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NUCLEAR GAS ENGINE

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NUCLEAR GAS ENGINE

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SUMMARY

The nuclear gas engine is a new type of prime mover that seems especially suited to marine propulsion. It consists of a reciprocating engine directly coupled to a gas-cooled reactor. The resulting system is inherently simplified relative to previously proposed nuclear power plants because there is only a single fluid circuit, there are no heat exchangers that must operate at high temperatures and pressures, the system is insensitive to small leaks, and the hazard potential is relatively low. Further, the neutron economy of the gas-cooled reactor is good and consequently the fuel costs are low. These factors appear to lead to over-all power plant costs that are about 60% and weights that are about 45% of those for pressurized-water reactors for units of 20,000 shp. Further, as the design power output is reduced below 20,000 shp the nuclear gas engine should give even lower costs and weights relative to the pressurized-water reactor plants.

Mechanically, the proposed engine in some respects would resemble the uniflow steam engine while in other respects it would resemble two-cycle diesels. Since there would be no fuel injection system, no combustion roughness to pound bearings and piston rings, and no oxygen in the atmosphere in the cylinder to oxidize the lubricating oil, the engine should have the reliability and freedom from service problems of the uniflow steam engine. At the same time, the basic thermodynamic cycle of the nuclear gas engine should give it the high specific output and the excellent fuel economy of the diesel.

The principal uncertainties in this preliminary study of the nuclear gas engine are associated with the pressure losses across the engine valve ports as the gas enters and leaves the engine and the amount of residual gas remaining in the cylinder from one pumping stroke to the next. These factors might make it desirable to reduce the piston speed from the value assumed in this study in which case the over-all power plant cost would be increased by about 15%, but the resulting increase in capital costs would be partially offset by a few per cent reduction in fuel costs which would result from the reduced mechanical friction at the lower piston speed.

INTRODUCTION

The nuclear gas engine was conceived in an effort to evolve a design not hobbled with the complexities and inherent limitations that have been responsible for the high costs characteristic of the nuclear power plants designed to date. In approaching the problem the first step was to review the engineering problems that heretofore have frustrated efforts to bring forth an economical nuclear power plant. These may be grouped in four major categories as follows:

1. Within the limitations imposed by reactor physics considerations it has proved difficult to find combinations of coolant, fuel, moderator, and structural materials that will permit operation at temperatures sufficiently high to give a good over-all thermal efficiency and to make possible the use of reasonably low-cost power-producing machinery.
2. The same limitations make it difficult to evolve a reactor and fuel element design that will give a sufficiently low fuel investment and yet adequate fuel life to yield low fuel costs.
3. Efforts to employ a fluid well suited to removing the heat from the reactor have usually led to separate circuits for the reactor coolant and the working fluid of the thermodynamic cycle. This has made necessary a multiplicity of heat exchangers, pumps, valves, and control equipment which have proved to be expensive. Further, the extra complexity of this equipment has had adverse effects on the inherent reliability of the over-all plant and this, in turn, has led to much duplication of equipment to minimize the effects of a failure.
4. Hazards and shielding considerations have led to elaborate and expensive precautions to minimize the probability of accidents and to provide adequate containment of any accident that might occur.

The need for a good thermal efficiency has led to the conclusion that a temperature of at least 500°F should be achieved in a steam plant, and that a temperature of 900°F would be more desirable. In gas turbine plants a gas temperature at the turbine inlet of at least 1300°F should be achieved, while a temperature of at least 1400 to 1500°F would be much more desirable. At the same time it has become increasingly apparent that the materials available now or in the immediate future cannot be expected to be satisfactory for reactor coolant outlet temperatures in excess of 1200°F.

The various aspects of power reactor design were reviewed recently in an effort to find a system suitable for marine applications. The additional requirements of low output shaft speed, smooth and efficient operation over a wide speed range, and the ability to reverse easily and rapidly provided an intriguing suggestion; a reciprocating engine operating on a gas cycle might give a simple and efficient power plant at a low cost. The preliminary study covered in this report indicates that a reciprocating engine coupled to a gas-cooled reactor with a system pressure of around 100 atmospheres will lead to a cycle efficiency of about 30% and unit outputs from the reciprocating engine equal to or greater than those from existing diesel engines. Such an engine can be coupled to a gas-cooled reactor core of the type delineated in ORNL-2500¹ but of the small size considered in subsequent studies directed at reductions in capital investment.² It has been shown that these small gas-cooled reactors will operate with low enrichments, with a low critical mass, and with a good conversion ratio; they should permit high fuel burnups with low fuel fabrication costs and thus the over-all fuel cost will be low. Studies indicate that, for modest power densities, the materials combination chosen leads to a system exceptionally good from the hazards standpoint, and hence the extra cost and complexity introduced by hazards requirements appears to be substantially less than for other types of reactor.

