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SUMMARY

This study was made for the purpose of exploring thermodynamic and turbomachinery concepts for low-power (2- to 10-kW) radioisotope and intermediate-power (100- to 300-kW) reactor Brayton-cycle systems. The effects of intercooling and reheating on the system were examined. Turbomachinery characteristics for one single- and two dual-shaft arrangements were determined parametrically as a function of rotative speed and included the effects of varying molecular weight, pressure level, and pressure ratio. Several favorable configurations were selected and presented as examples of geometries suitable for these systems.

For a low-power radioisotope system, where the achievement of high cycle efficiency is of prime importance, the maximum increase in cycle efficiency achievable by the use of both intercooling and reheating was about 0.05 out of a total of about 0.35 to 0.40. For a typical intermediate-power reactor system, where the achievement of minimum radiator area is of prime importance, intercooling offers no area reduction but reheating results in about a 15 percent reduction in minimum area.

Comparable size and specific speed designs can be evolved for the radioisotope- and reactor-system turbomachinery when a constant ratio of pressure to power is used. Rotative speeds, however, are considerably higher for the reactor system designs, but these can be reduced, if desired, by reducing the pressure to power ratio. For both systems, suitable turbomachinery can be evolved that will not be significantly different from those currently under experimental investigation for solar Brayton-cycle systems.

INTRODUCTION

Brayton-cycle power conversion systems are being considered for radioisotope (about 2 to 10 kW), solar (about 4 to 40 kW), and nuclear reactor (about 100 to 300 kW) space power systems (refs. 1 to 7). Possible applications include power for scientific satellites, small and large manned space stations, and electric propulsion systems for unmanned probes. In order to evaluate the potential of Brayton-cycle conversion systems for these applications, appropriate system performance characteristics must be determined. Cycle efficiency and/or radiator area are the usual performance characteristics of interest for preliminary system evaluations. The determination of these characteristics depends upon the nature of the selected cycle and cycle variables as well as the operating behavior of the components.

General thermodynamic, component, weight, and reliability characteristics of Brayton-cycle systems were examined at Lewis Research Center, and the results of these studies are summarized in reference 8. An intensive analytical and experimental program in solar Brayton-cycle technology was then undertaken (ref. 9) in order to gain a better understanding of the capability of such a system.

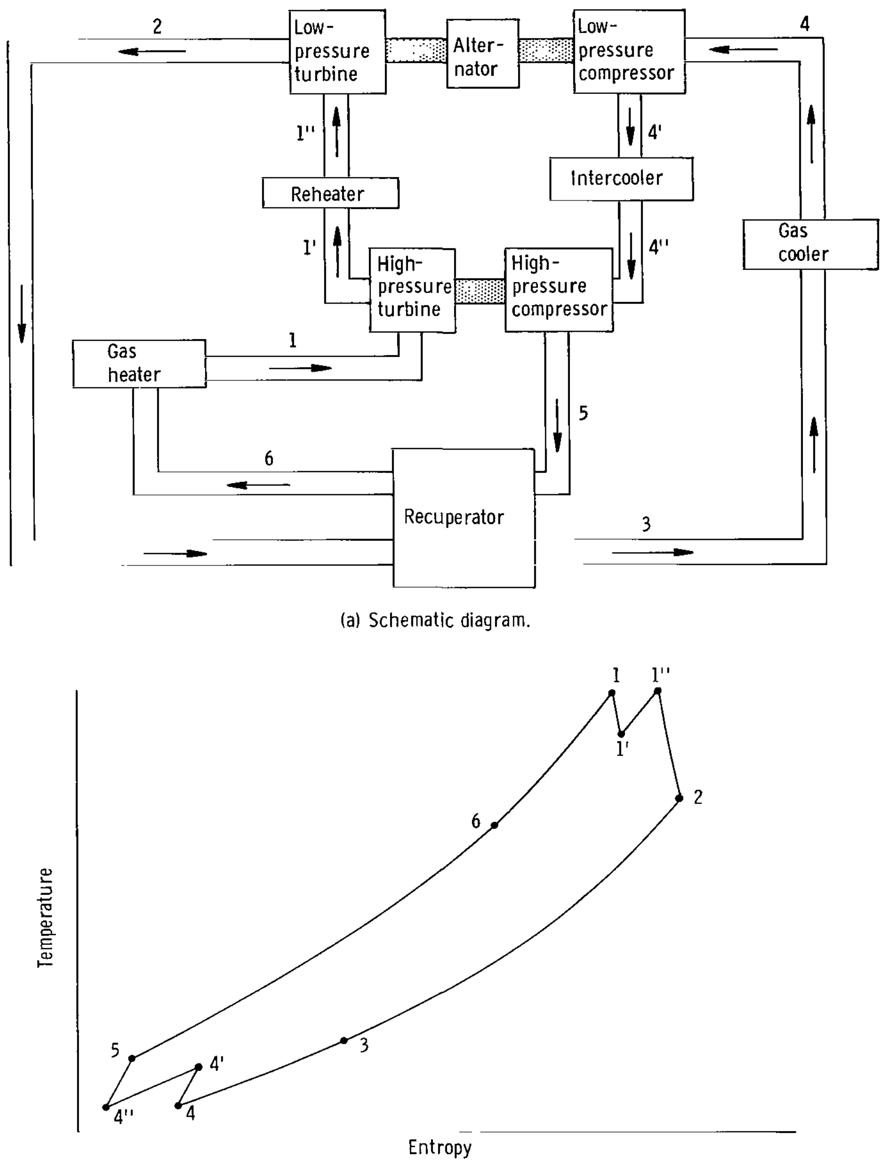
Presented in this report are the results of thermodynamic and turbomachinery concept studies made for low-power radioisotope (2200° R turbine inlet temperature) and intermediate-power reactor (2500° R turbine inlet temperature) Brayton-cycle systems. The use of intercooling and/or reheating were explored as potential methods for increasing cycle efficiency and/or reducing radiator area. Turbomachinery characteristics such as size and specific speed were explored parametrically as a function of rotative speed. The effects of fluid molecular weight, pressure ratio, pressure level, and turbomachinery arrangement on these characteristics were examined. Several favorable configurations were selected and presented as examples of geometries suitable for these systems.

SYSTEM CHARACTERISTICS

The radioisotope and reactor systems under consideration are basically similar except for the nature of the heat source and the level of output power. These differences result in appropriate differences in the selected design parameters as will be discussed later in this section. Also included in this section is a description of the basic system and a discussion of the turbomachinery selection criteria.

DESCRIPTION OF SYSTEM

A typical system with both reheating and intercooling is shown both schematically and thermodynamically in figure 1. The hot gas at point 1 expands through the high-pressure turbine, which drives the high-pressure compressor, to point 1' and is then reheated to point 1". In the low-pressure turbine, which drives the low-pressure compressor and the alternator, expansion to point 2 takes place. The gas is then cooled to point 3 in the recuperator, where heat is transferred to the cooler gas from the compressor, and further cooled to point 4 in the gas cooler. After partial compression to point 4' in the low-pressure compressor, the gas is cooled to point 4" in the intercooler and finally compressed to point 5 in the high-pressure compressor. Heating to



(a) Schematic diagram.

(b) Thermodynamic diagram.

Figure 1. - Brayton-cycle system with intercooling and reheating.

point 6 occurs in the recuperator and final heating to point 1 in the gas heater.

The gas heater and reheater can be either parts of the primary heat source itself or exchangers transferring heat from a heat-transfer-loop fluid to the gas. Similarly, the gas cooler and intercooler can be either gas radiators or exchangers transferring heat to a heat-transfer-loop fluid that carries the excess heat to the radiator.

TABLE I. - SELECTED DESIGN PARAMETERS FOR TYPICAL SYSTEMS

	Radioisotope system	Reactor system
Turbine inlet temperature, T_1, ^{o}R	2200	2500
Turbine efficiency, η_T	0.82	0.85
Compressor efficiency, η_C	.77	.80
Basic loss pressure ratio	.95	.90
Additional loss pressure ratio per reheater or intercooler	.99	.99
Recuperator effectiveness, E	.92	.80
Sink temperature, T_s, ^{o}R	400	400
Emissivity, ϵ	.86	.86
Radiator gas heat-transfer coefficient, h, Btu/(hr)(sq ft prime area)(^{o}R)	10	10
Ratio of reheat to turbine inlet temperatures	1.0	1.0
Ratio of intercool to compressor inlet temperatures	1.0	1.0

DESIGN PARAMETERS

A major difference between the low-power radioisotope and intermediate-power reactor systems is in the selection of certain of the design parameters for the typical systems to be considered. These parameters include turbine inlet temperature, turbomachinery efficiencies, pressure losses, and recuperator effectiveness. The selected values for these parameters as well as several other parameters pertinent to the analysis are shown in table I.

The turbine inlet temperature selected for the radioisotope system (2200^{o} R) reflects a hopefully achievable value considering the nature of the heat source and the use of advanced superalloy technology for the turbine. For the nuclear reactor system, the selected turbine inlet temperature (2500^{o} R) reflects a moderate advance in reactor technology and assumes the use of a molybdenum-alloy rotor for the turbine. The selection of a superalloy rather than a molybdenum alloy for the radioisotope system was based on the more immediate interest in this system and the more advanced state of superalloy technology.

