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ANALYSIS AND PRELIMINARY DESIGN STUDY OF
NUCLEAR ROCKET TURBOPUMP SYSTEMS

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Final Summary Report of Work Performed
Under Purchase Order No. J-29492

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NUCLEAR ROCKET TURBOPUMP SYSTEMS

Purchase Order No. 1214907

Final Summary Report of Work Performed
Under Purchase Order No. J-29492

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I. OBJECTIVES

A. This program was initiated to determine the problems which might reasonably be anticipated in the design of turbopumps for nuclear rocket propulsion systems. The objectives were to determine feasible propulsion system operating cycles; relate the turbine drive energy source and pump requirements thus established to the type, weight and performance of the necessary turbopump system components; consider the effects of the anticipated temperatures and radiation on the turbopump system; evaluate methods of steady state control; and describe the research and development programs which would be required for realization of such a system.

B. The program was originally outlined, as part of a more comprehensive nuclear rocket study, in Aerojet proposal PS5822, dated 9 September 1955 and subsequently authorized substantially as proposed, by University of California Radiation Laboratory Purchase Order J-29492, dated 14 March 1956. The program was active from 9 April 1956 to 15 October 1956 although work on other phases of the original study was not completed until a later date.

II. SUMMARY

A. CYCLE ANALYSIS

Initial effort under the program was directed toward evaluation of various operating cycles which were potentially desirable for nuclear rocket systems. Three basic cycles (topping, bypass, and chemical gas) were tentatively selected for further investigation and two versions of each were subsequently analyzed in detail in an attempt to determine which would be best for this application. Inasmuch as anticipated system requirements could not be predicted with any degree of certainty the system parameter ranges considered in the analyses were determined arbitrarily and thus may not be broad enough to cover all operating conditions which might subsequently be of interest. Results of these analyses were not expected to be definitive in themselves but only to indicate the kind of systems which would be required in each case and provide a basis for design studies which would ultimately permit a more comprehensive analysis. The final decision will of course involve an evaluation of the overall system hardware requirements in each case.

After evaluating each of the cycles and considering the overall aspects of their implementation it was concluded that all of them were probably feasible. In addition it was apparent that an extensive investigation of all the intimate details of each would be required to justify an unassailable conclusion as to which is best and that an equally valid conclusion could probably be based on information which would normally evolve from a study of the problems associated with systems of this type. It was recognized however that such a study would certainly be facilitated by the tentative selection of an operating cycle.

B. MECHANICAL PROBLEMS STUDY-PRELIMINARY DESIGN

This portion of the program was devoted to consideration of the problems which might be expected in the development of turbo pump systems for applications of this type. Since it had already been concluded that the nature of some of these problems would be a function of the hardware involved a preliminary design was formulated to establish hardware characteristics. The design is based on the general topping cycle which was selected arbitrarily for this purpose primarily because it probably requires the most severe turbo pump system specifications. The design finally developed consists of an integrated main turbo pump assembly composed of a single stage impulse turbine, with an inlet temperature of 1820°R, which drives pump stages with discharge pressures of 1100 and 1300 psia. This assembly is preceded by a jet booster which is driven by recirculating a portion of the flow from the main pump.

Certain of the fundamental problems which should be anticipated in this case are functions of the propellant properties. Perhaps the most significant of these results from the fact that no property data are available for portions of the anticipated operating range. However, the general characteristics of the propellant are fairly well defined. The density is unusually low, thus the attainable pressure rise per pump stage will probably be low; also, it is much more compressible than most fluids thus incompressible flow theory will not be applicable in some portions of the system and it does not have desirable lubricating properties, consequently, the development of propellant lubricated bearings might be difficult. For the most part the major problem areas indicated by the preliminary designs are associated with the large temperature differentials in the system and the severe bearing requirements.

C. WEIGHT STUDY

The desire here was to establish the validity of existing weight studies so that they could be used in the preliminary consideration of overall vehicle designs. However, it was concluded that these studies would not be applicable and an investigation was made to establish a rational basis for determination of the required information. Although the work was not completed, preliminary results indicate that the approach is fundamentally sound.

D. RADIATION HEATING STUDY

Results of this study, indicated that the maximum energy which would be generated in the turbo pump system as a result of the radiation field would be in the order of 2000 BTU/sec. This is about 4 BTU/lb of flow and is equivalent to a temperature rise of between 1 and 2°F.

E. CONTROLS

In considering the overall system it was evident that an integrated control system would ultimately be required. Consequently, the in-

F. RESEARCH AND DEVELOPMENT PROGRAMS

III. CONCLUSIONS

C. No one of the arrangements considered is inherently more conservative than any of the others. Consequently the magnitude of the development work required will be approximately the same in each case although the specific problems involved differ somewhat. The experimental work required to completely evaluate these problems, select an operating

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cycle and establish a design philosophy should be undertaken in an independent research program which could be initiated immediately. The development program, which will subsequently be required to prove designs, will differ from typical programs only in that the capacities of the components involved will be quite large and final testing in a radiation environment will be required.

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IV. DISCUSSION OF WORK

A. CYCLE ANALYSIS

1. Introduction

It was intended that the first step in the fulfillment of the objectives of this program would be an investigation of the various operating cycles which might be suitable for the applications in question in order to determine the best cycle for any particular application. Since it was evident that there are virtually an infinite number of variations of a variety of operating cycles which might be suitable, it was apparent that some judgement would have to be exercised in this area in order to establish reasonable limits for the overall problem. Therefore, efforts were limited to consideration of two variations of each of the three basic cycles described below and analyzed in Appendix A, e.g. the topping, bypass and chemical gas cycles. Results of a preliminary investigation of more complex versions of these and other cycle configurations confirmed the belief that the least complex cycles afforded the best opportunity of attaining maximum overall system performance thus justifying, at least to some extent, this arbitrary limitation.

It subsequently became evident that, as a result of the dearth of available information on reactor and nozzle requirements, the system parameter ranges which would have to be considered, in order to insure inclusion of all "reasonable" operating conditions, would be so broad that a complete analysis of the cycles selected would not be within the scope of the program. Furthermore, it was apparent that such an analysis could never be definitive because it would not account for the significant factors governed by hardware considerations. As a consequence, efforts were limited to arbitrarily selected ranges for the various significant parameters and results were evaluated in terms of estimates of system hardware requirements for the systems thus described. Inasmuch as these requirements are themselves a function not only of the cycle but also of the parameter ranges considered it is apparent that the validity of the results of these analyses depends on the applicability of the parameter ranges on which they are based.

In the case of the topping and bleed cycles it was assumed that the required turbine inlet temperature could be attained with no difficulty in a manner yet to be determined and that characteristics of the primary heat source were such that the friction losses involved would be negligible. Although this is an obvious oversimplification, in the absence of a hardware study there was no reliable information on which a better assumption could have been based.

2. Basic Operating Cycles

a. Topping Cycle

The distinguishing characteristic of the topping cycle is that all the cycle fluid (propellant) is admitted to the thrust chamber

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at maximum temperature and pressure. It also affords the possibility of designing a system with a minimum number of components which will be relatively simple and easy to control and it permits considerable flexibility in the determination of turbine operating conditions. However, the maximum pressure required is somewhat higher than that for the other cycles considered.

In the so-called general form of the topping cycle the pump system in addition to meeting the reactor requirement supplies propellant in the amount desired and at the pressure required for the pump drive turbine and cooling passages both of which are in series with the reactor. Thus, propellant at three different pressures is required in the most general case and it is obvious that if the sum of the coolant and turbine drive requirements exceeds the total propellant flow rate this variation of the topping cycle is not feasible.

A second variation of the topping cycle, the limited topping cycle, is that which results from the general case if it is assumed that there is no cooling requirement and all the propellant passes through the turbine before entering the reactor. Inasmuch as the turbine flow rate in this case is large compared to the pump power requirement a relatively low turbine inlet temperature is required for a given turbine expansion ratio.

Although this cycle is of no practical interest in the form described, because it does not provide for the system cooling requirements, it is an approximation of the case where the turbine and the cooling passages are in series with the reactor and, in the absence of detailed cooling requirement specifications, was of considerable interest for this reason. If, as now seems probable, a large portion of the total propellant flow is required for cooling and pressure losses associated with the cooling requirements are not very great this cycle will be a most attractive possibility, particularly since the temperature attained in the cooling passages is expected to be on the order of that required at the turbine.

b. Bypass (Bleed) Cycle

The bypass cycle is similar to the topping cycle except that the turbine is driven by a small portion of the cycle fluid (propellant) which is taken from some accessible point in the system, heated to the desired temperature and, after passing through the turbine, discharged from the vehicle with little or no propulsive effect. Since the pressures required at the reactor or cooling passages are more than adequate for this purpose, this in theory permits a simplification of the pump system because, in the most general case, only two propellant pressures are involved. Inasmuch as these pressures are determined by what may be considered fixed system requirements, it is impossible to devise a cycle which will operate with lower pressures.