¹ The ORNL Gas-Cooled Reactor, ORNL-2500, March 1, 1958.

² Design Considerations for High-Pressure Gas-Cooled Reactors with Small Cores, ORNL-CF-58-7-55, September, 1958.

While not mentioned above, a major consideration in the initial development of a new type of nuclear power plant is its sensitivity to small leaks. One of the outstanding features of the pressurized-water reactor and a major factor in its early development was its insensitivity to small leaks from one fluid system to another or from any of the fluid systems to the atmosphere. The ability to tolerate such small leaks helps greatly in achieving the rapid development of new types of equipment. Although experience with the construction of such varied systems as sealed refrigeration units and gaseous diffusion plants indicates that in the long run it is possible to achieve a very high degree of leak-tightness at a nominal increase in cost, the delays involved in shaking down the first few systems if they are sensitive to leaks may be very serious. For this reason a major advantage of the gas-cooled reactor and the nuclear gas engine proposed is its insensitivity to small leaks. In this respect it should be very similar to the British Calder Hall reactors.

The design features of the proposed nuclear engines are described in the following six major sections. The first section consists of an outline of the principles of operation of the proposed nuclear gas engine and a short discussion of typical variations. The second section gives a comparison of the nuclear gas engine with other reciprocating engines and air compressors. The third section presents a comparison between the ideal and the actual cycles on which the proposed nuclear engine would operate, with particular attention given to the pumping, heat, and other losses to be expected. The fourth section outlines a typical nuclear gas engine power plant and compares it with a power plant that is being built. The fifth section discusses the field of application and the development potential of the nuclear gas engine. The sixth section discusses the major development problems, and suggests approaches that might be followed to solve them.

PRINCIPLES OF OPERATION

The proposed nuclear gas engine is a reciprocating engine designed for a thermodynamic cycle in which it compresses a gas and delivers it to a reactor where the gas is heated. The hot gas is then returned to the cylinder in which it is expanded to do work. At the end of the expansion stroke the piston uncovers ports which permit loop-scavenging of the cylinder by a low-pressure gas system which includes a cooler and blower to complete the thermodynamic cycle. The cycle is shown schematically in Fig. 1. The piston would approach the top of the cylinder very closely so that the clearance volume within the cylinder would be trivial. The discharge valve would open as the piston approached top center and would close as the piston reached top center. The return valve would open at top center to admit hot gas and would remain open until the piston had moved part way down in the cylinder. A pressure-volume diagram for a typical cycle is shown in Fig. 2. The gas temperatures at key points in the cycle are indicated by the adjacent numbers.

Choice of Gas

From the materials compatibility standpoint the most promising working fluids for this application are helium and nitrogen. Either can be used with stainless steel, carbon steel, and graphite up to temperatures well above 1500^oF. Helium is attractive both because of its good heat transfer characteristics in the reactor and because its high ratio of specific heats makes possible a high thermal efficiency with a volumetric compression ratio of only 3 to 1, as compared with the volumetric compression ratio of around 25 to 1 that would be required to obtain a comparable efficiency in reciprocating steam engines. It was the low ratio of specific heats of steam which led to the use of triple-expansion engines as a means for improving the fuel economy of reciprocating steam engines. The lower pressure cylinders in these engines were so large and heavy that the engine weight became very large, even for moderate power outputs.