The higher level of turbomachinery efficiency assumed for the reactor system reflects the effect of a higher Reynolds number due to the higher flow rate. This effect is discussed in reference 10. The level of assumed efficiencies is based upon unpublished data.

The selected levels of loss pressure ratio reflect the size and nature of the systems. For the lower-power radioisotope system, the achievement of high cycle efficiencies is of primary importance as a result of isotope cost and availability considerations. At the low power levels associated with these systems, size and weight can be sacrificed in order to increase cycle efficiency. These low-power systems, therefore, can be designed for low pressure losses with the associated relatively large heat-transfer components. For the

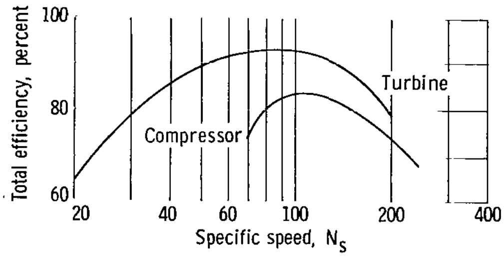


Figure 2. - Variation of total efficiency with specific speed for radial-flow turbomachinery.

intermediate-power reactor systems, size and weight are of major importance while cycle efficiency is of much lesser importance. Somewhat higher pressure losses, therefore, result from the design compromises made in order to obtain more favorable heat-transfer component configurations.

Recuperator effectiveness is selected based on the same philosophy as used for the selection of loss pressure ratio. The low-power radioisotope system can tolerate the relatively large recuperator associated with a high effectiveness as desired to achieve a high cycle efficiency. For the intermediate-power reactor system, a lower effectiveness is required in order to achieve a minimization of system weight.

NATURE OF TURBOMACHINERY

Previous studies (refs. 1, 3, 5, and 10) have shown that radial-flow configurations are suitable for most of the machines required for the applications of interest herein. Single-stage radial machines are compact, rugged, and perform well at the sizes and conditions of interest. For this study, therefore, single-stage radial machines are considered to be the favored configuration wherever their use is feasible. Where multistaging is required for the turbine, axial-flow configurations will be considered so as to avoid the duct losses associated with multiple radial machines on the same shaft.

The selection criterion for the turbomachinery will be specific speed. Efficiency as a function of specific speed for single-stage radial machines (ref. 10) is presented in figure 2. The region of best performance seems to be about 60 to 120 for turbines and 80 to 120 for compressors. From the standpoint of turbine geometry, specific speeds below about 80 may not be as favorable as indicated because of the reduction in blade height with decreasing specific speed. For this study, therefore, the specific speed region of 80 to 120 is considered favorable for single-stage radial turbines and compressors, but the lower limit, if desired, can be compromised somewhat for the turbine. Lower values of specific speed indicate the need for multistaging.

Another consideration for the turbomachinery is the shaft arrangement. With neither reheating nor intercooling, either a single- or dual-shaft arrangement is feasible. The selection would depend upon the desirability of the achievable geometry and performance characteristics as well as on any alternator speed requirements. Relatively low-speed alternators have been specified for some applications in order to utilize state-of-the-art alternator technology. For the parametric analysis presented herein, there is no specification made for alternator speed. With intercooling and/or reheating, dual-shaft systems are considered preferable in order to minimize shaft length and also to gain the extra degree of freedom afforded by the independent selection of a second rotative speed. For these systems, the amount of reheating and/or intercooling desired from a thermodynamic standpoint would also influence the

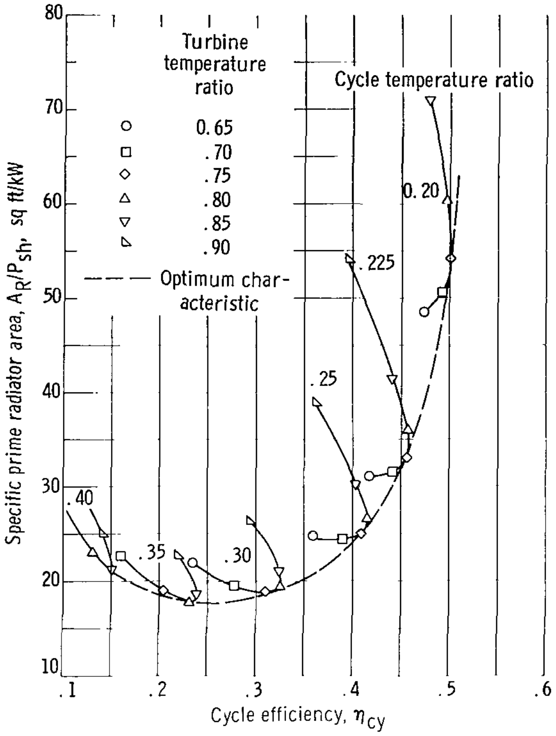


Figure 3. - Effect of cycle temperature variables on radiator area-cycle efficiency characteristics for typical system. Turbine work ratio, 0.4.

nature of the shaft arrangement and work split selection.

METHOD OF ANALYSIS

This study includes both a thermodynamic analysis and a turbomachinery analysis. The purpose of the thermodynamic analysis is to determine cycle performance as a function of the cycle variables and then select an operating point on the basis of the desired performance. Turbomachinery configurations compatible with the selected operating point can be determined by means of the turbomachinery analysis.

THERMODYNAMICS

For space power systems, the major performance criteria are usually radiator area and/or cycle efficiency. Specific prime radiator area (the area of a finless radiator per kilowatt of shaft power) for a gas radiator is used in this analysis as a radiator area indicator; this is quite satisfactory for the purpose of showing levels and trends for either gas or liquid radiators. All symbols used in this analysis are presented in appendix A. The detailed methods for determining the cycle efficiency and specific prime radiator area are presented in appendix B.

The results of the thermodynamic analysis will be presented in the form of radiator area against cycle efficiency with the cycle variables optimized for best performance. Radiator area will represent the sum of the gas cooler and intercooler areas. The optimized radiator area-cycle efficiency characteristics were obtained by a graphical optimization procedure performed as follows. As seen from appendix B, there are three independent variables; they are the cycle temperature ratio T_4/T_1, the turbine temperature ratio T_2/T_1, and the ratio of work in the high-pressure turbine to work in the low-pressure turbine (hereinafter called turbine work ratio) S_T. For each constant turbine work ratio, as shown in figure 3, radiator area-cycle efficiency characteristic curves can be constructed for various cycle temperature ratios and as a function of turbine temperature ratio. An optimum characteristic can then be drawn tangent to the series of cycle temperature ratio curves. This is the optimum characteristic for each turbine work ratio. For each cycle temperature ratio, let the point of tangency be called the optimum characteristic point. Optimum characteristic points are determined for a series of turbine work ratios and are plotted as in figure 4 for the various values of cycle temperature ratio. A curve drawn tangent to this set of curves represents the optimum radiator area-cycle efficiency characteristic for the system with each point on the optimum

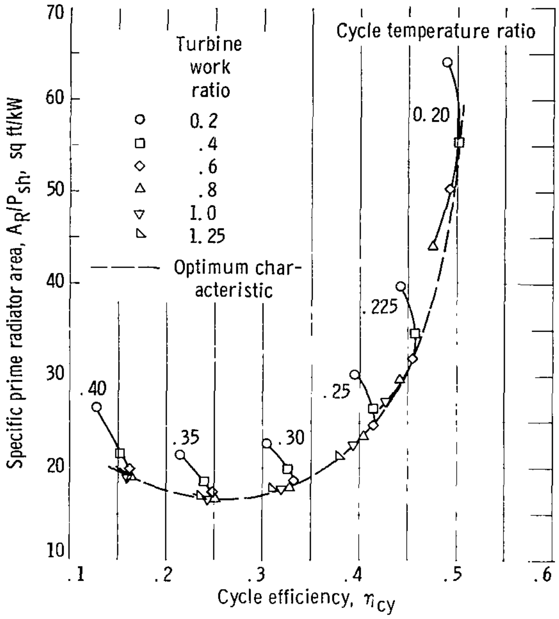


Figure 4. - Effect of turbine work ratio on radiator area-cycle efficiency characteristics for typical system.

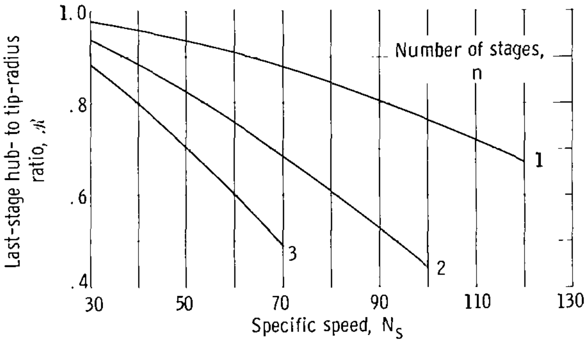


Figure 5. - Effect of specific speed on axial-flow turbine hub- to tip-radius ratio. Turbine efficiency, 0.85.

characteristic curve representing a different combination of the three cycle variables. It is these optimum characteristics that are used in presenting the analysis results.