In the idealized case there are only two conceivable variations of this cycle. In the first, the turbine inlet pressure corresponds to the reactor inlet pressure while in the second it corresponds to

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the coolant inlet pressure. Since the two pump systems are nearly identical and the turbine inlet pressures are not apt to differ by a large factor it is obvious that the resultant turbopump systems will be quite similar. Nevertheless both variations were considered so that the effect of turbine inlet pressure on the cycle could be properly evaluated.

c. Chemical Gas Cycle

The chemical gas cycle differs from the bypass cycle only in that the turbine drive gas is produced by the reaction of a portion of the propellant as a fuel with an oxidizer. It was of particular interest here because it is a cycle for which the turbine heat source requirements could be described in detail and, in the absence of similar data for the topping and bleed cycle turbine heat sources, it was the only cycle that could be described completely.

The turbine drive propellant requirement is somewhat less in this cycle than it is in the bypass cycle. However, it is expected that the oxidizer requirement will more than offset this advantage and that turbopump systems for use in this cycle would be unduly heavy, complex and difficult to control.

B. MECHANICAL PROBLEMS STUDY-PRELIMINARY DESIGN

1. Introduction

The objective of this portion of the study was to investigate and evaluate those problems which might reasonably be anticipated in the development of machines of the type under consideration. It was concluded that this could best be accomplished by selecting a typical application and specifying the problem in terms of a given preliminary design.

2. Preliminary Considerations - Pump Analysis

One of the fundamental problems in the formulation of a turbopump design is that of determining the general characteristics of the hardware which will best suit the application. The objective is essentially that of devising systems in which both the pump and turbine operate at optimum conditions. However, pump requirements and available turbine drive sources are not always consistent with this objective and some compromise in the design of components is generally necessary.

This aspect of the problem is of special significance because experience has shown that a particular turbopump configuration is generally suited only to relatively small ranges of operating conditions. Thus, the broad problem posed in the original objectives of this program would require the design of numerous turbopump systems.

In considering problems of this kind which involve a range of possible operating conditions it is desirable to investigate the geometry of that component for which requirements can be specified in greatest detail, in this case the pump, in order to establish the general characteristics of

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the systems which might be required. Results of such an investigation are shown in Figures 1 through 3. They are presented in terms of significant pump parameters which describe single suction or double suction stages that may be combined in various ways to meet a number of different pump system requirements. Limiting values consistent with good design practice were selected on the basis of previous development experience.

Significant parameters which may be combined to describe pump suction capability are shown in Figure 1. This establishes a relation between shaft speed and flow rate for various suction conditions. A shaft speed thus established may be used in Figure 2 to determine the required impeller diameter for various values of developed head from the minimum which is apt to be of interest to the maximum which might reasonably be expected considering materials limitations. Figure 3 is essentially an indication of the impeller configuration which corresponds to the various conditions previously selected. In combination with the suction conditions originally assumed it is also an indication of the pump efficiency which may reasonably be expected.

This is a major step in establishing the limits of the problem. However, the magnitude of the task is not reduced materially because there are numerous pump systems which may be designed within these limits and there are several different turbine types which may be used to extract energy from the drive gases. Furthermore, turbine requirements are to a considerable extent functions of the operating cycle involved. Consequently the possibilities are further multiplied.

3. Cycle Evaluation

The nature of the various cycles considered is such that there is no basis for a complete comparative evaluation and in the strictest sense there probably never will be. The necessity of incorporating design data in the above analyses has already been discussed; however, the broader aspects of the problem are such that development test data would also be required in order to establish the relative feasibility of various designs which might be proposed and to determine the reliability of the systems that would result. Furthermore, complete analyses should also include a more complete consideration of the operating characteristics of the various system components which might reasonably be utilized, the nature of the control problems which would be anticipated in each case and the complexity of the control hardware which would be required. It is evident then that these analyses would have to be quite comprehensive in order to permit a definitive evaluation of the cycles considered. Furthermore, it is quite probable that when the characteristics of this kind of propulsion system are more fully understood some of the tentative operating cycles can be eliminated just by exercising sound engineering judgement. In any event, it was evident that the requirements of the major components could not be specified in sufficient detail to permit a firm decision, however, certain significant aspects of the various cycles could be enumerated.

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The topping cycle is distinguished by the fact that all the cycle fluid enters into the propulsive flow. It also requires relatively high system pressures in order to provide for the drive turbine. The general configuration affords a variety of possibilities with respect to turbine and coolant flow. The limited version is relatively inflexible in this respect but it has the obvious advantage of simplicity.

The bypass cycle is characterized by the fact that it does not permit maximum utilization of propellant. However, the pressures involved are less than those in the topping cycle because they are independent of the turbine drive requirements. There is little difference between the two variations considered and a choice between them would probably be based entirely on hardware considerations having to do with accessibility of the turbine drive fluid.

The chemical cycle is essentially the same as the bypass cycle except that an oxidizer system is required. It is not apt to be particularly attractive because of its complexity and its turbine drive propellant requirement, assuming a fuel rich mixture, will not be appreciably less than that of the bypass cycle.

3. Cycle Selection

The nature of the problems associated with the definite selection of an operating cycle have already been discussed and the reasons why such a selection is not within the scope of this program have been enumerated. However, to facilitate the mechanical problems study it was desirable to formulate a preliminary design and this of course had to be based on some operating cycle. The general topping cycle was arbitrarily selected for this purpose primarily because it requires both high pressures and high temperatures and is therefore fairly representative of the various cycles under consideration. However, it also affords maximum utilization of propellant.

4. Turbopump System Specifications

Selection of an operating cycle only fixes a relation between the major system components. A flow rate and pressure schedule must also be established in order to specify pump requirements, and this information in combination with the cycle equation (Equation 4, Appendix A) fixes the relation between the significant turbine parameters. In this case the flow rate (500 lb/sec) was selected to correspond to that required for the Livermore Rover reactor and the pressure schedule was established for the same application using results of the supporting studies. Turbopump specifications thus established and estimates of related design parameters are as follows:

Pump Suction Pressure	P_s	25 psia
Reactor Inlet Pressure	P^s	800 psia
Coolant Pressure	P^c	1100 psia
Coolant Flow	P^r	
Pump Efficiency	x	.75
Turbine Inlet Pressure	η_p	.65
	P_t	1300 psia

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Turbine Inlet Temperature	T_t	1820 °R
Turbine Flow	y	.182
Turbine Efficiency	η_t	.55
Turbine Inlet Pressure Reduction Ratio	N	.91
Turbine Exhaust Pressure Reduction Ratio	M	.97

The following are average values of the fluid properties expected in this case.

Propellant Specific Heat at Constant Pressure	c_p	3.56 Btu/lb °R
Ratio of Specific Heats	k	1.4
Propellant Weight Density	ρ_w	4.2 lb/ft ³

5. Preliminary Design

Many turbopump systems could be devised to satisfy the requirements established by the turbopump specifications. If, as it now appears, circumstances are such that efficient utilization of heat is not a major consideration, selection of a particular system will depend primarily on weight considerations involving only the turbopump system and overall efficiency as such will not be a significant factor. However, the various factors which have a bearing on weight cannot be related except on an empirical basis.

In considering the basic relations governing pump design and the turbine requirements expressed by the system specifications, it was apparent that a high shaft speed is required for both the pump system, in order to minimize weight, and the turbine in order to permit the use of a single stage impulse machine, the simplest and probably the lightest type. Thus specific pumps, capable of providing the required initial pressure rise, were considered in order to determine the kind of system which would be required.

Preliminary calculations indicated that a single stage pump capable of developing the required reactor inlet pressure would have an impeller diameter of approximately 30 in. The tip speed required would be quite high by conventional standards, and the shaft speed which is limited by suction conditions, would be undesirably low. Such a pump would undoubtedly be quite heavy.

A double suction pump in this application would have an impeller diameter of approximately 21 in., however, the weight of such a pump would not be significantly different from that of the single stage machine; the permissible shaft speed would not be appreciably higher, and the tip speed would be the same.

Although the use of two pump stages to attain the desired head rise would result in lower tip speeds, the conventional two-stage pump is not particularly attractive for this application because it is

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limited by suction requirements to the same shaft speed as that of the single stage machine, while the diameter is the same as that of the double suction machine. However, it was apparent that a substantial reduction in size and weight, together with a significant increase in shaft speed could be realized by operating the two stages at different shaft speeds through a suitable speed reducer. This is an accepted method for use in applications where pump weight is a factor and high suction performance is required.

Initial design thinking was based on this type of machine and pump calculations were made using the method developed in the Pump Parameter Study, Aerojet-General Report 620, 12 February 1954 (Confidential). Subsequent turbine calculations indicated that the height of the turbine blades required was such that the blade stresses developed at the optimum pump drive speed (33,000 rpm) would be prohibitively high. Further investigation indicated that the maximum shaft speed at which the necessary nozzle area could be attained with sufficiently reduced blade height was about 24,000 rpm. This indicated that the required pressure rise might best be attained with an independent main pump stage preceded by a separate first or booster stage. In this case, the main pump, in which a major portion of the required head is developed, operates at the above shaft speed and the booster, which develops a head just sufficient to keep the main pump from being suction limited, need not necessarily be a centrifugal machine. Such an arrangement is much more flexible than the two-stage two-speed machine, and since suction requirements might be considerably more severe than the tentative specifications indicated, it is especially attractive.