FIG. 1

SCHEMATIC DIAGRAM OF A NUCLEAR GAS ENGINE WITH
A GAS-COOLED REACTOR

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FIG. 2. IDEALIZED P V DIAGRAM FOR A NUCLEAR
GAS ENGINE TO OPERATE ON HELIUM WITH
A COMPRESSION RATIO OF 3:1

Nitrogen could be used as the working fluid and a good thermal efficiency could be obtained with a compression ratio of about 5 to 1. Nitrogen has the advantage that it can be obtained simply and inexpensively, but the mean effective pressure and hence the output from an engine of a given size is about one-third less than for helium. The poorer heat transfer performance of nitrogen would also yield a somewhat lower output from a given size of reactor. Its most important shortcoming relative to helium is that it will entail much higher pumping losses in both the high- and the low-pressure circuits.

Comparison of Idealized Cycles

It is of interest to compare the performance of the more common operating cycles with that of the proposed nuclear gas engine. Figure 3 shows the over-all thermal efficiency for some typical idealized cycles. The relative performance of these cycles as a function of compression ratio is somewhat different in actual engines because of losses associated with deviations from the ideal isentropic compression and expansion processes, pumping losses, and mechanical friction losses. Figure 4 presents curves for the thermal efficiency as a function of the peak temperature in the cycle for cycles suitable for use in a nuclear power plant. The calculations for these included the effects of the principal losses. Note how severely the simple Brayton, or gas turbine cycle is penalized in Fig. 4 as compared with Fig. 3, particularly for the lower temperatures. This is caused mainly by the losses associated with deviations from the ideal isentropic compression and expansion processes.

In the Rankine cycle the work required for compression (i.e., for the feedwater pumps) is only a few per cent of that available from the expansion process so that a loss of 15% in each process will reduce the net work and over-all thermal efficiency by only about 16%. In the gas turbine, on the other hand, the ideal work input to the compressor is about half of the ideal work output from the turbine. As a result, if the eddy losses in the compressor and turbine blading increase the work input to the compressor by 15% and reduce the work output from the turbine by 15%, the net work and the thermal efficiency will be cut almost

FIG. 4. EFFECTS OF PEAK GAS TEMPERATURE ON OVERALL CYCLE EFFICIENCY FOR TYPICAL CYCLES SUITABLE FOR NUCLEAR POWER PLANTS USING OPTIMIZED COMPRESSION AND EXPANSION RATIOS. ALLOWANCES FOR DEVIATIONS FROM IDEAL ISENTROPIC PROCESSES, PUMPING LOSSES, AND MECHANICAL FRICTION HAVE BEEN INCLUDED, BUT THERMAL RADIATION AND OTHER MISCELLANEOUS LOSSES HAVE NOT.

in half. Much of this loss can be recuperated through the use of regenerative heat exchangers, but these are bulky, expensive, and complex, and they increase the pumping losses substantially.

While heat losses caused by thermal convection and radiation to the surroundings are appreciable, they are similar in magnitude for the various power plants considered here and should not exceed about 5% in large units. It is clear from Fig. 4 that the nuclear gas engine has an important advantage over the gas turbine in the temperature range achievable in reactors with present materials, namely gas temperatures of 1000 to 1200°F at the beginning of expansion.

Otto Cycle Nuclear Gas Engine

A nuclear gas engine could also be built with a molten salt or sodium-cooled fuel element reactor as a heat source. An opposed-piston engine could be employed as indicated schematically in Fig. 5. The pistons would be built with flat heads and installed so that there would be virtually no space between them at top center. As they approached top center, gas would be forced out into the toroidal heat exchanger surrounding the center of the cylinder where it would swirl between radial fins about 1/16 in. thick on 1/8-in. centers. The fins would be heated by NaK from a heat exchanger coupled to the reactor. Most of the gas heating would take place close to top center because the swirl velocity and the gas density in the heat exchanger would be high only at that time, and the engine would operate on an Otto cycle as indicated in Fig. 6. Very little gas would circulate into the torus during the scavenging process which would occur while the pistons were close to bottom center.

The arrangement of Fig. 5 has the advantages that the reactor and shield assembly would be lighter and more compact than for a gas-cooled reactor, and the NaK pipes would be much smaller in size than gas ducts. Another advantage of the engine of Fig. 5 is that it includes no valves.

