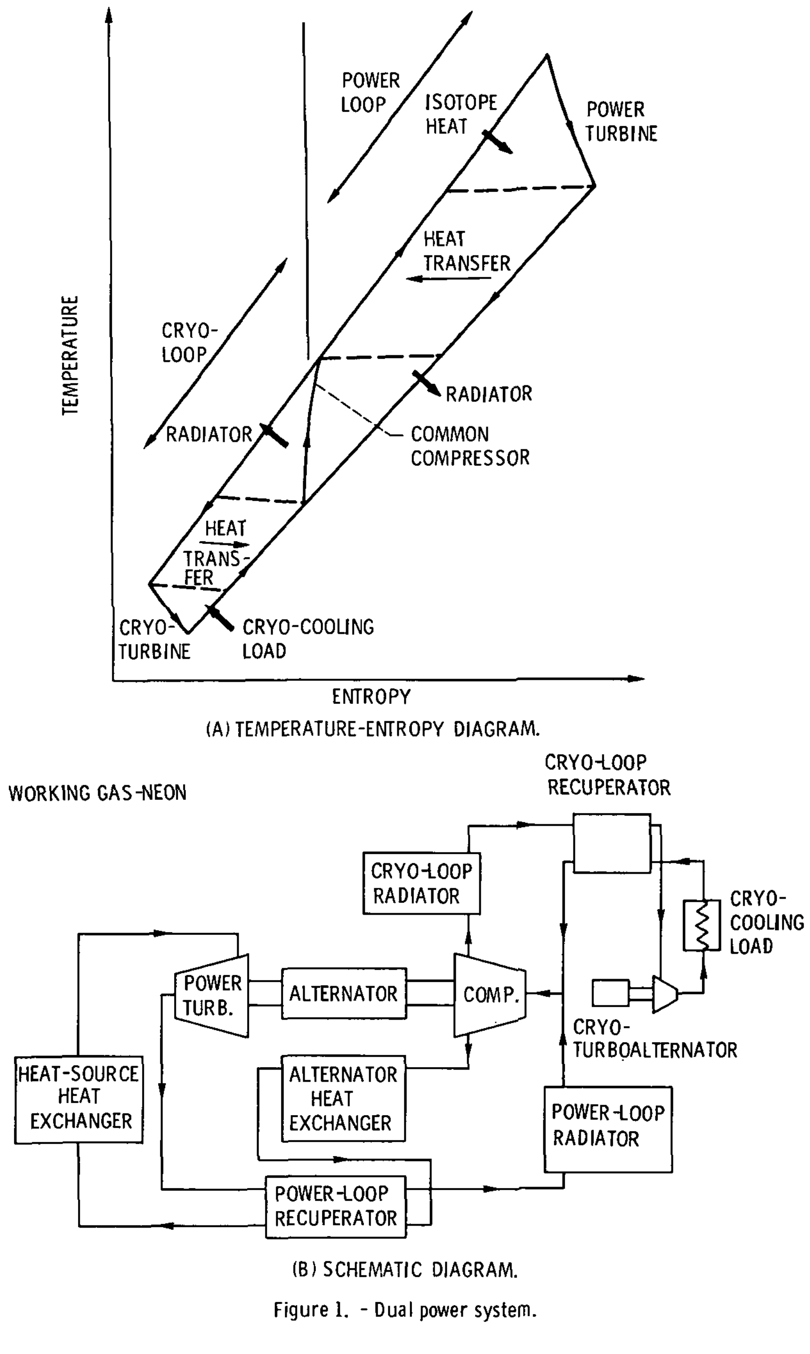

Figure 1. - Dual power system.