TURBOMACHINERY

The turbomachinery characteristics of interest are specific speed, as an indicator of performance and geometry, and tip diameter, as an indicator of size. The analysis method is presented in detail in appendix C. Rotative speed is used as the prime independent variable for this analysis. Pressure level, pressure ratio, and fluid molecular weight are additional variables included in the analysis, which will yield results in the form of tip diameter and specific speed as functions of rotative speed for the range of variables and shaft arrangements studied.

As mentioned previously, single-stage radial-flow machines are to be used wherever suitable. Where multistage axial-flow turbines are required, the relation between hub- to tip-radius ratio and specific speed (eqs. (C7) and (C12)) derived in appendix C serves as the basis for selecting the number of stages. Hub- to tip-radius ratio is plotted against specific speed in figure 5 for one, two, and three stages. A desirable range of last-stage hub- to tip-radius ratio is about 0.60 to 0.85. Corresponding to this range are specific speed ranges of about 35 to 60 for three stages, 45 to 80 for two stages, and above 80 for one stage. This will be the stage number selection criteria for the axial-flow turbines studied.

DISCUSSION OF RESULTS

The use of intercooling and/or reheating were explored thermodynamically as potential methods for increasing cycle efficiency and/or reducing radiator area. Turbomachinery characteristics were then explored parametrically for one radioisotope and one reactor system, each without intercooling or reheating.

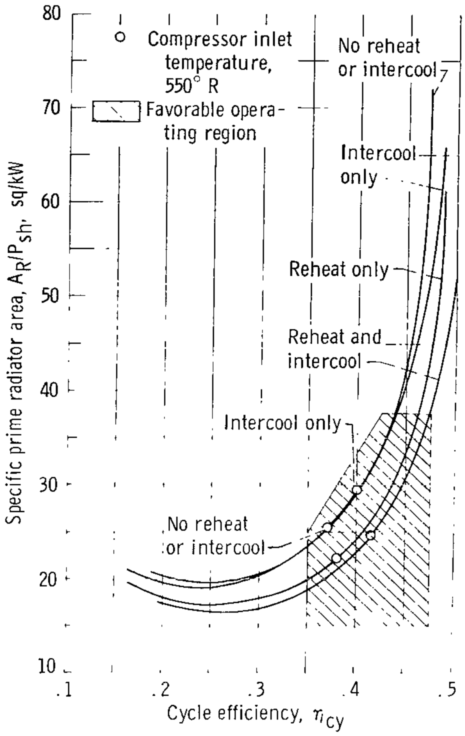


Figure 6. - Typical radiator area-cycle efficiency characteristics for low-power radioisotope Brayton-cycle systems. Turbine inlet temperature, 2200 R.

THERMODYNAMICS

Optimum radiator area-cycle efficiency characteristics were determined, in the previously described manner, for radioisotope and reactor systems with and without reheating and/or intercooling. The selected design parameters for these systems were discussed in the section SYSTEM CHARACTERISTICS and were presented in table I (p. 4).

Radioisotope System

The optimum area-efficiency characteristics for the selected typical system with and without intercooling and/or reheating are presented in figure 6. Since the achievement of high cycle efficiency is of primary importance for the radioisotope system, the desirable operating points would be located in a region somewhere to the right of the minimum area region. The cycle efficiency region of about 0.35 to 0.45 is herein arbitrarily specified as a favorable operating region for the systems represented by figure 6. Movement of this region further to the right results in a very rapid increase in radiator area as a result of the compressor inlet temperature (see fig. 3) approaching the sink temperature.

The effects of reheating and intercooling on system performance can be seen from figure 6. One way to evaluate these effects is on the basis of constant maximum and minimum temperature limits. The optimum characteristic points for a compressor inlet temperature of 550° R ($T_4/T_1 = 0.25$) are shown in figure 6. It can be seen that the general effect of intercooling is an increase in cycle efficiency accompanied by an increase in radiator area, while the general effect of reheating is a smaller increase in cycle efficiency than with intercooling but accompanied by a reduction in radiator area. In particular, a system operating between the temperature limits of 2200° and 550° R without reheating or intercooling can achieve a cycle efficiency of about 37 percent. For the same temperature limits and as compared to the base system (1) the use of intercooling alone results in cycle efficiency being increased by about 3 points (8 percent) with a 16 percent increase in radiator area, (2) the use of reheating alone results in cycle efficiency being increased by about 1 point (3 percent) with a 12 percent reduction in radiator area, and (3) the use of both intercooling and reheating results in cycle efficiency being increased by 4.5 points (12 percent) with a 4 percent reduction in radiator area.

Another way to evaluate the effects of reheating and intercooling on this system is on the basis of constant specific radiator area. On this basis and within the previously specified favorable operating region, it is seen from

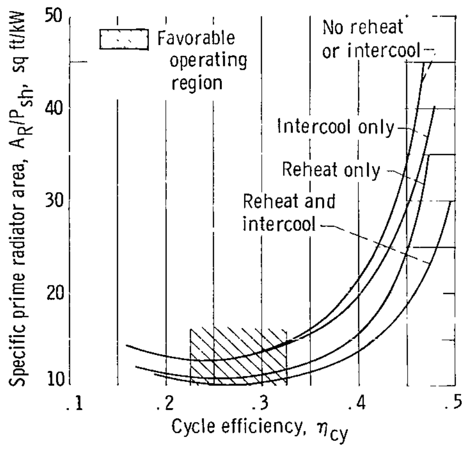


Figure 7. - Typical radiator area-cycle efficiency characteristics for intermediate-power reactor Brayton-cycle systems. Turbine inlet temperature, 2500° R.

figure 6 that intercooling has no effect on efficiency while the use of reheating with or without intercooling results in cycle efficiency being increased by about 3 to 5 points (7 to 13 percent).

Reactor System

The optimum area-efficiency characteristics for the selected typical reactor system with and without reheating and/or intercooling are presented in figure 7. Since the achievement of minimum size and weight is of considerable importance for this system, operation in the minimum area region is desired. For the systems represented by figure 6, the minimum area region corresponds to a compressor inlet temperature of about 750° R ($T_4/T_1 = 0.30$). It can be seen that the use of intercooling offers no reduction in the achievable minimum area. The use of reheating, on the other hand, offers a reduction in minimum radiator area of about 15 percent for these systems.

TURBOMACHINERY

The radioisotope and reactor systems selected for turbomachinery evaluation were those with neither reheating nor intercooling. For those systems with reheating and/or intercooling, the basic levels and trends would be similar to those for the system examined. Selection of typical operating points were made in order to examine specific speed and diameter parametrically as functions of rotative speed as well as for variations in pressure level, pressure ratio, and molecular weight. The specific speed values to be presented are overall values for a given machine and are independent of number of stages or flow path (axial or radial). The diameter values to be presented are tip diameters for one-stage radial machines. For one-stage axial turbines, mean-section blade diameter is about 10 percent less than the tip diameter for a radial turbine (this can be deduced from eqs. (C11) and (C14a)). For multistage machines with equal stage work, diameter is inversely proportional to the square root of the number of stages.

As a result of the parametric analysis, several favorable configurations were selected and presented as examples of geometries suitable for the systems of interest. Both single- and dual-shaft turbomachinery arrangements were studied. Two dual-shaft systems were considered in order to illustrate more clearly the effect of machine arrangement. These were a two-compressor arrangement with an approximately equal work split and a one-compressor arrangement. The turbomachinery arrangements are illustrated in figure 8.