The principal difficulties to be expected in an engine built to operate as indicated in Fig. 5 are associated with control of the output and with thermal stresses in the high-performance toroidal heat exchanger. Control of the engine output could be accomplished by varying the pressure

FIGURE 5

OPPOSED PISTON NUCLEAR GAS ENGINE

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FIG. 6. OTTO CYCLE FOR A NUCLEAR GAS ENGINE
OPERATING ON HELIUM WITH A COMPRESSION
RATIO OF 3:1.

level in the cylinder, by varying the temperature of the heat exchanger surfaces in the torus, or by varying the scavenging air flow rate. It would be difficult to accomplish rapid variations in either the system pressure or the heat exchanger temperature, and hence only control of the scavenging gas flow rate appears to be suited to applications requiring rapid changes in output. The engine would not be reversible because the timing of the piston porting for the cooling portion of the cycle would permit operation in only one direction unless the phasing of the crankshaft were changed, in which case a quite complicated and massive linkage would be required. The most important disadvantages of this system are that it would be moderately sensitive to leaks, and thermal stresses and vibration in the high-performance toroidal heat exchanger would pose very serious problems.

COMPARISON OF THE NUCLEAR GAS ENGINE WITH CONVENTIONAL
RECIPROCATING ENGINES AND AIR COMPRESSORS

Hot-Air Engine

Probably the first engine with which a comparison should be drawn is the old Sterling, or Erickson, hot-air engine. These engines are very low-speed, bulky, heavy machines. This came about in part because the pressure at the beginning of compression was atmospheric whereas in the proposed nuclear engine the corresponding pressure would be approximately 11 atmospheres. Because of this the indicated mean effective pressure in the nuclear gas engine would be increased about 11 times while the brake mean effective pressure would be increased about 15 times relative to the hot-air engines. In the second place, the old hot-air engines had to operate at very low speeds because the heat transfer processes took place within the cylinder and the heat transfer area was insufficient to accomplish the required heat transfer quickly. As a result, piston speeds far below those commonly used in other types of reciprocating engine were necessary. In the proposed engine, normal piston speeds appear to be practicable, and therefore the output from a given size of engine could be increased by a factor of about 10 relative to the old hot-air engines. The combined effects of these two factors should lead to an increase in brake horsepower output from a given size of engine by a factor of ~ 150 . The importance of these factors is indicated by the marked improvement achieved by Phillips in its efforts to develop a modern version of the old hot-air engine.³

Reciprocating Steam Engine

The similarity of the reciprocating steam engine to the proposed nuclear gas engine makes it important to consider the reasons for the modern preference for steam turbines. The principal reason is fuel economy -- modern steam turbine plants have overall thermal efficiencies half again as high as those of the best reciprocating steam engines.

³ Phillips' Technical Review, Vol. 8, No. 5, May 1946, p. 129.

The latter are limited to relatively low steam pressures and temperatures by cylinder wall lubrication considerations. If the cylinders are not cooled, experience shows that it is not practicable to use steam at pressures and temperatures above 285 psi and 650°F (the saturated steam temperature at this pressure is 417°F). If the cylinder walls were cooled to facilitate lubrication, the heat losses would be excessive because of the high heat transfer coefficient associated with condensing steam. Another limitation is the expansion ratio feasible in a single cylinder. The drop in steam temperature on expansion coupled with the high heat transfer coefficient causes the bore to vary between the cylinder head and the lower dead center position of the cylinder. This results either in excessive diametral clearance at the top center position or insufficient clearance at bottom center (which would cause the piston to seize in the cylinder). The use of a multiple-expansion engine (double-, triple-, or quadruple-expansion) leads to large cylinders and excessively expensive, bulky, and heavy engines.

The flexibility of the power plant and the ease with which it can be started and reversed are very important considerations for some types of application, such as in harbor tugs, ferries, and ice breakers. For such types of service the reciprocating steam engine is unequalled in performance. It can be accelerated, decelerated, and reversed more rapidly than any other type of power plant. While this feature is an advantage also for merchant marine vessels, the advantage is less important because only a relatively small portion of their operation is in harbors.

Diesel Engines and Steam Turbines

In comparing diesel engines with steam turbines it has been found that the fuel economy of the diesel is somewhat better than that of the steam turbine plant. However, the weight and cost of a diesel engine increases as approximately the 1.5 power of the horsepower because the propeller shaft speed must be reduced with increasing horsepower. At the same time, the specific weight and cost of a steam turbine plant fall off as the power output is increased. This comes about in large measure because the weight and cost of the auxiliaries such as feedwater pumps,

feedwater heaters, etc., become relatively less. Figure 7 shows these effects and indicates the reason for the almost universal use of diesels for the power range below 5,000 bhp, and the widespread use of steam turbines for applications requiring over 10,000 bhp.