System specifications require three different delivery pressures; however, the flow rate in one case is negligible, consequently the necessary pressures can be developed with the booster-driven main-stage and one auxiliary stage. The booster and main stage will raise the pressure of the total flow of propellant from the pressure at the tank to the pressure for regenerative cooling (1100 psia). Approximately 75 percent of the total flow will be admitted to the cooling passages at this pressure; 7 percent will be supplied to the reactor through a regulator at 800 psia, and the balance will be raised to the turbine inlet pressure (1300 psia) in the auxiliary pump stage. Figure 4 shows a preliminary design of an integrated main turbopump which was developed for this application. This machine, in combination with a suitable booster pump, will constitute the turbopump system.

Several different booster types might be suited to this application and each may be driven in several different ways. However, time permitted consideration of only two arrangements. The first consisted of a centrifugal booster driven by a hydraulic turbine which was in turn driven by recirculating a portion of the discharge fluid from the main pump stage. This arrangement was subsequently abandoned in favor of a second possibility which is less complex and probably lighter in weight. A jet pump (Figure 5) which is driven directly by recirculated flow from the main pump is employed in this case. The jet pump is essentially a thin walled cylinder or pipe, the upstream end of which contains one or more nozzles (in this case, seven) through which high energy drive fluid is admitted at

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high velocity. Some of this energy is transferred to the fluid being pumped with a resultant increase in head. This is a particularly attractive means of obtaining the desired boost because it does not involve any moving parts. Thus, there are no bearing or seal problems.

6. Mechanical Problem Areas

Most of the major problems that can be anticipated in applications of this kind are in some way related to the properties of the working fluid involved. Since fluid property data have not been determined for the anticipated operating ranges, available property data are substantially different in some respects from those for more familiar materials, and very little applicable experimental data are available, it is evident that a consideration of these problems must, of necessity, be somewhat speculative in nature. However, certain conclusions can be drawn from evaluation of available information. For instance, the propellant is less dense and more compressible than most liquids. Thus, it is quite probable that structural damage by cavitation in this case would not be as severe as it would in other liquids under comparable conditions, and it might be reasonable to expect that the effect of cavitation on pump characteristics would be less pronounced. On the other hand, at these low densities the head corresponding to a given pressure is unusually high. Thus, in general, more pump stages will be required here than might otherwise be expected. Furthermore, the slope of the propellant vapor curve is steeper than that of most liquids. Consequently, a higher booster pump efficiency (and thus a heavier booster) would probably be required in this case in order to attain the desired increase in net positive suction head. It is also evident that the properties of the propellant might, under certain circumstances, approach critical at some region in the pump system. Unusually low sonic velocities (probably about 2000 fps) would be expected in this event, and it is conceivable that supersonic relative velocities could be attained in some pump designs. This is not necessarily an undesirable situation provided it is recognized and designs are developed accordingly. In addition, the propellant has very poor lubricating properties. However, many improbable fluids have been used successfully as lubricants in applications of this kind.

Inasmuch as experimental work would be required to properly evaluate these more subtle aspects of the problem such an evaluation was not within the scope of the program. For this reason, and because the preliminary designs are the results of a first attempt at designing turbomachinery for this new, and in some respects, unusual application, they are reasonably conventional machines which reflect current design practice in so far as possible. The system considered is composed of a turbine wheel operating at approximately 24,000 rpm and 1800 °R and pump elements carrying fluid at temperatures below 60 °R. Space and critical speed considerations require locating the various components in close proximity to each other. The situation is further complicated by the fact that these components, if supported on ball and/or roller bearings, will probably have to be lubricated in a conventional manner since hydrogen has very poor lubricating properties. Thus, it will be necessary not only to confine

the propellant to the system, but to restrict it to certain regions of the system.

Maintenance of bearing temperature within the limits of current "acceptable design practice" will require temperature differentials of several hundred degrees between the bearing region and the fluid passage of the pump, a condition that will have to be established and maintained prior to operation. The DN (diameter x rpm) values for bearings on shafts large enough to drive systems such as these are at the upper limit of the experimental range. This indicates that the bearings might be a limiting factor in the design of this kind of equipment.

Satisfactory control of temperature conditions in the turbopump system will require a thermal barrier or heat sink between the turbine and the pumps. No difficulty is anticipated in controlling thermal radiation from the turbine and the convective effect is expected to be negligible. Heat conduction along the shaft and housing between the turbine and the pumps will, however, be a problem. If the bearing closest to the turbine is to operate at what is now considered a satisfactory temperature from a lubrication standpoint, the temperature differential along the shaft and housing between the two main support bearings will be in the order of 500 °F. This will require cooling at the turbine bearings and heating at the pump bearings. Assuming that suitable temperature conditions are maintained at the turbine bearings, the desired temperature differentials along the shaft will be of a reasonable order of magnitude and could probably be maintained with little difficulty except for the strong cooling effect of the fluid in the pump. Provisions can be made to offset this effect by providing heat sources at the pump bearings. In this respect, it is well to keep in mind that the quantity of heat which may be absorbed by the working fluid in the pumps is limited to that amount which will not have a significant effect on the rate of vaporization in the fluid passages.

Temperature considerations for portions of the systems beyond the stage closest to the turbine will be similar to those above, except that turbine temperature conditions will have practically no effect in these areas. Heat will be required to maintain satisfactory conditions at the bearings in the case of the centrifugal booster and again the situation will be critical with respect to the quantity of heat which can be permitted to pass to the fluid passages; particularly since this effect will be cumulative through the various pump stages. There will be no booster bearing or seal problems with the jet pump; however, heating in the recirculated drive fluid may be such that provision for cooling will be required.

C. WEIGHT STUDY

The objective here was to develop a simple means for determining the weights of turbopump systems which would be required for any conceivable nuclear rocket application. Inasmuch as available weight studies were based on chemical systems and their accuracy was at best questionable it was concluded that a new study would be required. It was also decided that

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this study, unlike the older ones, should account, in so far as possible, for all the factors effecting turbopump design including the possibility that several different systems can be devised for any particular application. To accomplish this it would be necessary to express results in terms of significant overall system parameters (i.e., propellant flow rate and density, pump suction and discharge pressures). This in theory would permit not only accurate weight forecasting but also pre-design optimization of turbopump systems. Such a study would be based on layouts of several different types and sizes of machines which results of complete preliminary design studies indicate might be desirable for various nuclear applications. Weights of the parts of these machines, which would be determined from the layouts, would be related to the appropriate overall system and significant component design parameters by a constant in each case. The constants for each part would be correlated over the range of system specifications considered and, if possible, the functions thus established will be collected into a single expression for each major component. This expression will fix the relation between component specifications and weight for the range of specifications and the particular design parameters considered.

An investigation was made to determine the feasibility of such an approach and basic weight equations were developed (Appendix B) in order to determine the complexity of the analysis which would be required. It was concluded that the idea was fundamentally sound. However, time did not permit completion of the required design work.

D. RADIATION HEATING

The heating study (Appendix B) is based on a turbopump configuration similar to that of the preliminary design. For purposes of this study the assembly was considered as being approximately two feet from the reactor with no intervening material, and it was assumed that the overall system had been in operation for 300 sec. Results indicate that under these conditions the total heat developed in the turbopump system as a result of radiation heating is in the order of 2000 Btu/sec. This is an indication of the conditions to be expected near the point of maximum heating. It appears at this time that any system which might be devised to meet current design criteria will not be significantly different, with respect to radiation heating, from the design considered in the study.

E. CONTROLS

Controlling the turbopump is really only part of the overall propulsion system control problem which in this case is so closely related to the characteristics of every major system component that it cannot be properly evaluated until all the preliminary designs have been completed and analyzed. However, it is expected that adequate control of steady state propellant flow can be obtained by throttling the turbine inlet with a valve which is actuated by pressure and/or temperature sensing elements at the reactor or thrust chamber. It might also be necessary to provide a bypass in the turbine feed system in order to insure that flow through the turbine heat source is always great enough to prevent overheating.

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No insurmountable problems are anticipated in securing control hardware which will be suited to the anticipated environmental conditions although it will, no doubt, be necessary to make special provisions in the design for protection of the sensing elements.

F. RESEARCH AND DEVELOPMENT PROGRAM

The feasibility of turbopump systems for application of this type is dependent on the solution of certain basic design problems and the nature of their solutions will determine to a considerable extent the kinds of systems ultimately developed. Consequently it is desirable to think in terms of an independent research program which would be concerned with the fundamental aspects of design and a separate components development program the requirements of which would be determined in part by results of the research program.

In this case the research program would involve such activities as materials testing, preliminary bearing development and determination of propellant characteristics. Of these activities materials testing is perhaps most basic to the overall problem. The objective here would be to select materials which will meet normal design requirements at cryogenic temperatures in an intense radiation field. Inasmuch as very little property data is available for materials under these conditions and numerous materials might be of interest in the design, this could be a problem which would require extensive test work. However, only a few structural materials will finally be used and these could be selected initially on the basis of existing knowledge with the reservation that they be tested properly when their desirability has otherwise been established. The few substitutions which might be required will not upset the design program and testing would thus be held to a minimum. Fluids will probably not be of interest in this case because the objective will be to develop propellant lubricated bearings.