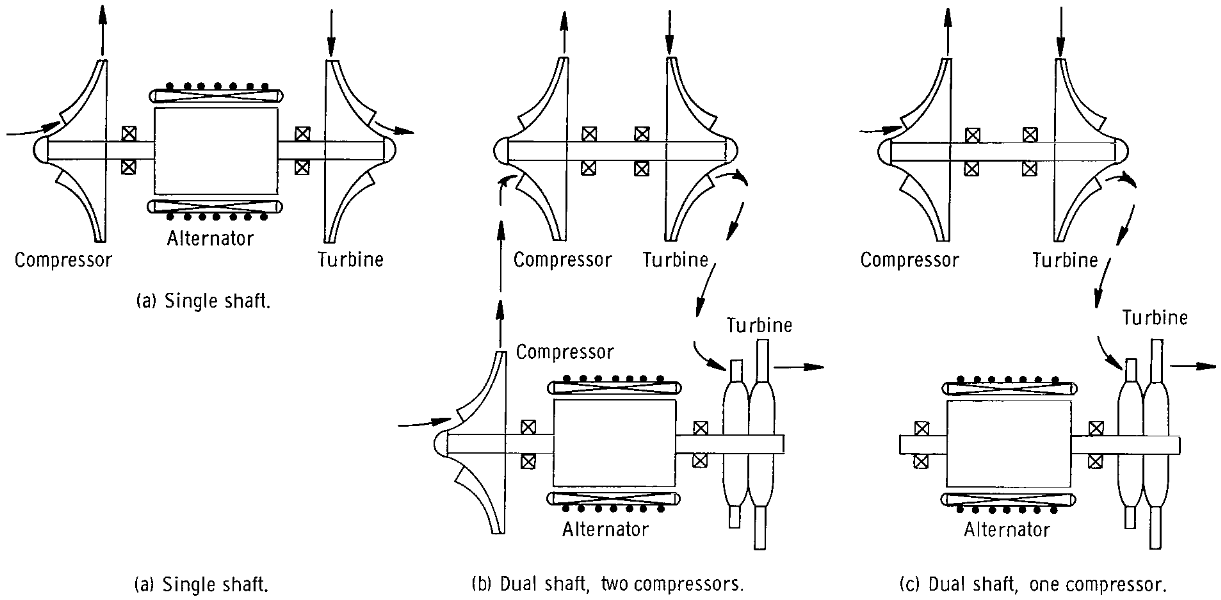
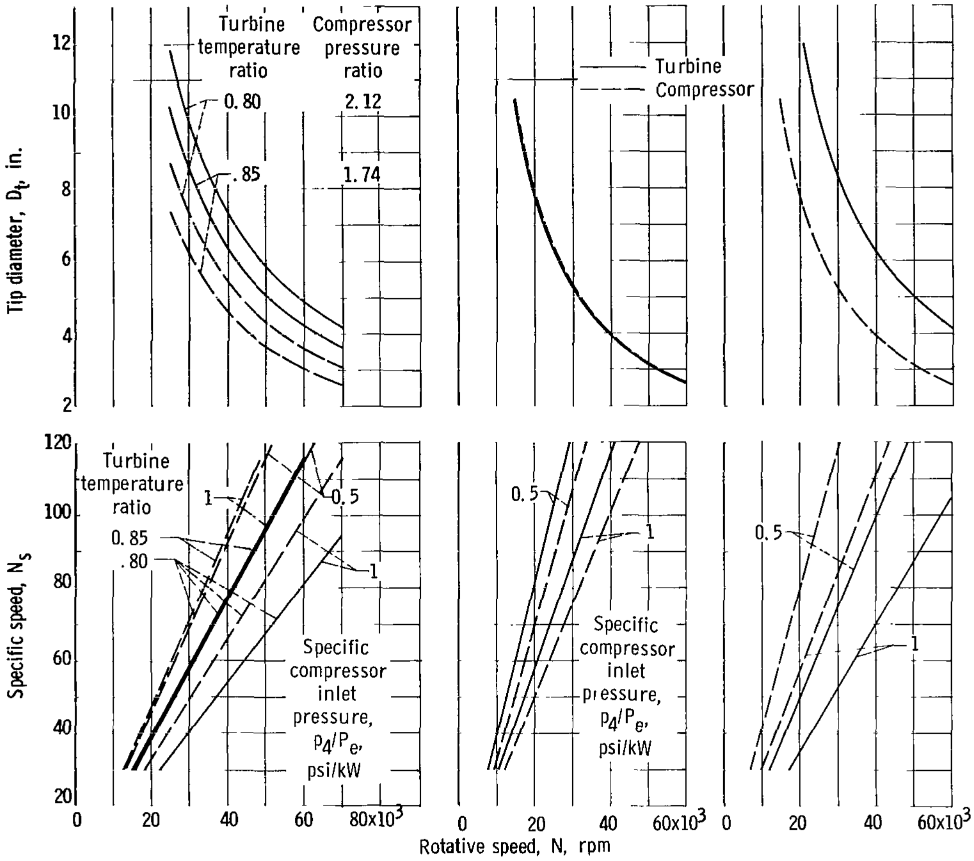


Figure 8. - Turbomachinery arrangements.

Radioisotope System

A cycle temperature ratio of 0.25 and a turbine temperature ratio of 0.80 were selected as the base operating point for examination of the radioisotope system turbomachinery. The cycle temperature ratio corresponds to a compressor inlet temperature of 550° R and places the operating point within the favorable operating region specified in figure 6. The selected turbine temperature ratio corresponds to the optimum characteristic point for the selected cycle temperature ratio and corresponds to a compressor pressure ratio (overall for the two-compressor arrangement) of 2.1. Unless otherwise specified, the fluid molecular weight is assumed to be 40. System pressure level is represented by specific compressor inlet pressure p_4/P_e, which is pressure in pounds per square inch per kilowatt of electric power. A pressure parameter such as that used allows the representation of specific speed independent of system power level (see appendix C). An alternator efficiency of 0.85 was assumed for the purpose of relating shaft power to electric power.

Presentation of basic characteristics. - In order to obtain suitable turbomachinery as specified in the section SYSTEM CHARACTERISTICS, it is necessary to match a reasonable diameter with a desirable specific speed by appropriate selection of the pertinent variables. The effects of these variables on the turbomachinery are presented in figure 9. Tip diameter and specific speed are plotted against rotative speed in figure 9(a) for the single-shaft arrangement, figure 9(b) for the dual-shaft two-compressor arrangement, and figure 9(c) for the dual-shaft one-compressor arrangement. For the dual-shaft arrangements (figs. 9(b) and (c)), the left-hand curves are for the compressor shaft and the right-hand curves are for the alternator shaft. Two specific compressor inlet pressures, 0.5 and 1.0, are used for each specific speed figure in order to show the pressure level effect and aid selection of a favorable pressure. This



(a) Single-shaft arrangement. Molecular weight, 40; ratio of work in high-pressure turbine to work in low-pressure turbine, 0.

(b-1) Compressor shaft.

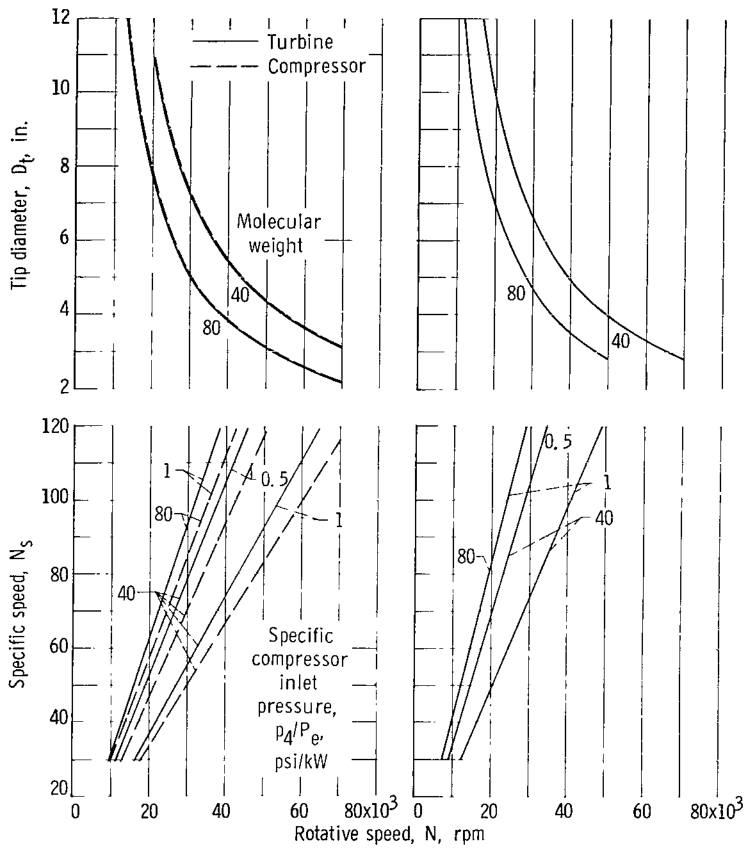
(b-2) Alternator-compressor shaft.

(b) Dual-shaft two-compressor arrangement. Turbine temperature ratio, 0.80; molecular weight, 40; ratio of work in high-pressure turbine to work in low-pressure turbine, 0.4.

Figure 9. - Turbomachinery characteristics for low-power radioisotope system. Turbine inlet temperature, 2200° R; compressor inlet temperature, 550° R.

pressure parameter has no effect on tip diameter. Also indicated on these figures are the effects of variations in turbine temperature ratio (fig. 9(a)) and molecular weight (fig. 9(c)). The basic effects shown in figure 9 are the decreasing diameter and increasing specific speed with increasing rotative speed. The general nature of the effects of rotative speed, as well as those of pressure level, turbine temperature ratio, and molecular weight, are well known. The effects of these variables are shown herein only for the purpose of illustrating their exact influence on the systems being discussed. Discussion of the nature of the effects, therefore, is concerned primarily with the manner in which they interact with other system considerations.

Pressure level. - With an operating point and fluid specified, a reasonable tip diameter, which would be about 4 to 10 inches, is obtained by selecting an appropriate rotative speed. A desirable specific speed can then be achieved through the proper choice of pressure level. A reduction in pressure level, as can be determined from figure 9, at a constant specific speed value



(c-1) Compressor shaft. (c-2) Alternator shaft.