Another factor has been that the U. S. Navy has been interested in power plants of very high horsepower, and since in these large power plants the steam turbine plant is lighter than the diesel, the bulk of U. S. naval power development has been centered on the steam turbine-type of plant. Since there was little market in the United States for maritime power plants between World Wars I and II, U. S. diesel engine manufacturers made no particular effort to develop a market for large diesel engines. However, in Europe large diesel engines are more commonly used than steam turbines in new units for the merchant fleet. Units up to 25,000 bhp are currently being built. A recent survey of the literature indicates that 28 of 34 ocean-going merchant ships under construction in European and Japanese yards in 1957 were to be diesel-powered. All of these large diesel engines were of a size such that the bore and stroke in each case was very nearly 30 in. x 50 in. Of the 28 diesel installations, 24 were direct drive, 2 used reduction gears, and 2 electric drives. The European preference for the diesel apparently springs from its better fuel economy, which averaged around 0.35 lb/shp-hr as compared with about 0.50 lb/shp-hr for comparable steam turbine plants.

The proposed nuclear gas engine employs a peak pressure of 1500 psi which is somewhat higher than the 900 to 1300 psi usual in current diesels. However, irregularities in diesel engine combustion cause very high rates of pressure rise which impose severe shock loads on engine parts, particularly the pistons, rings, and bearings. The low rate of pressure rise in the nuclear gas engine should permit the use of substantially higher peak pressures than those common in diesels for an engine of similar proportions and weight.

The mechanical efficiency of steam, diesel, and gas engines, as well as that of air compressors, is from 90 to 95% for the piston speeds usual in marine engines. The nuclear gas engine would yield somewhat

FIG. 7. TREND OF EFFECTS OF POWER ON TYPICAL PROPULSION PLANT WEIGHTS AND COSTS FOR BOTH STEAM TURBINES.

lower values because of the higher ratio of the average cylinder pressure to the mean effective pressure.

The bulk of the service difficulties experienced with diesel engines spring from combustion or corrosion difficulties associated with the fuel oil, or from engine deposits caused by oxidation of the lubricating oil. In the nuclear engine, elimination of the fuel injection nozzles, combustion roughness, and sulphur and water contamination of the cylinder walls, coupled with a nonoxidizing atmosphere for the lubricating oil should go a long way toward increasing the reliability and reducing the maintenance required to the levels experienced with air compressors and uniflow steam engines.

Valve Gear

The valve gear of the nuclear gas engine will have to be different from that of an internal combustion engine because the pressure differential is in toward the cylinder rather than outward from it, and hence the valves should open outwardly from the cylinder. Thus the valve gear for the nuclear engine will look more like that for a steam engine than for an internal combustion engine. However, the fairly high gas temperature would make the type of balanced pressure composite poppet valve used in many reciprocating steam engines unsatisfactory. Good starting and reversing characteristics could be obtained with poppet valves somewhat similar to internal combustion engine valves if pneumatically-operated bleed valves of the type used for air-starting diesels were employed. These would be phased to lead the cylinder discharge by about 5 deg so as to break the pressure differential across the valve and hence reduce the opening loads to nominal values.

Oil Entrainment and Coking Problems

Oil entrainment by the gas flowing through the cylinders may pose problems peculiar to the nuclear gas engine. In the first Calder Hall reactor, faulty shaft seals on the main circulating blowers permitted relatively large quantities of lubricating oil to enter the gas system. Some of this oil was deposited in the reactor where it formed a tarry coke in the joints between many of the fuel elements. The Calder Hall reactor design is such that six fuel elements are stacked in each channel