There is little doubt that the bearing problem is one of the most significant of those expected here. Although the preliminary design incorporates rolling contact bearings which would probably require a conventional lubricant for satisfactory operation it is doubtful if any such material can withstand the radiation anticipated. Some sort of shielding might be feasible or a compromise solution involving a less desirable radiation resistant lubricant and perhaps some other bearing configuration might be acceptable. However, from an overall design standpoint, the best arrangement by far would be propellant lubricated bearings. Virtually no sealing would be required in this case and temperature differentials in the pumps would be negligible. Consequently the initial objective of this portion of the program should be to determine if such a bearing is practical.

Although a satisfactory pump for handling liquid hydrogen has been developed here at Aerojet and the feasibility of pumping this fluid under the conditions anticipated in this case is not in doubt, relatively little is known about certain of its characteristics. Of particular interest are the cavitation characteristics and minimum sonic velocity.

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Extensive testing would not be required to obtain the desired information and it could effect design thinking significantly.

The development program will be very similar to typical programs except that the power levels are quite high and as a result the components will probably have to be developed simultaneously. Perhaps the most difficult problem, aside from handling the large quantities of liquid hydrogen which will be involved, will be the procurement of an adequate heat source for the test unit. Final testing in a radiation environment will of course be required and test procedures will thus be further complicated.

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LOGARITHMIC KEUFFEL & ESSER CO. MADE IN U.S.A.
2 X 1 CYCLES

EP/TAC

225:56-75

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Figure 1

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Pump Impeller Diameter as a Function of Shaft Speeds for
Several Values of Head Rise, Head Coefficient, $\psi = .5 = \text{Const.}$

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Figure 2

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K&E LOGARITHMIC 359-103
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2 X 1 CYCLES

Specific Speed - Shaft Speed Ratio, N_s/N

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Figure 3

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Figure 4

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Figure 5

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APPENDIX A

CYCLE ANALYSIS

I. INTRODUCTION

A. This is a general analysis of those basic cycles which appear to be best suited to the nuclear rocket applications under consideration. The objective was to determine the best cycle for a particular application and also to establish component specifications for that particular cycle.

B. For these purposes a constant density, non-viscous incompressible fluid was assumed in order to establish pump power requirements. It was also assumed that the propellant is thermally and calorically perfect in the turbine and that pressure losses in the connecting hardware would be negligible. In order to facilitate the analysis, the efficiencies of the components of a particular pump system were assumed equal. This is equivalent to establishing a system efficiency which is a function of the efficiencies of the components and the work done by each component. The analysis is based on ranges of parameter values which, in the absence of detailed system component characteristics were determined arbitrarily and it does not involve a consideration of the physical aspects of the hardware which might be required.

C. The parameters involved here are so numerous as to preclude a single simple presentation for each cycle which would show all the possible combinations. Consequently they were treated in pairs and results are presented in terms of the limiting combinations of those not considered variable in each case.

II. ANALYSIS

A. Work was limited to consideration of the simplest forms of two variations of each of three basic operating cycles. They are the topping cycle, the bypass or bleed cycle and the chemical gas cycle. For these purposes the topping cycle may be described as one in which all the propellant passes through the reactor after the turbine drive and system cooling requirements have been fulfilled. Pressures are, in general, somewhat higher here than for the other cycles, however, this cycle affords maximum utilization of propellant. The bypass cycle is one in which the pump drive turbine is driven by a small amount of heated propellant from which virtually no useful thrust is obtained. Pressure requirements in this case are at a minimum established by reactor and nozzle requirements. The chemical gas cycle is similar to the bypass cycle except that turbine drive gas is obtained from the reaction of a portion of the propellant with an oxidizer, which is supplied from a pressurized system.

B. Although the nature of the turbine heat sources required for the topping and bleed cycles could not be established, an analysis was made to determine the pressure losses which might reasonably be associated with the required heat addition in each case. This analysis assumes one dimensional flow in a constant area channel in which friction effects

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Appendix A, continued

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are negligible. Information and techniques presented in "Dailey, C. L. and Wood, F. C. - Computation Curves for Compressible Fluid Problems - John Wiley and Sons, Inc. 1949" were used in the calculations. Results are shown in Figures 1 through 5. These results were used to determine reasonable values for the ratio (N) of the turbine inlet pressure (P_t) to the appropriate nominal turbine inlet pressure in each case. They were also used to establish values for the ratio (M) of the general topping cycle turbine exhaust pressure (P'_c) to reactor inlet pressure (P_c). This ratio accounts for the fact that it might be necessary or desirable to admit the spent turbine drive gas to the reactor at some pressure lower than the reactor inlet in order to match the turbine exhaust gas temperature with the reactor temperature.

C. Equations describing the various cycles, as they might be applied to regeneratively cooled systems, may be developed as follows:

1. Topping Cycle - General Form

a. For this cycle (Figure 6) the pump system must raise the pressure of the total flow of propellant ($\dot{w}$) from the suction pressure (P_s) to the nominal reactor inlet pressure (P_c). It must also raise the pressure of the regenerative cooling flow ($x\dot{w}$) from P_s to the regenerative cooling pressure (P_r) and the pressure of the turbine flow ($y\dot{w}$) from P_c to the nominal turbine inlet pressure (P_t). For these conditions the pump drive requirement may be expressed as follows:

$$\text{Power} = 144 \left[\frac{\dot{w} (P_c - P_s)}{\rho_w \eta_p} + \frac{x\dot{w} (P_r - P_c)}{\rho_w \eta_p} + \frac{y\dot{w} (P_t - P_c)}{\rho_w \eta_p} \right] \frac{\text{ft lb}}{\text{sec}} \quad (1)$$

where ρ_w = propellant density lb/ft³

η_p = pump efficiency

$$(X+Y) \leq 1$$

b. The output of a turbine operating under the condition just described may be expressed as follows:

$$\text{Power} = 778 y \dot{w} c_p T_t \eta_t \left[1 - \left(P_c / P_t \right)^{\frac{k-1}{k}} \right] \frac{\text{ft lb}}{\text{sec}} \quad (2)$$

where c_p = propellant specific heat at constant pressure BTU/lb^oR

T_t = Turbine inlet temperature ^oR

η_t = Turbine efficiency

k = Ratio of specific heats

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Appendix A, continued

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c. Since for satisfactory operation the power requirement established by Equation (1) must be identical to the available power expressed by Equation (2), these expressions may be combined to establish the basic cycle equation shown below.

$$y = \frac{P_c - P_s + x (P_r - P_c)}{\frac{778}{144} c_p T_t \eta_t \rho_w \eta_p \left[1 - \left(P_c / P_t \right)^{\frac{k-1}{k}} \right] - (P_t - P_c)} \quad (3)$$

d. By incorporating the results of the flow studies and combining the parameters which are either functions of turbopump system hardware or essentially constant, Equation (3) may be written

$$y = \frac{P_c - P_s + x (P_r - P_c)}{A T_t \left[1 - \left(P'_c / P'_t \right)^{\frac{k-1}{k}} \right] - (P_t - P_c)} \quad (4)$$

where $A = (778/144) c_p \rho_w \eta_t \eta_p$

$$P'_c = M P_c$$

see II B

$$P'_t = N P_t$$

e. Parameter ranges for which this expression was evaluated are shown in Figure 6. Results of the evaluation are shown in Figures 7 through 11.

2. Limited Topping Cycle

a. In this case (Figure 12) it was assumed that the cooling requirement would be negligible and that all the propellant would be used in the turbine. Therefore the pump must raise the total flow of propellant ($\dot{w}$) from the suction pressure (P_s) to the nominal turbine inlet pressure (P_t). The turbine will operate with an inlet temperature (T_t) and a nominal exhaust pressure corresponding to the reactor inlet pressure (P_c)

b. For these conditions the appropriate equations may be written as follows:

$$\text{Pump Power} = 144 \dot{w} \frac{(P_t - P_s)}{\rho_w \eta_p} \frac{\text{ft lb}}{\text{sec}} \quad (5)$$

$$\text{Turbine Power} = 778 \dot{w} c_p T_t \left[1 - \left(P_c / P_t \right)^{\frac{k-1}{k}} \right] \frac{\text{ft lb}}{\text{sec}} \quad (6)$$

The basic cycle equation shown below may then be determined by combining Equations (5) and (6) as indicated in l.c. and

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Appendix A, continued

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modifying and simplifying as explained in l.d.

$$T_t = \frac{P_t - P_s}{A \left[1 - \left(P_c / P_t \right)^{\frac{k-1}{k}} \right]} \quad (7)$$

c. This expression was evaluated for the parameter ranges indicated in Figure 12. Results are shown in Figures 13 and 14.