(c) Dual-shaft one-compressor arrangement. Turbine temperature ratio, 0.80; ratio of work in high-pressure turbine to work in low-pressure turbine, 1.27.

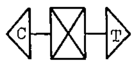
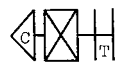
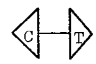
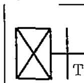
Figure 9. - Concluded.

results in a decrease in rotative speed and, consequently, an increase in diameter. The choice of pressure level, however, depends not only on the turbomachinery but also on the heat-transfer components in a Brayton-cycle system (ref. 8), and a compromise must be made between lower pressures desired for the turbomachinery and higher pressures desired for the heat exchangers. One way this compromise is effected is by selection of rotative speed to yield minimum diameter machines and thus maximize the pressure level corresponding to a desired specific speed.

Turbine temperature ratio. - The selected turbine temperature ratio affects the turbomachinery characteristics through its effect on specific work. As seen from figure 9(a), a reduction in turbine temperature ratio results in a reduction in rotative speed and an increase in pressure level while maintaining equal diameters and specific speeds. From a thermodynamics standpoint, turbine temperature ratio can be varied to some extent without severely penalizing system performance. For the typical system being considered herein, a shift in turbine temperature ratio from 0.80 to 0.85 results in a very small decrease, less than one point in 36, in cycle efficiency and about a 15 percent increase in radiator area. This shift in turbine temperature ratio corresponds

TABLE II. - TYPICAL TURBOMACHINERY CONFIGURATIONS FOR LOW-POWER RADIOISOTOPE BRAYTON-CYCLE SYSTEM

[Molecular weight, 40; specific compressor inlet pressure, 0.75 psi/kW.]

	Single shaft		Dual shaft, two compressors				Dual shaft, one compressor		
			 Compressor shaft		 Alternator-compressor shaft		 Compressor shaft		 Alternator shaft
	Compressor	Turbine	Compressor	Turbine	Compressor	Turbine	Compressor	Turbine	Turbine
Inlet temperature, T_1, $^{\circ}R$	550	2200	677	2200	550	2074	550	2200	1954
Pressure ratio, r	2.12	2.01	1.40	1.20	1.50	1.67	2.10	1.44	1.38
Type	Radial	Radial	Radial	Radial	Radial	Axial	Radial	Radial	Axial
Number of stages, n	1	1	1	1	1	2	1	1	2
Rotative speed, N, rpm	48 000	48 000	30 000	30 000	30 000	30 000	45 000	45 000	24 000
Tip diameter, D_t, in.	4.6	6.2	5.2	5.2	5.2	6.1	4.9	4.9	6.3
Specific speed, N_s	92	75	88	102	97	60	88	97	69
Last-stage hub- to tip-radius ratio, $\mathscr{R}$	---	---	---	---	---	0.74	---	---	0.68

to a reduction in compressor pressure ratio from about 2.1 to 1.7, thus offering a potential improvement in achievable compressor efficiency (ref. 10) and, consequently, in overall cycle performance.

Working fluid. - Another consideration affecting the turbomachinery characteristics is the selection of the working fluid. The effect of increasing molecular weight, as illustrated in figure 9(c), is a reduction in rotative speed and an increase in pressure level while maintaining equal diameters and specific speeds. For the pure inert gases, an increase in molecular weight results in a decrease in heat-transfer coefficient, a factor that must be considered when selecting working fluid. It was shown in reference 11, however, that binary mixtures of these inert gases can be selected such that both molecular weight and heat-transfer coefficient increase with respect to those of one of the pure gases.

Shaft arrangement. - The effect of shaft arrangement on the turbomachinery characteristics can be determined from figures 9(a), (b), and (c), which are for the single-shaft, dual-shaft two-compressor, and dual-shaft one-compressor arrangements, respectively. Comparison of a two-compressor arrangement with the one-compressor arrangements shows that lower rotative speeds for the compressor as well as higher pressure levels for the system are achievable with two compressors. The rotative speed of the alternator may be an important design consideration for a system. As seen from figure 9, a single-shaft arrangement requires a relatively high alternator speed, a dual-shaft two-compressor arrangement requires a somewhat lower alternator speed, and a dual-shaft one-compressor arrangement allows for almost complete freedom of alternator-speed selection.

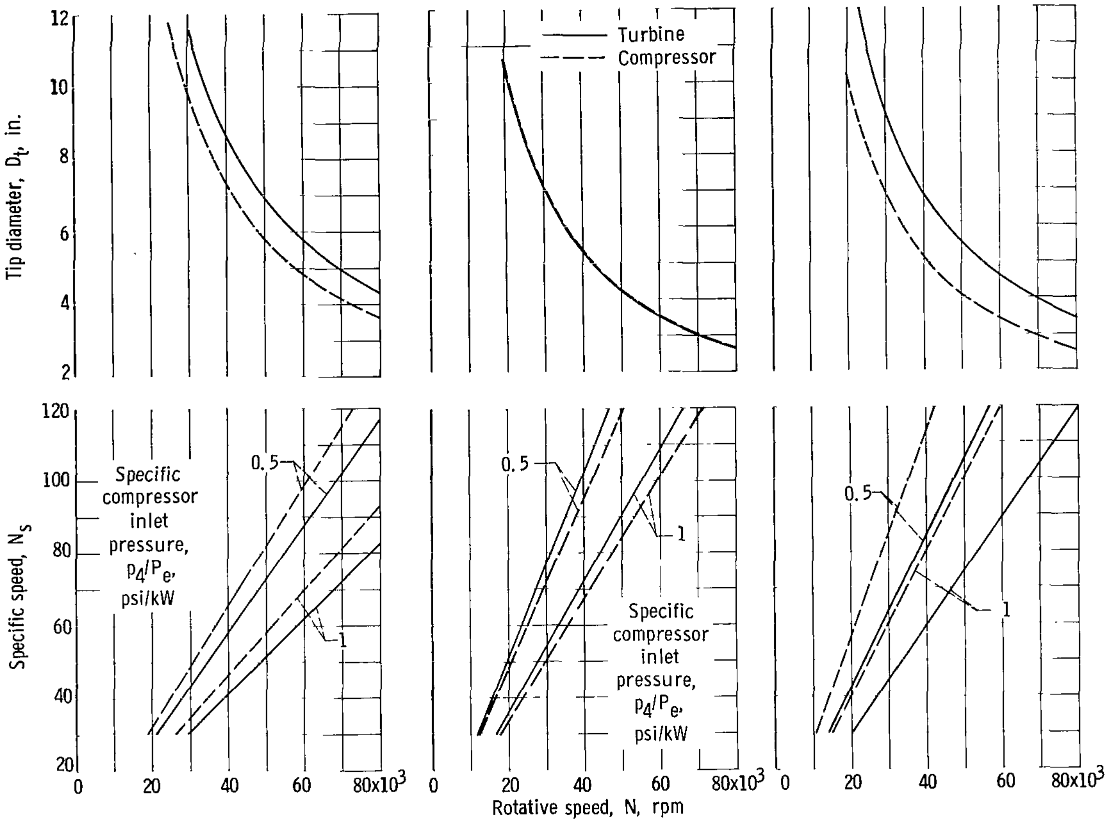
Typical configurations. - The specification of turbomachinery characteristics for any particular Brayton-cycle system depends upon the selection of such factors as rotative speed, pressure level, pressure ratio, molecular weight, and shaft arrangement so as to achieve favorable turbomachinery geometry and performance as well as favorable system performance. The effects of these factors have just been discussed. In order to illustrate typical turbomachinery configurations that might be suitable for a low-power radioisotope system, one configuration for each shaft arrangement was selected and is presented in table II. These configurations are not necessarily optimum but are merely typical, from a standpoint of size and performance, of those capable of achieving desired performance. Selection of these configurations was based upon the assumptions of a molecular weight of 40, a turbine temperature ratio of 0.80, and a specific compressor inlet pressure of 0.75.

The single-shaft arrangement features radial-flow machines rotating at 48 000 rpm. Compressor and turbine tip diameters are 4.6 and 6.2 inches, respectively, and specific speeds are 92 and 75, respectively. For the dual-shaft two-compressor arrangement, rotative speeds are 30 000 rpm for each shaft. The turbocompressor unit features one-stage radial-flow machines with compressor and turbine tip diameters of 5.2 inches, each, and specific speeds of 88 and 102, respectively. The turboalternator unit features a one-stage radial-flow compressor and a two-stage axial-flow turbine with tip diameters of 5.2 and 6.1 inches, respectively, and specific speeds of 97 and 60, respectively. For the dual-shaft one-compressor arrangement, rotative speeds are 45 000 rpm for the turbocompressor unit and 24 000 rpm for the turboalternator unit. The turbocompressor unit features one-stage radial-flow machines with compressor and turbine tip diameters of 4.9 inches, each, and specific speeds of 88 and 97, respectively. The turboalternator unit features a two-stage axial-flow turbine with a tip diameter of 6.3 inches and a specific speed of 69. These typical configurations for the radioisotope Brayton-cycle system are not significantly different from those currently under development for a solar Brayton-cycle system (refs. 9 and 10).