with the lower fuel elements supporting those above. The formation of the tarry coke deposit in the joints between the fuel elements caused them to stick together and led to serious difficulties in removing them. Because of this experience the gas at Calder Hall has since been monitored for hydrocarbon content (as well as H_2O) and the hydrocarbon concentration is kept to less than 5 ppm. However, the proposed reactor will differ from those at Calder Hall in two vital respects. First, the gas will not recirculate in the high-temperature loop but will be discharged to the cooler where any oil vapor will condense, deposit on the heat exchanger tubes, and drain off to be returned to the oil system through the bypass filter. Secondly, both the gas and the fuel element surface temperature will run much hotter, the former 600 to 1100°F and the latter 800 to 1300°F with some local hot spots as high as 1500°F. Thus the oil will be in the vapor phase as it passes through the reactor for it will be in a nitrogen atmosphere and the partial pressure of the oil vapor will be small. Reciprocating engine and air compressor experience indicates that the oil concentration in the gas will be around 15 to 60 ppm. Thus the total quantity of oil circulated through the reactor from all of the engines will be of the order of 1.0 lb per min.

Experience with thermal cracking units in petroleum refineries indicates that only a small amount of thermal cracking would occur during the short time interval in which the gas passes through the reactor, i.e., about 0.12 sec. Some further cracking would also occur during the 1.0 sec that the gas would remain at 1100°F while traversing the return duct to the reactor, but this, too, would affect only small amounts. In either case, if carbonization did occur, it would give a very finely divided carbon black that would be swept along with the gas rather than the tarry coke characteristic of liquid-phase cracking.

Radiation damage data indicate that the effects of radiation are simply additive to those of high-temperature and are of the same character. The gas transit time through the reactor is so short that the radiation dose to the oil in a single transit would be only about 6000 rad as compared to the 10^8 rad required for noticeable radiation damage.

Thus practically all of the damage to the oil will result from thermal effects. This will lead to the formation of unsaturated

hydrocarbons. While the returning oil vapor will tend to deposit on cool surfaces such as the valves, cylinder walls, and pistons, the damaged molecules will be diluted even further by the oil films already present, and will mix into the oil circulated through the engine.

To bring the matter into perspective, it is helpful to compare the above conditions with those in a conventional diesel engine. In such an engine all of the oil entrained in the exhaust gas is lost and accounts for a major part of the relatively small oil consumption characteristic of the engine. On the other hand, in the nuclear gas engine all of this oil will be returned to the cylinder with only a small part damaged. Thus the effective oil consumption will be drastically reduced. It is also possible to draw a parallel with the organic-cooled and moderated reactor. Again the comparison is favorable -- the quantity of organic material in such reactors is about one million times greater than would be the case for the proposed nuclear gas engine.

The tendency toward the formation of engine deposits will depend in part on the type of lubricating oil. The lower temperatures of the cylinder wall and gas in the nuclear gas engine will help in that they will permit the use of lighter, more stable oils than can be used in diesel engines. A vacuum-distilled straight mineral oil free of additives looks promising. Radiation damage data indicate that a glycol base lubricant should give much less difficulty with deposits than a petroleum base lubricant because thermal or radiation damage do not tend to produce gums, varnish, or tars. This suggests that a glycol base lubricant should be considered.

Oil accumulation on the surfaces in the cool low-pressure portion of the loop would be certain to occur, and allowances for this should be made in the design of the heat exchanger. The oil film formed should be thin and relatively free of sludge, tars, and varnish because little oxidation of the oil would occur in the inert atmosphere of the system. The cooler should, of course, be designed to facilitate drainage of the oil that deposits on the surfaces, and the geometry should be chosen so that the oil will drain off to a sump from which it can be returned to the main oil system, preferably through the filter.

COMPARISON OF THE IDEAL AND THE ACTUAL OPERATING CYCLES

The power and over-all thermal efficiency of an actual engine can be estimated by comparing the ideal cycle and the actual cycle in detail with full allowance for the various losses to be expected. Fortunately, there is a substantial background of experience in carrying out such analyses for both steam engines and internal combustion engines, so that the techniques available and the types of loss to be expected are well understood.

Pumping Losses

The greatest single loss associated with the actual cycle will be the pumping power required to circulate the gas through the reactor for the heating phase of the cycle and to force the gas in and out of the cylinder and through the cooler for the cooling phase of the cycle. The nature of these losses is indicated in the PV diagram of Fig. 8. Analyses indicate that, with nitrogen, these losses would be of the order of 15% of the brake horsepower output, whereas with helium they would be only about 3% of the brake horsepower output. The difference, of course, arises from the difference in the densities and specific heats of the two gases.