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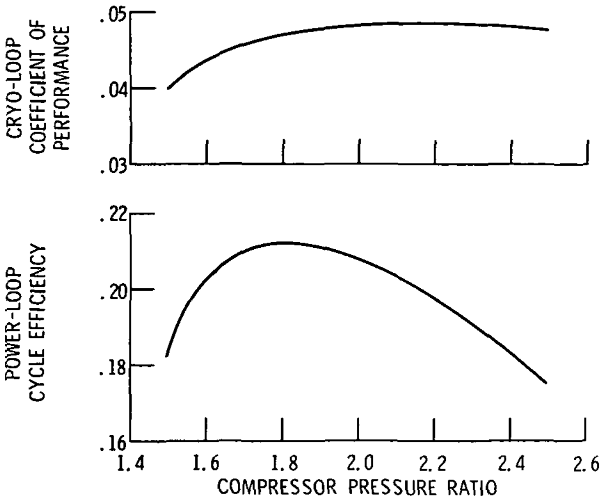

Figure 2. - Comparison of refrigeration and power cycle performance. Other parameter values, table 1.

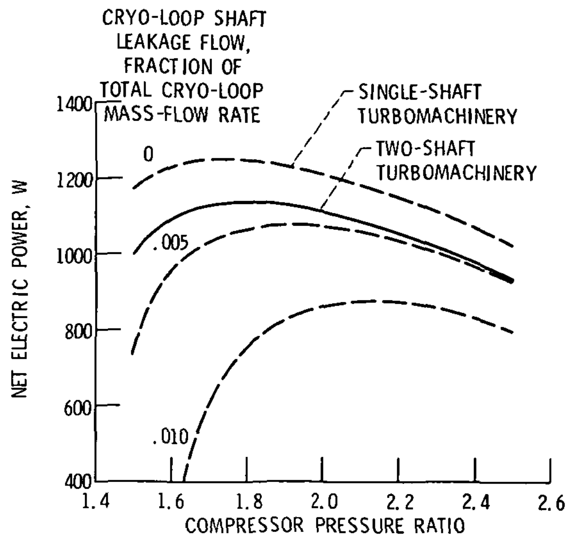

Figure 3. - Comparison of performance with turbomachinery arrangement. Other parameter values, table 1.

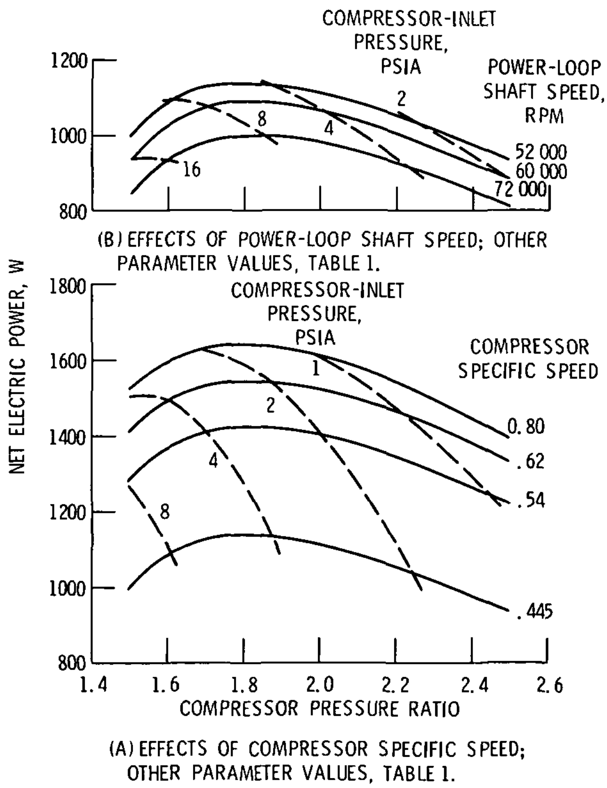

Figure 4. - Turbomachinery trade-offs between output capacity and design pressure level.