Reactor System

A cycle temperature ratio of 0.30 and a turbine temperature ratio of 0.75 were selected as the base operating point for examination of the reactor system turbomachinery. The cycle temperature ratio corresponds to a compressor inlet temperature of 750° R and places the operating point in the minimum-area region shown in figure 7 (p. 9). The selected turbine temperature ratio is the minimum-area value for the selected cycle temperature ratio and corresponds to a compressor pressure ratio (overall for the two-compressor arrangement) of about 2.6. Unless otherwise specified, the fluid molecular weight is assumed to be 40. As with the radioisotope system, pressure level is represented by specific compressor inlet pressure, and an alternator efficiency of 0.85 is assumed for the purpose of relating shaft power to electric power.

Presentation of basic characteristics. - There are some basic differences in turbomachinery characteristics between the radioisotope system previously discussed and the reactor system as can be seen from the parametric characteristics for the reactor system that are presented in figure 10. Tip diameter and specific speed are plotted against rotative speeds in figures 10(a) to (c) for the selected turbomachinery arrangements in a manner similar to figure 9. Two specific compressor inlet pressures are used in each figure in order to aid the selection of a favorable pressure. A molecular weight variation is shown in figure 10(c) in order to indicate the effects of using a relatively low molecular weight fluid as may be required by working fluid activation considerations. As seen in figure 10, the achievement of equal size and specific speed



(a) Single-shaft arrangement. Molecular weight, 40; ratio of work in high-pressure turbine to work in low-pressure turbine, 0.

(b-1) Compressor shaft.

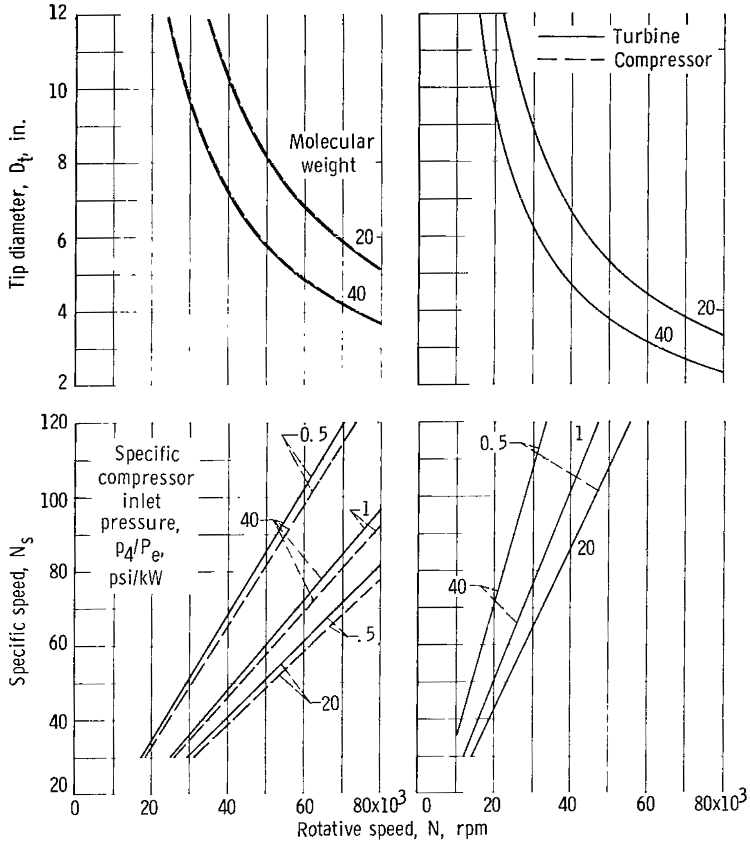
(b-2) Alternator-compressor shaft.

(b) Dual-shaft two-compressor arrangement. Molecular weight, 40; ratio of work in high-pressure turbine to work in low-pressure turbine, 0.6.

Figure 10. - Turbomachinery characteristics for intermediate-power reactor system. Turbine inlet temperature, 2500° R; compressor inlet temperature, 750° R; turbine temperature ratio, 0.75.

turbomachinery requires a higher rotative speed and a lower specific compressor inlet pressure with the reactor system as compared to the radioisotope system. These effects are caused primarily by the higher maximum temperature, the lower turbine temperature ratio, and the higher cycle temperature ratio selected for the reactor system. The general effects of rotative speed, pressure level, turbine temperature ratio, molecular weight, and shaft arrangement are the same as discussed previously for the radioisotope system and, therefore, will not be discussed here for the reactor system. This discussion of the reactor system turbomachinery will emphasize the differences in machinery characteristics and variable selection criteria as compared to the radioisotope system.

Pressure level. - The considerably higher power level of the reactor system as compared to the radioisotope system affects the criteria used for selection of rotative speed and specific compressor inlet pressure. For the radioisotope system, the very low power level makes it highly desirable to operate with high values of specific compressor inlet pressure in order to achieve a reasonable system pressure level. The higher power level of the reactor system, on the other hand, allows the use of a somewhat lower specific compressor



(c-1) Compressor shaft.

(c-2) Alternator shaft.

(c) Dual-shaft one-compressor arrangement. Ratio of work in high-pressure turbine to work in low-pressure turbine, 2.40.

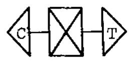
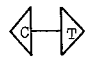
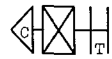
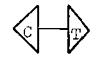
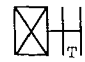
Figure 10. - Concluded.

inlet pressure while still obtaining a satisfactory pressure level. Consequently, the selection of a high rotative speed corresponding to minimum size machinery is not as strong a requirement for the reactor system as for the radioisotope system. Lower rotative speeds yielding larger-than-minimum diameter machines can be selected and specific speed requirements can be met by a reduction in specific compressor inlet pressure. This will be illustrated subsequently when some typical configurations are selected and discussed.

Working fluid. - In a system where the gaseous working fluid passes through the reactor, consideration must be given to the problem of activation of the gas (ref. 8). The nature and amount of activation depends on the selected working fluid. If the system configuration and mission are such that this activation must be reasonably minimized, the use of neon (molecular weight of 20) may be specified rather than any of the higher molecular weight fluids, which are more susceptible to activation. The significance of this requirement, as far as the turbomachinery are concerned, can be seen from figure 10(c). Suitable turbomachinery can be evolved for this lower molecular weight but as compared to the higher molecular weight of 40, the designs would

TABLE III. - TYPICAL TURBOMACHINERY CONFIGURATIONS FOR INTERMEDIATE-POWER REACTOR BRAYTON-CYCLE SYSTEM

[Molecular weight, 40.]

	Single shaft		Dual shaft, two compressors				Dual shaft, one compressor		
			 Compressor shaft		 Alternator-compressor shaft		 Compressor shaft		 Alternator shaft
	Compressor	Turbine	Compressor	Turbine	Compressor	Turbine	Compressor	Turbine	Turbine
Inlet temperature, T_1, $^{\circ}R$	750	2500	968	2500	750	2266	750	2500	2059
Pressure ratio, r	2.65	2.39	1.58	1.34	1.68	1.76	2.62	1.79	1.32
Specific compressor inlet pressure, 0.75 psi/kW									
Type	Radial	Radial	Radial	Radial	Radial	Axial	Radial	Radial	Axial
Number of stages, n	1	1	1	1	1	2	1	1	2
Rotative speed, N, rpm	64 000	64 000	44 000	44 000	36 000	36 000	64 000	64 000	24 000
Tip diameter, D_t, in.	4.5	5.4	4.9	4.9	5.7	5.7	4.6	4.6	6.1
Specific speed, N_s	86	76	85	92	84	62	85	89	71
Last-stage hub- to tip-radius ratio, $\mathscr{R}$	---	---	---	---	---	0.74	---	---	0.68
Specific compressor inlet pressure, 0.40 psi/kW									
Type	Radial	Radial	Radial	Radial	Radial	Axial	Radial	Radial	Radial or axial
Number of stages, n	1	1	1	1	1	2	1	1	1
Rotative speed, N, rpm	48 000	48 000	32 000	32 000	30 000	30 000	45 000	45 000	24 000
Tip diameter, D_t, in.	6.0	7.2	6.7	6.7	6.9	7.1	6.5	6.5	7.9
Specific speed, N_s	88	78	84	91	96	71	82	86	97
Last-stage hub- to tip-radius ratio, $\mathscr{R}$	---	---	---	---	---	0.68	---	---	0.78 (for axial)

feature either very high rotative speeds, low pressure levels, or multistaging for both the compressor and the turbine.

Typical configurations. - In order to illustrate typical turbomachinery configurations that might be suitable for an intermediate-power reactor system, two configurations for each shaft arrangement were selected and are presented in table III. One set of configurations is for a specific compressor inlet pressure of 0.75, the same value as selected for the radioisotope system. The other set of configurations is for a specific compressor inlet pressure of 0.40, a value that yields a reduction in rotative speed and an increase in machine diameter. These configurations, as were those for the radioisotope system, are not necessarily optimum but are merely typical, from a standpoint of size and performance, of those capable of achieving the desired performance. Selection of these configurations was based upon the assumptions of a molecular weight of 40 and a turbine temperature ratio of 0.75.