Charge Dilution by Residual Gas

A second loss to be expected in the actual cycle arises from dilution of the charge at both upper and lower dead centers. At upper dead center not all of the gas will be discharged from the cylinder at the end of the compression stroke. The residual gas remaining in the clearance volume will mix with and dilute the gas admitted to the cylinder from the reactor during the power stroke. This dilution will reduce the effective temperature of the gas during the power stroke and will result in a loss in power relative to an ideal cycle based on the reactor exit gas temperature. Similarly, loop scavenging of the cylinder close to bottom dead center in the cooling portion of the thermodynamic cycle will not reduce the temperature of the gas remaining in the cylinder to the level of the gas leaving the cooler. For this reason the temperature of the gas and

FIG. 8. EFFECTS OF PUMPING LOSSES ON THE IDEALIZED INDICATOR DIAGRAM FOR THE NUCLEAR GAS ENGINE.

its volume at the beginning of the compression stroke will be higher than for an ideal cycle based on the temperature and density in the intake manifold. The difference between the ideal and the actual temperatures at the beginning of compression will depend in part on the scavenging ratio; that is, the ratio of the volume of gas forced through the cylinder each cycle to the volume remaining in the cylinder at the beginning of compression. The usual scavenging ratio in two-cycle diesel engines is approximately 1.5. It is believed that in the nuclear engine it will be worthwhile to make use of a scavenging ratio of approximately 2 because of the greater sensitivity of the cycle at low peak temperatures to the initial gas temperature. Further, the power loss associated with a high scavenging ratio may be less for an engine with a closed loop than for one in which the scavenge air would be dumped to the atmosphere.

The pumping power required to scavenge the cylinder in the low-pressure portion of the cycle might be supplied by making use of an exhaust turbine-driven blower. This would make use of the higher energy in the exhaust gas to drive a turbine which would in turn supply power to a compressor for circulating the fluid in the low-pressure loop. It is possible that it might be worthwhile to attempt to recover some of the energy in the exhaust blowdown from the cylinder by using a carefully designed exhaust manifold system between the cylinder exhaust ports and the turbine nozzle box. However, the low exhaust temperatures in the nuclear reciprocating engine will make it more difficult to employ an exhaust turbine-driven blower than has proved to be the case in diesel engines. It may prove best to use an engine-driven blower supplemented by a turbine-driven unit.

Another major loss associated with pumping the fluid in and out of the engine is the loss in effective stroke associated with the scavenging operation near lower dead center. During the scavenging process the piston movement is not accompanied by any useful work, but simply serves to control the intake and exhaust ports. Thus the mean effective pressure for the entire stroke is reduced by the ratio of the effective portion of the working stroke to the total stroke. This can be deduced readily from Fig. 9.

FIG. 9. EFFECTS OF HEAT LOSSES TO THE CYLINDER WALLS AND FLOW LOSSES THROUGH THE PORTS ON THE INDICATOR DIAGRAM OF A NUCLEAR GAS ENGINE.

Heat Losses to the Cylinder Walls

Heat losses to the cylinder wall are also important. Fortunately, heat losses to the walls during the compression stroke reduce the work input required for compression. The reduced gas temperature can then be made up by a greater temperature rise through the reactor. The heat losses to the walls during the expansion stroke cause a loss in pressure and hence a loss in power (see Fig. 9). The magnitude of these heat losses can be estimated on the basis that they should be similar to the losses experienced in internal combustion engines. Experience with a wide variety of engines in large sizes has indicated that the heat losses to the cylinder walls are normally fairly close to 25% of the heat added during combustion. If the nature of the heat losses in internal combustion engines is examined carefully and compared with that of the heat losses to be expected in a nuclear reciprocating engine, a number of important points emerge. A substantial portion of the losses in an internal combustion engine occur in the form of thermal radiation to the cylinder walls during the combustion process because of the high emissivity of CO_2 and H_2O vapor in the temperature range prevailing during combustion. These losses certainly would not appear in the nuclear reciprocating engine. On the other hand, turbulence in the gas stream returning from the reactor to the engine cylinder would tend to give a rather high heat transfer coefficient between the charge in the cylinder and the cylinder walls. However, the effects of this turbulence on the heat transfer coefficient are probably similar to those of the intense turbulence associated with combustion.