3. Bypass (Bleed) Cycles - First Configuration

a. In this cycle (Figure 15) the propellant requirement may best be expressed as $(1+y)\dot{w}$ where $(\dot{w})$ is the propulsive flow requirement and $(y\dot{w})$ is the turbine drive requirement. The pump then is required to raise the pressure of the total propellant flow $(1+y)\dot{w}$ from the suction pressure (P_s) to the nominal thrust chamber pressure (P_c) . It must also raise the pressure of the coolant flow $(x\dot{w})$ from (P_c) to the cooling passage inlet pressure (P_r) . The turbine will operate with an inlet temperature (T_t) , and inlet pressure (P_t) and an exhaust pressure (P_{ex}) .

b. The power balance is fixed by the following equations:

$$\text{Pump Power} = 1114 \left[\frac{\dot{w} (1+y) (P_c - P_s)}{\rho_w \eta_p} + \frac{x \dot{w} (P_r - P_c)}{\rho_w \eta_p} \right] \frac{\text{ft lb}}{\text{sec}} \quad (8)$$

$$\text{Turbine Power} = 778 y \dot{w} c_p T_t \left[1 - \left(P_{ex} / P_t \right)^{\frac{k-1}{k}} \right] \frac{\text{ft lb}}{\text{sec}} \quad (9)$$

The cycle equation is

$$y = \frac{P_c - P_s + x (P_r - P_c)}{A T_t \left[1 - \left(P_{ex} / P_t \right)^{\frac{k-1}{k}} \right] - (P_c - P_s)} \quad (10)$$

Note that $P_t = N P_c$ where N is a function of the pressure loss which will result from heating the turbine drive fluid.

c. This cycle is evaluated in Figures 16 through 20.

4. Bypass (Bleed) Cycle - Second Configuration

a. This arrangement (Figure 21) is very similar to that just considered. In this case however the turbine drive gas is at the higher pressure corresponding to the regenerative cooling pressure. Therefore the pump requirement is the same except that the pump system

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Appendix A, continued

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must also raise the pressure of the turbine flow ($y\dot{w}$) from the reactor inlet pressure (P_C) to (P_R) the pressure for regenerative cooling. The turbine may be described in terms of the same parameters except that in this case the turbine inlet pressure (P_t) is related to the regenerative cooling pressure (P_R).

b. The cycle equation may be developed as follows in the manner used previously.

$$\text{Pump Power} = 144 \left[\frac{(y+1) \dot{w} (P_C - P_S)}{\rho_w \eta_p} + \frac{(x+y) \dot{w} (P_R - P_C)}{\rho_w \eta_p} \right] \frac{\text{ft lb}}{\text{sec}} \quad (11)$$

$$\text{Turbine Power} = 778 y \dot{w} c_p \eta_t T_t \left[1 - \left(P_{\text{ex}}/P_t \right)^{\frac{k-1}{k}} \right] \frac{\text{ft lb}}{\text{sec}} \quad (12)$$

$$y = \frac{P_C - P_S + x(P_R - P_C)}{A T_t \left[1 - \left(P_{\text{ex}}/P_t \right)^{\frac{k-1}{k}} \right] - (P_R - P_S)} \quad (13)$$

c. This cycle is evaluated in Figures 22 through 26.

5. Chemical Cycle

a. The variations of the chemical cycle which are considered here are very similar to the respective bypass cycle variations described above and, insofar as the parametric presentation is concerned, the pump requirements may be considered identical. The expression for turbine power could also be of the same form as that used previously. However, it is convenient to describe the turbine requirements in terms of turbine nozzle spouting velocity (c_o) and turbine drive fluid mixture ratio (iR) as follows:

$$\text{Turbine Power} = \eta_t y \dot{w} (1 + \text{iR}) \frac{c_o^2}{2g} \frac{\text{ft lb}}{\text{sec}} \quad (14)$$

$$\text{where } c_o = c^* C_f$$

The characteristic velocity (c^*) and the mixture ratio (iR) are both functions of turbine inlet temperature (T_C), (Figure 27), while the thrust coefficient (C_f) is a function of the turbine expansion ratio and the turbine drive gas specific heat ratio, (Figure 28).

b. The power balance for the first configuration of the chemical cycle (Figure 29) may be established by combining Equations (8) and (14). The cycle equation thus established may be written as follows:

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Appendix A, continued

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$$y = \frac{P_c - P_s + x (P_r - P_c)}{B (1+MR) (\sigma^* C_f)^2 - (P_c - P_s)} \quad (15)$$

where $B = \gamma_t \gamma_p \rho_w / 288g$

This expression is evaluated in Figures 30 through 34

c. In a similar manner Equations (11) and (14) may be used to determine the following relation which describes the second variation of the chemical cycle (Figure 35) which is evaluated in Figures 36 through 40.

$$y = \frac{P_c - P_s + x (P_r - P_c)}{B (1+MR) (\sigma^* C_f)^2 - (P_r - P_s)} \quad (16)$$

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FLOW SCHEMATIC
FOR CALCULATION OF PRESSURE LOSS DUE TO ADDITION OF HEAT

Where stagnation pressure ratios P_{s2}/P_{s1} are determined for
Mach Numbers

$$M_1 = .04 \text{ to } .10$$

$$M_2 = .3, .4, .5$$

Stagnation Temperature Ratios

$$T_{s2}/T_{s1} = 15, 20, 30, 40, 50, 60, 70$$

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CONTRACT NO.		USED ON		MACH NUMBER & STAGNATION PRESSURE RATIO VS. STAGNATION TITLE TEMPERATURE RATIO, $M_2 = 0.5$	
REFERENCE TEST NO'S.					
REFERENCE FIG. NO'S.				COMPONENT:	
PREPARED BY: D.F. VANICA	DATE 6-22-56	PART NO.	SERIAL NO.		
APPROVED BY:	DATE	COMPONENT MFG.	PAGE NO.		
TESTING AGENCY	REPORT NO.	AEROJET-GENERAL CORPORATION		FIG. NO. 2	

DECLASSIFIED

DECLASSIFIED

CONTRACT NO.		USED ON		MACH NUMBER & STAGNATION PRESSURE RATIO VS. STAGNATION TITLE TEMPERATURE RATIO, $M_2 = 0.4$	
REFERENCE TEST NO'S.					
REFERENCE FIG. NO'S.				COMPONENT:	
PREPARED BY: <i>D. F. VANICA</i>	DATE	PART NO.	SERIAL NO.		
APPROVED BY:	DATE	COMPONENT MFG.	PAGE NO.		
TESTING AGENCY	REPORT NO.	AEROJET-GENERAL CORPORATION		FIG. NO. <i>3</i>	

DECLASSIFIED

MACH NUMBER, M_1

CONTRACT NO.		USED ON		<p>MACH NUMBER & STAGNATION PRESSURE RATIO VS. STAGNATION TITLE TEMPERATURE RATIO, $M_1 = 0.3$</p>	
REFERENCE TEST NO'S.					
REFERENCE FIG. NO'S.					
PREPARED BY: <i>D.F. VANICA</i>		DATE <i>6/23/56</i>		COMPONENT:	
APPROVED BY:		DATE		PART NO.	
TESTING AGENCY		REPORT NO.		SERIAL NO.	
				COMPONENT MFG.	
				AEROJET-GENERAL CORPORATION	
				PAGE NO.	
				FIG. NO. <i>4</i>	

DECLASSIFIED

CONTRACT NO.		USED ON		PRESSURE LOSS vs TEMPERATURE RATIO VARIATION FOR VARIABLE INLET TITLE VELOCITY & CONSTANT EXIT VELOCITY	
REFERENCE TEST NO'S.					
REFERENCE FIG. NO'S.				COMPONENT:	
PREPARED BY: D. F. VANICA		DATE 6-23-56		PART NO.	SERIAL NO.
APPROVED BY:		DATE		COMPONENT MFR.	PAGE NO.
TESTING AGENCY		REPORT NO.		AEROJET-GENERAL CORPORATION	
				FIG. NO. 5	

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GENERAL TOPPING CYCLE

DIAGRAM

EQUATION

$$y = \frac{P_c - P_s + x (P_r - P_c)}{A T_t \left[1 - (P_c' / P_t')^{\frac{k-1}{k}} \right] - (P_t - P_c)}$$

when $x + y \leq 1$

LIST OF SYMBOLS

$$A = (778.16/144) c_p \rho_w \eta_t \eta_p$$

$$P_c' = M P_c$$

$$P_t' = N P_t$$

$$P_s = 30 \text{ psia}$$

$$k = 1.4$$

PARAMETER RANGES

$$c_p = 3.42 - 3.70 \frac{\text{BTU}}{\text{lb}^\circ\text{R}}$$

$$\rho_w = 4.0 - 4.4 \text{ lb/ft}^3$$

$$\eta_t = 0.4 - 0.7$$

$$\eta_p = 0.5 - 0.8$$

$$P_c = 400 - 1000 \text{ psia}$$

$$P_t - P_c = 200 - 600 \text{ psi}$$

$$T_t = 1460 - 2060^\circ\text{R}$$

$$x = 0 - .8$$

$$\lambda = 14.8 - 49.2$$

$$M = .95 - 1.0$$

$$N = .87 - .95$$

$$P_r = P_c + 200 \text{ psi}$$

Fig. 6 - 1

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TURBINE FLOW vs. COOLANT FLOW

	MIN Y	MAX Y
P_c	1600 PSIA	600 PSIA
T_c	2060°R	1460°R
M	.95	1.0
N	.95	.87
A	49.2	14.8

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TURBINE FLOW vs. INLET PRESSURE