The characteristics for the configurations with a specific compressor inlet pressure of 0.75 are as follows. The single-shaft arrangement features single-stage radial-flow machines rotating at 64 000 rpm. Compressor and turbine tip diameters are 4.5 and 5.4 inches, respectively, and specific speeds are 86 and 76, respectively. For the dual-shaft two-compressor arrangement, rotative speeds are 44 000 rpm for the turbocompressor unit and 36 000 rpm for the turboalternator unit. The turbocompressor unit features one-stage radial-flow machines with tip diameters of 4.9 inches, each, and specific speeds of 85 and 92 for the compressor and turbine, respectively. The turboalternator unit features a one-stage radial-flow compressor and a two-stage axial-flow turbine with tip diameters of 5.7 inches, each, and specific speeds of 84 and 62, respectively. For the dual-shaft one-compressor arrangement, rotative speeds are 64 000 rpm for the turbocompressor unit and 24 000 rpm for the turboalternator unit. The turbocompressor unit features one-stage radial-flow machines with compressor and turbine tip diameters of 4.6 inches, each, and specific speeds of 85 and 89, respectively. The turboalternator unit features a two-stage axial-flow turbine with a tip diameter of 6.1 inches and a specific speed of 71.

The aforementioned configurations presented for the reactor system are quite similar in size and specific speed to those presented for the radioisotope system. For the reactor-system machines, however, the rotative speeds are significantly higher than those for the radioisotope system, and the speeds for the one-compressor arrangements are higher than those for the machines currently being developed for a solar Brayton-cycle system. As mentioned in the previous discussion, a reduction in rotative speed can be effected, if desired, by reducing pressure level. With a single-shaft arrangement, as seen in table III, a reduction in specific compressor inlet pressure from 0.75 to 0.40 results in rotative speed being reduced from 64 000 to 48 000 rpm. At this lower speed, the compressor and turbine tip diameters are 6.0 and 7.2 inches, respectively, (as compared to 4.5 and 5.4 in.) and the specific speeds are 88 and 78, respectively (as compared to 86 and 76). For the other shaft arrangements, the effects of the pressure level reduction are of a similar nature.

SUMMARY OF RESULTS

This study was made for the purpose of exploring thermodynamic and turbomachinery concepts for low-power radioisotope and intermediate-power reactor Brayton-cycle systems. The effects of intercooling and reheating were examined and turbomachinery characteristics were determined. The major results of this study are summarized as follows.

1. For low-power radioisotope systems, the achievement of high cycle efficiency is of prime importance. The general effect of intercooling is an increase in cycle efficiency accompanied by an increase in radiator area. The general effect of reheating for such systems is a smaller increase in cycle efficiency than with intercooling but accompanied by a reduction in radiator area. For the typical system examined in this study, the maximum increase in cycle efficiency achievable by the use of both intercooling and reheating was about 0.05 out of a total of about 0.35 to 0.40.

2. For intermediate-power reactor systems, the achievement of minimum radiator area is of prime importance. Intercooling offers no reduction in minimum radiator area, while reheating does offer some area reduction. The use of reheat with the typical reactor system examined in this study results in about 15 percent reduction in radiator area.

3. Comparable size and specific speed designs can be evolved for the radioiosotope- and reactor-system turbomachinery when a given ratio of pressure to power is used. Rotative speeds, however, are considerably higher (about 30 to 50 percent) for the reactor system designs. If necessary, these speeds can be lowered by appropriate reductions in pressure to power ratio.

4. Suitable turbomachinery can be evolved that will not be significantly

different from those currently under experimental investigation for solar Brayton-cycle systems. The compressors and compressor-drive turbines could be single-stage radial-flow machines with tip diameters in the 4.5 to 7.5-inch range and specific speeds of about 75 to 100. For a fluid molecular weight of 40, rotative speeds can be maintained below 50 000 rpm. The alternator-drive turbines for the dual-shaft arrangements could be two-stage axial-flow machines with tip diameters of about 6 inches and specific speeds of about 60 to 70.

Lewis Research Center,
National Aeronautics and Space Administration,
Cleveland, Ohio, April 28, 1965.

APPENDIX A

SYMBOLS

A area, sq ft

C_1 constant ratio of reheat to turbine inlet temperatures

C_4 constant ratio of intercool to compressor inlet temperatures

c_p specific heat, Btu/(lb)($^{\circ}$R)

D diameter, ft

E effectiveness

g gravitational constant, 32.2 ft/sec^2

H specific enthalpy, (ft)(lb)/lb

h radiator gas heat-transfer coefficient, Btu/(hr)(sq ft prime area)($^{\circ}$R)

J mechanical equivalent of heat, 778 (ft)(lb)/Btu

K ratio of last-stage rotor exit to inlet axial kinetic energies

M molecular weight, lb/lb mole

N rotative speed, rpm

N_s specific speed, $NQ^{1/2}/\Delta H^{3/4}$

n number of stages

P power, kW

p absolute pressure, lb/sq in. abs

Q volumetric flow rate, cu ft/sec

R universal gas constant, 1544 (ft)(lb)/(lb mole)($^{\circ}$R)

$\mathcal{R}$ last-stage hub- to tip-radius ratio

r pressure ratio

S_T ratio of work in high-pressure turbine to work in low-pressure turbine

T absolute temperature, $^{\circ}$R

U blade speed, ft/sec

V axial component of absolute velocity, ft/sec

w weight flow rate, lb/sec

x function of γ, $(\gamma - 1)/\gamma$

α mean-section stator-exit angle

γ specific heat ratio

ϵ emissivity

η efficiency

λ speed-work parameter, $U_m^2/g\ \Delta H$

ρ density, lb/cu ft

σ Stefan-Boltzmann constant, 0.173×10^{-8} Btu/(hr)(sq ft)($^{\circ}R^4$)

Subscripts:

A alternator

an last-stage annulus

C compressor

cy cycle

e electric

h high pressure

i inlet

id ideal

jk ratio of variable value at point j to variable value at point k

L loss

l low pressure

m mean section

o outlet

R radiator surface

s sink

sh shaft

T turbine

t tip

w wall

1-6 state points defined in fig. 1

APPENDIX B

CYCLE THERMODYNAMIC ANALYSIS

Cycle efficiencies were determined with respect to net shaft work as output. The basic assumptions of the analysis were that (1) the working fluid is an ideal gas; consequently, specific heat is a constant with respect to temperature, and (2) there are no thermal or mechanical losses

Cycle efficiency is defined as

$$\eta_{cy} = \frac{\text{net shaft power}}{\text{heat supplied}}$$

Symbolically, if the station nomenclature as specified by figure 1 is used,

$$\eta_{cy} = \frac{(T_{1''} - T_2) - (T_{4'} - T_4)}{(T_1 - T_6) + (T_{1''} - T_{1'})}$$

Division by T_1 yields

$$\eta_{cy} = \frac{\frac{T_{1''}}{T_1} - \frac{T_2}{T_1} - \frac{T_{4'}}{T_1} + \frac{T_4}{T_1}}{1 - \frac{T_6}{T_1} + \frac{T_{1''}}{T_1} - \frac{T_{1'}}{T_1}} \tag{B1}$$

For brevity, let T_{jk} represent T_j/T_k and p_{jk} represent p_j/p_k. With this representation, equation (B1) becomes

$$\eta_{cy} = \frac{T_{1''1} - T_{21} - T_{4'1} + T_{41}}{1 - T_{61} + T_{1''1} - T_{1'1}} \tag{B1a}$$

Specification of the cycle temperature ratio T_{41}, turbine temperature ratio T_{21}, and a turbine work split in addition to component efficiencies and pressure losses are sufficient to determine cycle efficiency as well as all cycle temperature ratios and specific capacity rate, wc_p/P_{sh}.