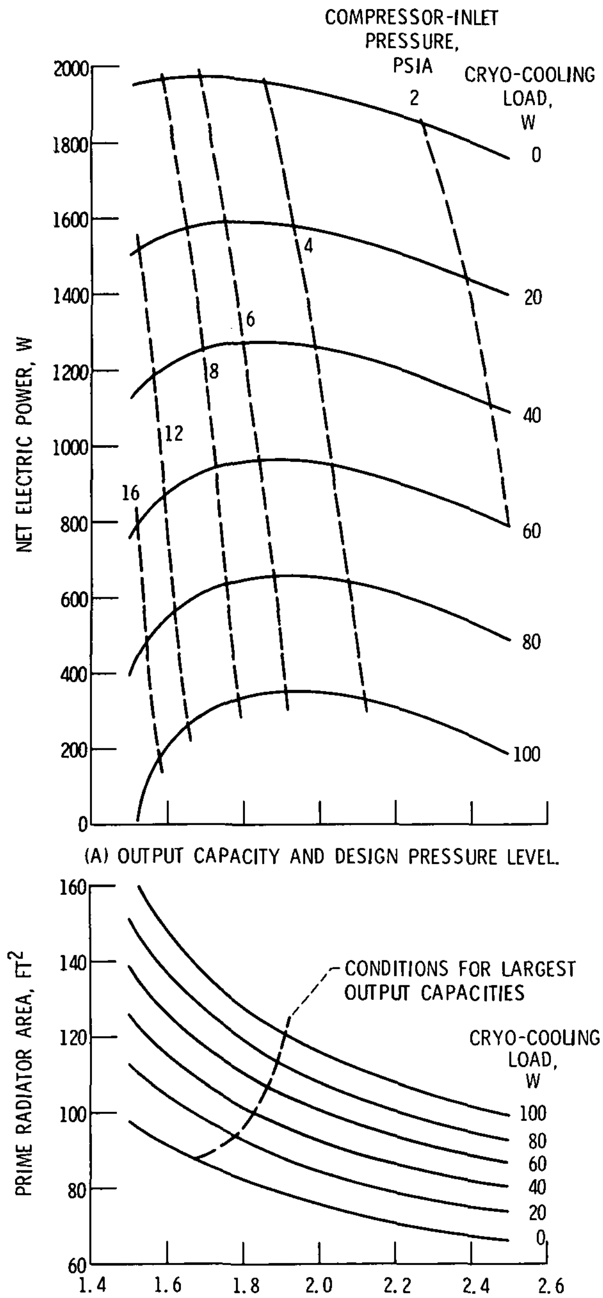

Figure 5. - Dual system performance. Power-loop shaft speed, 72 000 rpm; compressor specific speed, 0.54; other parameter values, table 1.

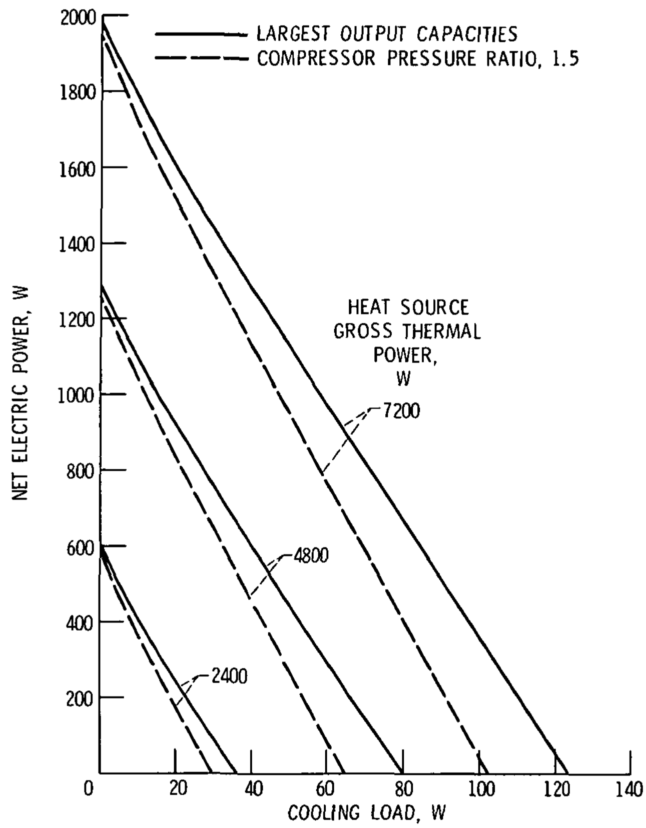

Figure 6. - Effects of heat-source capacity on dual system performance. Power-loop shaft speed, 72 000 rpm; compressor specific speed, 0.54; other parameter values, table 1.

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