	MIN Y	MAX Y
T_F	2060°R	1460°R
X	0	.8
M	.95	1.0
N	.95	.87
A	49.2	14.8

TURBINE FLOW VS. PRESSURE RATIO

	MIN Y	MAX Y
T_F	2060 °R	1460 °R
X	0	.8
M	.95	1.0
N	.95	.87
A	49.2	14.8

Fig 9

TURBINE FLOW vs. INLET TEMPERATURE

	MIN Y	MAX Y
P_T	1600 PSIA	600 PSIA
X	0	.8
M	.95	1.0
N	.95	.87
A	49.2	14.8

FIG 10

TURBINE FLOW VS. LUMPED PARAMETER

	MIN Y	MAX Y
P_t	1600 PSIA	600 PSIA
T_t	2060 °R	1460 °R
X	0	.8
M	.95	1.0
N	.95	.87

NOTE: $A = \frac{178.16}{144} C_p P_{0t} \eta_T \eta_P$

LIMITED TOPPING CYCLE

DIAGRAM

EQUATION

$$T_t = \frac{P_t - P_s}{A \left[1 - (P_c/P_t')^{\frac{k-1}{k}} \right]}$$

SYMBOLS AND PARAMETER RANGES

$$A = (778.16/144) c_p \rho_w \eta_t \eta_p$$

$$P_t' = N P_t$$

$$c_p = 3.42 - 3.70 \text{ BTU/lb}^\circ\text{R}$$

$$\rho_w = 4.0 - 4.4 \text{ lb/ft}^3$$

$$\eta_t = 0.4 - 0.7$$

$$\eta_p = 0.5 - 0.8$$

$$P_c = 400 - 1000 \text{ psia}$$

$$P_s = 30 \text{ psia}$$

$$k = 1.4$$

$$P_t - P_c = 200 - 600 \text{ psi}$$

$$T_t = 1460 - 2060 \text{ }^\circ\text{R}$$

$$x = 0 - .8$$

$$A = 14.8 - 49.2$$

$$N = .95 - 1.0$$

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TURBINE INLET TEMPERATURE
vs.
INLET PRESSURE

	MIN T	MAX T
A	99.2	14.8
N	1.0	.95

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TURBINE INLET TEMPERATURE VS. PRESSURE RATIO

	MIN T	MAX T
A	492	14.8
N	1.0	.95

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BYPASS CYCLE First Configuration

DIAGRAM

EQUATION

$$y = \frac{P_c - P_s + x(P_r - P_c)}{A T_t \left[1 - (P_{ex}/P_t)^{\frac{k-1}{k}} \right] - (P_c - P_s)}$$

SYMBOLS AND PARAMETER RANGES

$$A = (778.16/144) c_p \rho_w \eta_t \eta_p$$

$$c_p = 3.42 - 4.4 \text{ BTU/lb}^\circ\text{R}$$

$$\rho_w = 4.0 - 4.4 \text{ lb/ft}^3$$

$$\eta_t = 0.4 - 0.7$$

$$\eta_p = 0.5 - 0.8$$

$$P_s = 30 \text{ psia}$$

$$P_{ex} = 20 \text{ psia}$$

$$k = 1.4$$

$$P_t = N P_c$$

$$N = .87 - .95$$

$$P_c = 400 - 1000 \text{ psia}$$

$$P_t - P_c = 200 - 600 \text{ psi}$$

$$T_t = 1460 - 2060 \text{ }^\circ\text{R}$$

$$x = 0 - .8$$

$$A = 14.8 - 49.2$$

$$P_r = P_c + 200 \text{ psi}$$

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TURBINE FLOW vs. COOLANT FLOW

	MIN Y	MAX Y
T_f	2060°R	1460°R
N	.95	.87
A	49.2	14.8

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FIG. 16

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TURBINE FLOW VS. INLET PRESSURE

	MIN Y	MAX Y
T_f	2060 °R	1460 °R
X	0	.8
N	.95	.87
A	49.2	14.8

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Fig. 17

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TURBINE FLOW
VS.
PRESSURE RATIO

	MIN Y	MAX Y
T_f	2060°R	1460°R
X	0	.8
N	.95	.87
A	49.2	14.8

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FIG. 1B

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TURBINE FLOW VS. INLET TEMPERATURE

	MIN Y	MAX Y
X	0	.8
N	.95	.87
A	49.2	14.8

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TURBINE FLOW vs. COMBINED PARAMETER

	MIN Y	MAX Y
T_f	2060°R	1460°R
X	0	.8
N	.95	.87

NOTE: $\frac{798.16}{144} C_p P_w \eta_t \eta_p = A$

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BYPASS CYCLE
Second Configuration

DIAGRAM

EQUATION

$$y = \frac{P_c - P_s + x(P_r - P_c)}{A T_t \left[1 - (P_{ex}/P_t)^{\frac{k-1}{k}} \right] - (P_r - P_s)}$$

SYMBOLS AND PARAMETER RANGES

$$A = (778.16/144) c_p \rho_w \eta_t \eta_p$$

$$c_p = 3.42 - 4.4 \text{ BTU/lb } ^\circ\text{R}$$

$$\rho_w = 4.0 - 4.4 \text{ lb/ft}^3$$

$$\eta_t = 0.4 - 0.7$$

$$\eta_p = 0.5 - 0.8$$

$$P_s = 30 \text{ psia}$$

$$P_{ex} = 20 \text{ psia}$$

$$k = 1.4$$

$$P_t = N P_r$$

$$N = .87 - .95$$

$$P_c = 400 - 1000 \text{ psia}$$

$$T_t = 1460 - 2060 \text{ } ^\circ\text{R}$$

$$x = 0 - .8$$

$$A = 14.8 - 49.2$$

$$P_r = P_c + 200 \text{ psi}$$

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FIG. 21

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TURBINE FLOW VS. COOLANT FLOW

	MIN Y	MAX Y
T_t	2060°R	1460°R
N	.95	.87
A	49.2	14.8

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FIG. 22

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TURBINE FLOW
VS.
INLET PRESSURE

	MIN Y	MAX Y
T_F	2060 °R	1460 °R
X	0	.8
N	.95	.87
A	49.2	14.8

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FIG. 23

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TURBINE FLOW VS. PRESSURE RATIO

	MIN Y	MAX Y
T_T	2060 °R	1960 °R
X	0	.8
N	.95	.87
A	49.2	14.8

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.09
.08
.07
.06
.05
Y
.04
.03
.02
.01
0

.016 .020 .024 P_{ex}/P_T .028 .032 .036 .040

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TURBINE FLOW VS. INLET TEMPERATURE

	MIN Y	MAX Y
X	0	.8
N	.95	.87
A	49.2	14.8

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Y

.09
.08
.07
.06
.05
.04
.03
.02
.01
0

$P_c = 1000$ PSIA
 $P_c = 800$
 $P_c = 600$
 $P_c = 400$

$P_c = 1000$ PSIA
 $P_c = 800$
 $P_c = 600$
 $P_c = 400$

1400 1500 1600 1700 1800 1900 2000 2100

T_r °R

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FIG. 25

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TURBINE FLOW VS. COMBINED PARAMETER

	MIN Y	MAX Y
T_f	2060°R	1460°R
X	0	.8
N	.95	.87

NOTE: $A = \frac{778.16}{144} C_p P_w \gamma + \gamma P$

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FIG. 26

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CHARACTERISTIC VELOCITY & MIXTURE RATIO VS. CHAMBER TEMPERATURE

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MAXIMUM THRUST COEFFICIENT AND CORRESPONDING AREA RATIO
AS FUNCTIONS OF
PRESSURE RATIO AND SPECIFIC HEAT

A_e = nozzle exit area
 A_t = nozzle throat area
 C_f = nozzle thrust coefficient
 k = ratio of specific heats
 M = Mach number
 P_a = ambient pressure
 P_c = chamber pressure
 P_e = nozzle exit pressure

Thrust Coefficient, C_f

Area Ratio, A_e/A_t

Pressure Ratio, $P_c/P_e = P_c/P_a$

1.50 1.10

1.50

1.10

$M < 1$

$M > 1$

k

1.10
1.15
1.20
1.25
1.30
1.35
1.40
1.45
1.50

k

1.10
1.15
1.20
1.25
1.30
1.40
1.50
1.67

28
26
24
22
20
18
16
14
12
10
8
6
4
2
0

2.0
1.8
1.6
1.4
1.2
1.0
0.8
0.6
0.4
0.2

1
2
5
10
20
50
100
200
500
1000

30
28
26
24
22
20
18
16
14
12
10
8
6
4
2
0

30
28
26
24
22
20
18
16
14
12
10
8
6
4
2
0

30
28
26
24
22
20
18
16
14
12
10
8
6
4
2
0

30
28
26
24
22
20
18
16
14
12
10
8
6
4
2
0

30
28
26
24
22
20
18
16
14
12
10
8
6
4
2
0

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CHEMICAL CYCLE First Configuration