The following development covers all cases with and without reheating and/or intercooling. Turbine work split S_T is defined as the work in the high-pressure turbine divided by the work in the low-pressure turbine.

$$S_T = \frac{1 - T_{1'1}}{T_{1''1} - T_{21}} \tag{B2}$$

For those cases with reheating, the reheat temperature $T_{1''}$ is specified as a constant C_1 times turbine inlet temperature; therefore,

$$T_{1''1} = C_1$$

and, from equation (B2),

$$T_{1'1} = 1 - S_T(T_{1''1} - T_{21}) \tag{B2a}$$

For those cases with no reheating,

$$T_{1''1} = T_{1'1}$$

and equation (B2) yields

$$T_{1'1} = \frac{S_T T_{21} + 1}{S_T + 1} \tag{B2b}$$

Turbine isentropic efficiency is defined as

$$\eta_T = \frac{1 - T_{jk}}{1 - p_{jk}^x} \tag{B3}$$

where x represents the ratio $(\gamma - 1)/\gamma$. The turbine pressure ratios, therefore, can be expressed as

$$p_{1'1} = \left(1 - \frac{1 - T_{1'1}}{\eta_{T,h}}\right)^{1/x} \tag{B3a}$$

and

$$p_{21''} = \left(1 - \frac{1 - T_{21''}}{\eta_{T,l}}\right)^{1/x} = \left(1 - \frac{1 - \dfrac{T_{21}}{T_{1''1}}}{\eta_{T,l}}\right)^{1/x} \tag{B3b}$$

The overall turbine pressure ratio is

$$r_T = p_{1'1}p_{21''} \tag{B4}$$

and the overall compressor pressure ratio is

$$r_C = \frac{1}{r_T r_L} \tag{B5}$$

where the overall loss pressure ratio r_L represents

$$r_L = p_{1''1'}p_{32}p_{43}p_{4''4'}p_{65}p_{16}$$

Work in the high-pressure turbine must equal work in the high-pressure compressor; therefore,

$$1 - T_{1'1} = T_{51} - T_{4''1} \tag{B6}$$

For those cases with intercooling, the intercool temperature $T_{4''}$ is specified as a constant C_4 times the compressor inlet temperature; therefore,

$$T_{4''1} = C_4 T_{41} \tag{B7a}$$

For those cases with no intercooling

$$T_{4''1} = T_{4'1} \tag{B7b}$$

Compressor isentropic efficiency is defined as

$$\eta_C = \frac{p_{jk}^x - 1}{T_{jk} - 1} \tag{B8}$$

The pressure ratio for the high-pressure compressor, therefore, is

$$p_{54''} = \left[1 + \eta_{C,h}(T_{54''} - 1)\right]^{1/x} = \left[1 + \eta_{C,h}\left(\frac{T_{51}}{T_{4''1}} - 1\right)\right]^{1/x} \tag{B8a}$$

while that for the low-pressure compressor is

$$p_{4'4} = \frac{r_C}{p_{54''}} \tag{B9}$$

From equation (B8),

$$T_{4'4} = 1 + \frac{p_{4'4}^x - 1}{\eta_{C,l}} \tag{B8b}$$

and

$$T_{4'1} = T_{4'4} T_{41} \tag{B10}$$

With intercooling, the temperature ratios $T_{4''1}$, T_{51}, and $T_{4'1}$ are determined by direct substitution in equations (B6) to (B10). With no intercooling, these equations are solved simultaneously for the desired temperature ratios.

The remaining temperature ratios T_{31} and T_{61} are determined from recuperator effectiveness

$$E = \frac{T_{21} - T_{31}}{T_{21} - T_{51}} = \frac{T_{61} - T_{51}}{T_{21} - T_{51}} \tag{B11}$$

From equations (B11),

$$T_{61} = ET_{21} + (1 - E)T_{51} \tag{B11a}$$

$$T_{31} = ET_{51} + (1 - E)T_{21} \tag{B11b}$$

All cycle temperature ratios are now established, and substitution of the appropriate ratios into equation (B1a) yields cycle efficiency.

Flow rate can be expressed as

$$w = \frac{\text{net shaft power}}{\text{net shaft work per pound of fluid}}$$

or symbolically

$$w = \frac{0.9487\ P_{sh}}{c_p T_1\left[(T_{1''1} - T_{21}) - (T_{4'1} - T_{41})\right]}$$

Specific capacity rate, therefore, is

$$\frac{wc_p}{P_{sh}} = \frac{0.9487}{T_1(T_{1''1} - T_{21} - T_{4'1} + T_{41})} \tag{B12}$$

An equation derived in reference 12, for determining specific prime radiator area, is

$$\frac{A_R}{P_{sh}} = 3600\ \frac{wc_p}{P_{sh}}\left\{\frac{1}{h}\ \ln\ \frac{T_{w,i}^4 - T_s^4}{T_{w,o}^4 - T_s^4} + \frac{1}{4\sigma\epsilon T_s^3}\ \left[\ln\ \frac{(T_{w,i} - T_s)(T_{w,o} + T_s)}{(T_{w,o} - T_s)(T_{w,i} + T_s)} - 2\left(\arctan\ \frac{T_{w,i}}{T_s} - \arctan\ \frac{T_{w,o}}{T_s}\right)\right]\right\} \tag{B13}$$

where T_w is related to T by

$$T = T_w + \frac{\sigma\epsilon}{h}\ (T_w^4 - T_s^4) \tag{B14}$$

For this study, the gas cooler and intercooler are assumed to be the radiators; in equation (B13), consequently, the subscripts i and o equal 3 and 4, respectively, for the gas cooler and 4' and 4", respectively, for the intercooler.

APPENDIX C

TURBOMACHINERY CHARACTERISTICS ANALYSIS

The turbomachinery characteristics of interest for this study are specific speed and tip diameter for single-stage radial-flow turbines and compressors and multistage axial-flow turbines.

SPECIFIC SPEED

Specific speed for any machine can be defined as

$$N_s = \frac{NQ^{1/2}}{\Delta H_{id}^{3/4}} \tag{C1}$$

where the volumetric flow rate Q is that at the low-pressure end (inlet for compressor and outlet for turbine) of the machine. Volumetric flow rate is

$$Q = \frac{w}{\rho} \tag{C2}$$

Weight flow can be determined from specific capacity rate (eq. (B12))

$$w = \frac{wc_p}{P_{sh}} \frac{P_e}{\eta_A c_p} \tag{C3}$$

where $P_e = \eta_A P_{sh}$. Density is obtained from the ideal gas law

$$\rho = \frac{pM}{RT} \tag{C4}$$

From the cycle thermodynamic analysis and an assumed breakdown of the overall loss pressure ratio, any pressure used for calculating density can be expressed in terms of the compressor inlet pressure as $p_j = p_{j4}p_4$. Equation (C4), therefore, becomes

$$\rho = \frac{p_{j4}p_4 M}{RT} \tag{C4a}$$

Ideal work for a turbine is

$$\Delta H_{id,T} = \frac{\Delta H_T}{\eta_T} = \frac{Jc_p}{\eta_T} \Delta T_T \tag{C5a}$$

while that for a compressor is

$$\Delta H_{id,C} = \eta_C \, \Delta H_C = \eta_C J c_p \, \Delta T_C \tag{C5b}$$

For an ideal monatomic gas, as is being considered in this study,

$$c_p = \frac{4.97}{M} \tag{C6}$$

If the previous equations are combined, it can be seen that with working fluid and operating temperatures specified,

$$N_s = \frac{\text{constant}}{\left(\frac{p_4}{P_e}\right)^{1/2}}$$

It is the ratio of pressure to power, therefore, that determines specific speed rather than the separate effect of each.

For axial-flow turbines, last-stage hub- to tip-radius ratio $\mathcal{R}$ can be related to specific speed. Stage work and stage speed-work parameter λ are assumed equal for the turbines considered in this study; consequently the mean-section blade speed and diameter are constant for any given turbine. In addition, zero exit whirl is assumed for the fluid. Last-stage hub- to tip-radius ratio can be expressed as

$$\mathcal{R} = \frac{1 - \frac{A_{an}}{\pi D_m^2}}{1 + \frac{A_{an}}{\pi D_m^2}} \tag{C7}$$

Annulus area is

$$A_{an} = \frac{Q}{V} \tag{C8}$$

where, from geometric considerations,

$$V = \sqrt{K} \cot \alpha \, \frac{U_m}{\lambda} \tag{C9}$$

Blade mean-section diameter can be expressed

$$D_m = \frac{60 U_m}{\pi N} \tag{C10}$$

From the definition of λ

$$U_m^2 = \lambda g \frac{\Delta H}{n} = \lambda g \eta_T \frac{\Delta H_{id}}{n} \tag{C11}$$

Combining equations (C8) to (C11) and (C1) yields

$$\frac{A_{an}}{\pi D_m^2} = \frac{\pi N_s^2}{3600 \cot \alpha (K\lambda)^{1/2}} \left(\frac{n}{g\eta_T}\right)^{3/2} \tag{C12}$$

Equation (C12) in combination with equation (C7) yields last-stage hub- to tip-radius ratio in terms of specific speed. In order to attain maximum performance, λ is assumed equal to 1. The assumed typical values for K and α were 1.5 and 70°, respectively.

TIP DIAMETER

Tip diameter can be expressed

$$D_t = \frac{60U_t}{\pi N} \tag{C13}$$

The method for estimating tip speed differs for each machine. For a radial-flow turbine, theoretical (and very nearly actual) maximum performance can be achieved when

$$U_{t,T} = 0.707 \sqrt{2g\ \Delta H_{id,T}} \tag{C14a}$$

For a radial-flow compressor, a straight radial exit flow is assumed along with a slip factor of 0.85. The basic moment of momentum relation, therefore, yields

$$U_{t,C} = \sqrt{\frac{g\ \Delta H_C}{0.85}} \tag{C14b}$$

For the axial-flow turbine,

$$D_{t,T} = \frac{2D_m}{1 + \mathcal{R}} \tag{C15}$$

where $\mathcal{R}$ is obtained from equation (C7) and D_m from equations (C10) and (C11).

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