DIAGRAM

EQUATION

$$y = \frac{P_c - P_s + x (P_r - P_c)}{B (1 + r) (C_f C^*)^2 - (P_c - P_s)}$$

SYMBOLS AND PARAMETER RANGES

$$B = \frac{\gamma_p \gamma_t \rho_w}{288 g}$$

$$\rho_w = 4.0 - 4.4 \text{ lb/ft}^3$$

$$\gamma_t = 0.4 - 0.7$$

$$\gamma_p = 0.5 - 0.8$$

$$P_s = 30 \text{ psia}$$

$$k = 1.4$$

$$P_c = 400 - 1000 \text{ psia}$$

$$x = 0 - 0.8$$

$$B = 0.08 \times 10^{-3} - .266 \times 10^{-3}$$

$$P_r = P_c + 200 \text{ psi}$$

$$MR = .72 - 1.10$$

$$C^* = 6000 - 7340 \text{ ft/sec.}$$

$$C_f = 1.26 - 1.46$$

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TURBINE FLOW VS. COOLANT FLOW

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FIG. 30

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TURBINE FLOW
VS.
INLET PRESSURE

	MIN Y	MAX Y
T_F	2060°R	1460°R
X	0	.8
B	.000220	.000310

.11

.10

.09

.08

.07

.06

.05

.04

Y

.03

.02

.01

0

0

100

200

300

400

500

600

700

800

P_F PSIA

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FIG. 31

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TURBINE FLOW
VS.
PRESSURE RATIO

	MIN Y	MAX Y
T _T	2060°R	1460°R
X	0	8
B	.000026	.000086

DECLASSIFIED FIG. 32

DECLASSIFIED

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TURBINE FLOW VS. INLET TEMPERATURE

	MIN Y	MAX Y
X	0	.8
B	.000226	.000086

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TURBINE FLOW VS. COMBINED PARAMETER

MIN Y MAX Y
T_T 2060°R 1460°R
X 0 .8
NOTE: $B = \frac{\gamma + \gamma P_w}{288 g}$

P_c = 1000 PSIA

P_c = 800

P_c = 600

P_c = 400

P_c = 1000 PSIA

P_c = 800

P_c = 600

P_c = 400

Y
X
B x 10³

DECLASSIFIED FIG 34

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CHEMICAL CYCLE Second Configuration

DIAGRAM

EQUATION

$$y = \frac{P_c - P_s + x (P_r - P_c)}{B (1 + r) (C_f C^*)^2 - (P_r - P_s)}$$

SYMBOLS AND PARAMETER RANGES

$$B = \frac{\eta_p \eta_t \rho_w}{288 g}$$

$$\rho_w = 4.0 - 4.4 \text{ lb/ft}^3$$

$$\eta_t = 0.4 - 0.7$$

$$\eta_p = 0.5 - 0.8$$

$$P_s = 30 \text{ psia}$$

$$k = 1.4$$

$$P_c = 400 - 1000 \text{ psia}$$

$$x = 0 - 0.8$$

$$B = 0.08 \times 10^{-3} - .266 \times 10^{-3}$$

$$P_r = P_c + 200 \text{ psi}$$

$$MR = .72 - 1.10$$

$$C^* = 6000 - 7340 \text{ ft/sec.}$$

$$C_f = 1.37 - 1.49$$

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TURBINE FLOW VS. COOLANT FLOW

.11

.10

.09

.08

.07

.06

.05

.04

Y

.03

.02

.01

0

$P_c = 1000$ PSIA

$P_c = 800$

$P_c = 600$

$P_c = 400$

$P_c = 1000$ PSIA

$P_c = 800$

$P_c = 600$

$P_c = 400$

	MIN Y	MAX Y
T_f	2060°R	1460°R
B	.000226	.000056

0

.1

.2

.3

.4

.5

.6

.7

.8

X

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FIG. 36

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TURBINE FLOW vs. INLET PRESSURE

11

10

09

08

07

06

05

04

03

Y

02

01

0

400

500

600

700

800

900

1000

P_T

PSIA

	MIN Y	MAX Y
T _T	2060°R	1460°R
X	0	.8
B	.000226	.000086

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F16.37

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TURBINE FLOW
VS.
PRESSURE RATIO

MIN Y MAX Y
T_T 2060°R 1460°R
X 0 .8
B .000226 .000086

DECLASSIFIED FIG. 38

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C/1

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TURBINE FLOW
vs.
COMBINED PARAMETER

MIN Y MAX Y
T_T 2060 °R 1460 °R
X 0 .8

NOTE: $B = \frac{\eta + \eta_p P_w}{288g}$

APPENDIX B

BASIC EQUATIONS FOR TURBOPUMP SYSTEM WEIGHT STUDY

I. INTRODUCTION

The objective here was to determine the kinds of equations which would be required to express turbopump component weights in terms of basic overall system and significant component design parameters.

The development assumes a general turbopump system composed of a single stage axial flow hot gas turbine and a centrifugal pump. The pump was considered as either a radial flow, a mixed flow, or an axial flow machine and it was recognized that more complex systems could be devised by suitable combinations of the various basic components considered.

II. PUMP

A. Two expressions were developed for the range of impeller types. The first (Equation (6)) describes radial flow impellers and those mixed flow impellers in which the flow tends toward the radial direction. The second (Equation (8)) is a better expression for impellers in which the flow parallels the impeller axis.

1. The basic equations describing an impeller are:

$$\psi = \frac{g H}{U_2^2} \quad (1)$$

$$N_s = \frac{N Q^{1/2}}{H^{3/4}} \quad (2)$$

$$q_s = \frac{231 Q}{N d_e^3} \quad (3)$$

$$S = \frac{N Q^{1/2}}{h_{sv}^{3/4}} \quad (4)$$

2. In describing a radial flow impeller, Equation (1) may be rewritten as follows:

$$\psi = \frac{g H}{U_2^2} = \frac{(12 \times 60)^2 g H}{(\pi d_{2m} N)^2}$$

Solving for d_{2m} and substituting for N from Equation (2) we have

$$d_{2m} = \frac{(12)(60)}{\pi N} \sqrt{\frac{g H}{\psi}} = \frac{720}{\pi N_s H^{1/4}} \sqrt{\frac{g Q_p}{\psi}} \quad (5)$$

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Appendix B, continued

-2-

Introducing this equation into the expression for the weight of a cylinder, with an effective length of $\propto$ times the port width, results in the following weight relation:

$$W = \gamma_{I1} \propto \frac{(720)^2 b g Q_p}{4 \pi \psi (N_s H^{1/4})^2} \quad (6)$$

3. Treating the axial flow impeller in a similar manner we have from Equation (3)

$$q_s = \frac{231 Q}{N d_e^3}$$

Solving for d_e^2 and substituting for N from Equation (4) we have

$$d_e^2 = Q_p \left[\frac{231}{q_s S H_{sv}^{3/4}} \right]^{2/3} \quad (7)$$

If the impeller be represented by a cylindrical shell of diameter d_e , length $.4 d_e$ and thickness β inches, then from Equation (7) we have

$$W = \gamma_{I2} \beta \frac{(.4) \pi Q_p}{h_{sv}^{1/2}} \left[\frac{231}{q_s S} \right]^{2/3} \quad (8)$$

B. For purposes of this analysis the volute was approximated by a torus with a minor diameter equal to the volute discharge diameter, and a major diameter D equal to the sum of the volute discharge diameter and the impeller mean diameter. Its thickness is t inches.

1. The basic equation describing volute discharge conditions is:

$$C_3 = \frac{Q_p}{1449 A_v}$$

which in terms of the volute discharge diameter d_v is

$$d_v = \left[\frac{1}{\pi} \frac{144 Q_p}{1449 C_3} \right]^{1/2} \quad (9)$$

2. From stress considerations we have

$$2 S_v t = \frac{H d_v \rho_w}{144}$$

which may be written

$$t = \frac{H d_v \rho_w}{288 S_v} \quad (10)$$

3. Volute weight will be expressed as

$$W = \gamma_v \frac{1}{4} \pi^2 \frac{D}{2} \frac{d_v}{2} t$$

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which, in terms of Equations (5), (9), and (10), may be written as

$$W = \gamma_v \frac{2 Q_p^{3/2} H \rho_w}{449 C_3 S_{tv}} \left[\frac{(12)(60)}{N_B H^{1/4}} \sqrt{\frac{g}{\psi}} + \sqrt{\frac{4\pi}{C_3} \frac{144}{449}} \right] \quad (11)$$

III. TURBINE

A. The turbine wheel was considered as a solid disc with a thickness L inches and a diameter d_t .

1. This development applies to hot wheels the tip speeds of which are limited to 1500 fps in which case

$$u_t = \frac{\pi d_t N}{(12)(60)} = 1500 \text{ fps}$$

which may be written

$$d_t = \frac{(12)(60)u_t}{\pi N} \quad (12)$$

2. The weight of such a wheel may be expressed as

$$W = \gamma_w \int \frac{\pi d_t^2 L}{4}$$

Substituting for d_t from Equation (12) we have

$$W = \gamma_w \int \frac{L}{4\pi} \left[\frac{(12)(60) u_t}{N} \right]^2 \quad (13)$$

B. If the turbine manifold is described in a manner similar to that used for the pump volute, the torus will have a minor diameter (d_m) equal to the manifold inlet diameter and a major diameter equal to the wheel diameter (d_t). Its thickness is t inches.

1. Manifold inlet conditions may be described as follows:

$$Q_t = A V_m$$

which in terms of the manifold inlet diameter d_m is

$$d_m = 24 \sqrt{\frac{Q_t}{\pi V_m}} \quad (14)$$

2. From stress considerations we have

$$2 t S_{tm} = P_t d_m$$

which may be written

$$t = \frac{P_t d_m}{2 S_{tm}}$$

Appendix B, continued

-4-

Substituting Equation (14) we have $t = \frac{12 P_t}{S_{tm}} \sqrt{\frac{Q_t}{\pi V_m}}$ (15)

3. Manifold weight can be expressed as

$$W = \gamma_m \delta L \pi^2 \frac{dt}{2} \frac{d_m}{2} t$$

Substituting from Equations (12), (14), and (15) we have

$$W = \gamma_m \delta \frac{(2)(144)(12)(60) Q_t P_t u_t}{N S_{tm} V_m} \quad (16)$$

IV. RELATED ELEMENTS

A. The shaft was approximated by a solid cylinder the diameter of which may be fixed by stress considerations.

1. Considering the power required to perform the pumping operation we have

$$HP = \frac{(h_d - h_s) \rho_w Q_p}{(1449)(550) \eta_p} \quad (17)$$

2. Power requirements may be related to the shaft diameter as follows:

$$HP = \frac{T N}{63000}$$

where

$$T = S_s \frac{\pi D_s^3}{16}$$

Combining and rearranging we have

$$D_s = \sqrt[3]{\frac{HP \times 16 \times 63000}{S_s \pi N}} \quad (18)$$

3. The shaft weight may be expressed as

$$W = \gamma_s \frac{\pi D_s^2 L}{4}$$

where

$$L = M D_s$$

Substituting from Equations (17) and (18) we have

$$W = \gamma_s \sqrt[3]{\frac{(16)(63000)}{(4)(1449)(550)}} \frac{M \rho_w Q_p (h_d - h_s)}{S_s N \eta_p} \quad (19)$$

B. STRUCTURAL MEMBERS

1. The bearings, seals, and bearing housing will be considered as a hollow cylinder with a diameter D_H , a length $K D_s$ and a thickness t .

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Appendix B, continued

-5-

- a. The weight of such a cylinder is given by

$$W = \gamma_H \Theta \pi D_H K D_s t_H$$

- b. Substituting from Equations (17) and (18) we have

$$W = \gamma_H \Theta D_H K t_H \sqrt[3]{\frac{(16)(63000)}{(449)(550)}} \frac{\pi^2 \rho_w Q_p (h_d - h_s)}{S_s N \gamma_p} \quad (20)$$

2. That portion of the pump and turbine housings not covered in the analyses for the pump volute and for the turbine manifold were treated as follows:

- a. The turbine back plate was treated as an annular ring with an outside diameter d_t , an inside diameter D_H and thickness t_{p1} .

- (1) The weight of such a component is given by

$$W = \gamma_p \lambda t_{p1} \frac{\pi}{4} (d_t^2 - D_H^2)$$

- (2) Substituting for d_t from Equation (12) we have

$$W = \gamma_p \lambda t_{p1} \frac{\pi}{4} \left[\left(\frac{(12)(60) u_t}{\pi N} \right)^2 - D_H^2 \right] \quad (21)$$

- b. Treating the pump back plate in a similar manner we have

$$w = \gamma_p \lambda t_{p2} \frac{\pi}{4} (d_{2m}^2 - D_H^2)$$

Substituting for d_{2m} from Equation (5) we have

$$W = \gamma_p \lambda t_{p2} \frac{\pi}{4} \left[\left(\frac{(12)(60)}{N_s H^{1/4}} \sqrt{\frac{g Q_p}{\psi}} \right)^2 - D_H^2 \right] \quad (22)$$

V. COMPLETE SYSTEMS

Inasmuch as this development is based on the conclusion that a weight study for complete systems cannot be sufficiently general for the application under consideration, no single expression for total system weight was derived. However, such an expression can be developed by combining Equations (6) or (8) with Equations (11), (13), (16), (19), (20), (21), and (22).

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Appendix B

NOTATIONS

b	Impeller port width	in
C_3	Volute discharge velocity	fps
d_e	Impeller eye diameter	in
d_{2m}	Impeller mean diameter	in
D_H	Bearing housing diameter	in
g	Gravitational constant	ft/sec ²
h_d	Total discharge head	ft
h_s	Total head at suction	ft
h_{sv}	Net positive suction head	ft
H	Head rise	ft
K	Ratio of bearing housing length to shaft diameter	-
L	Thickness of turbine wheel at hub	in
M	Shaft length diameter ratio	-
N	Shaft speed	rpm
N_s	Specific speed	-
P_t	Turbine inlet pressure	psi
q_s	Suction specific capacity	-
Q_p	Pump capacity	gpm
Q_t	Turbine flow rate	cfs
S	Suction specific speed	-
S_s	Shaft design stress in sheer	psi
S_{tv}	Volute design stress in tension	psi
S_{tm}	Manifold design stress in tension	psi
t_H	Bearing housing thickness	in
t_{p1}	Turbine back plate thickness	in
t_{p2}	Pump back plate thickness	in

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Appendix B, continued

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U_2	Impeller peripheral velocity	fps
u_t	Turbine wheel tip speed	fps
V_m	Turbine manifold inlet velocity	fps
γ	Weight density of component materials	lb/in ³
ρ_w	Weight density of fluid through pump	-
η_p	Pump efficiency	-
ψ	Head coefficient	

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APPENDIX C

TURBOPUMP RADIATION HEATING STUDY

I. INTRODUCTION

A. The objective of this analysis, which is based on the turbopump design shown in Figure 4 of the text, was to determine the maximum radiation heating which might reasonably be expected in the turbopump system. For this purpose it was assumed that the turbine manifold and the rotating elements in this system are made of stainless steel and that the balance of the system is composed of aluminum alloy. The reactor was assumed to be a right circular cylinder the axis of which coincides with that of the turbopump system. The turbopump assembly was assumed to be two feet from the reactor with no intervening material.

II. NEUTRON HEATING

A. Inasmuch as fast neutrons interact significantly only with materials having a low atomic weight, the only significant fast neutron heating will be that associated with the hydrogen in the system. It was assumed that the effect of secondary gamma radiation is insignificant in comparison to the effect of primary gamma radiation.

B. Consideration of the leakage over the fission energy spectrum from 0.1 to 10 MEV indicates that about 36% of all fast neutrons produced in the reactor leak out. (Assuming that the power of the reactor is 10^{10} watts.) For purposes of calculating the neutron flux, ϕ_r , the reactor is considered as an extended source with a cosine distribution. The flux is considered to be anisotropic at a point on the surface and to obey a cosine law. Thus, the neutron flux at the surface of the reactor is assumed to have the following surface distribution.

$$\phi_r = \phi_o \cos \left(\frac{\pi}{2} \frac{r}{R} \right)$$

where $\phi_o = 15.4 \times 10^{15}$ neutrons/cm² sec

D - Distance from point of flux calculation to reactor face as measured on centerline of system. (cm)

R - Radius of reactor (cm)

r - Radius of point on reactor face (cm)

C. The number of neutrons arriving at any point on the centerline of the system a distance D cm from the reactor face is as follows:

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$$\phi_D = 15.40 \times 10^{15} \int_0^R \frac{D}{2} \left[\cos\left(\frac{\pi}{2} \frac{r}{R}\right) \frac{rdr}{(r^2 - D^2)^{3/2}} \right]$$

D. The average fast neutron energy was assumed to be about 1 MEV and it was assumed that all the energy of a fast neutron was lost on collision with a hydrogen atom. The heating per unit volume of propellant is then

$$\Delta E \sum_s \phi_D \quad \text{MEV/cm}^3 \text{ sec}$$

where

$$\Delta E = 1 \text{ MEV}$$

$$\sum_s = 0.218 \text{ cm}^{-1} \text{ (Macroscopic scattering cross section)}$$

$$\phi_D = \text{Neutron flux at a point D cm from the reactor face (neutrons/cm}^2 \text{ sec)}$$

E. A summary of results indicates that the total heating in the propellant in the turbopump system due to neutron radiation is approximately 2,000 Btu/sec.

III. GAMMA-RAY HEATING

A. Gamma-ray heating in the turbopump system was calculated from the following equation:

$$P = \int_0^l \int_0^{\rho} \int_0^{\rho} \frac{S_s A \mu_a [1 + (\mu_t - \mu_a)l] e^{-\mu_t l} 2\pi \rho d\rho dEdl}{4\pi(\rho_1^2 + \rho^2)}$$

where

P - Heat developed - watts cm²

A - Transverse cross sectional area of system at 1 cm²

μ_a - Gamma-ray absorption coefficient cm⁻¹

μ_t - Total gamma collision coefficient cm⁻¹

ρ - Radial distance from system centerline to point

l - Distance traveled after entering pump system

ρ_1 - Centerline distance from reactor to plane of point

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Appendix C, continued

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B. On this basis and assuming that the gamma-ray heating in the propellant is negligible, the total gamma-ray heating for the entire turbopump assembly is, to a first approximation, 22.5 Btu/sec. This is so small compared to the neutron heating that for first approximations it can be neglected.

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