Closely associated with these losses are the heat losses to the intake and exhaust port walls. About one-third of the heat loss from an internal combustion engine is to the walls of the exhaust port. In the nuclear engine the use of both discharge and return ports will tend to increase the losses during the high temperature portion of the cycle. This loss can be reduced drastically if the ports are lined in such a way as to obtain some thermal isolation of the port walls from the water-cooled cylinder head. Fortunately, a lining of this character can be made a portion of the internal manifolding for the hot gases in the cylinder head. It will not be essential to use thermal insulation between

the lining and the cylinder head since a gas gap will suffice to reduce the losses to a trivial level.

The mean effective gas temperature for the entire cycle is usually around 1200°F for the cylinder head of a spark ignition engine, while the cylinder head metal temperature will usually be around 250°F . This gives a mean effective temperature difference between the cylinder head and the gas of about 950°F . A mean effective gas temperature of around 650°F prevails for the lower portions of the cylinder barrel. A rough estimate of the mean effective gas temperature for the nuclear engine indicates that it will be around 500°F for the cylinder head and about 150°F for the lower portions of the cylinder barrel. Thus, if the cylinder head is water-cooled, the metal temperature for the inner surfaces will be roughly 250°F for the nuclear engine, giving a mean effective temperature difference between the gas and the cylinder head of about 250°F . This is approximately one-fourth of the mean effective temperature difference for the spark ignition engine, and implies that the heat losses from the nuclear engine would be only about one-fourth those for a spark ignition engine. On the other hand, the gas density in the nuclear engine will be about ten times higher. Since the heat transfer coefficient between the gas and the cylinder walls would be expected to vary as the 0.8 power of the density, the heat transfer coefficient should be expected to be about six times higher for the nuclear engine. At the same time, the mean effective gas temperature for the lower cylinder wall (as contrasted to the cylinder head) will be about equal to the cylinder wall temperature in the nuclear engine, except for the upper third of the stroke and the immediate vicinity of the exhaust port leading to the cooler. This should reduce the heat losses in a nuclear engine.

It is evident from the above discussion that the situation is quite complex. However, it does appear that the heat losses to the walls of a properly designed nuclear engine might very well be a smaller percentage of the total heat input than has been found to be the case in internal combustion engines.

Valving Losses

Another important source of loss between the ideal and the actual indicator diagrams is that associated with rounding of the corners of the diagram for the actual cycle because the valves and ports do not open instantaneously and because the gas flow through these valves and ports takes an appreciable period of time. These losses depend in large measure on the speed with which poppet valves may be opened and closed. Fortunately, the dynamics of poppet valve operation have been studied intensively by many organizations and much information is available.

A simple preliminary estimate of practicable valve opening and closing rates can be made on the basis of similarity considerations. If the valve diameter and lift are kept directly proportional and the same cam profile is used, the relative times for opening can be derived from the fundamental relation:

$$S = at^2/2$$

where

S = half the lift, ft

a = average acceleration, ft/sec²

t = time, sec

The angle of crank rotation, α , for lifting the valve halfway can be related to the rpm, N , and to the time by:

$$t = \frac{\alpha}{360} \cdot \frac{60}{N} = \frac{\alpha}{6N}$$

Substituting in the previous equation gives:

$$S = \frac{a}{2} \frac{\alpha^2}{36N^2} = \frac{a\alpha^2}{72N^2}$$

Rearranging to put the expression in terms of α gives:

$$\alpha = N \sqrt{\frac{72S}{a}}$$

The crank angles required to lift the valve halfway in two geometrically similar engines of different size in which the average valve acceleration is kept the same thus becomes:

$$\frac{\alpha_1}{\alpha_2} = \frac{N_1}{N_2} \sqrt{\frac{S_1}{S_2}} = \frac{N_1}{N_2} \sqrt{\frac{D_1}{D_2}}$$

where D is the cylinder diameter.

The above analysis does not take into consideration the constant velocity ramp normally employed for the initial lift. This factor may tend to increase somewhat the crank angle required for low-speed engines.

Table I has been prepared to show the results obtained by applying the above relations to scale the Rolls Royce Merlin valve gear up to an 18-in. diameter cylinder for a nuclear gas engine. The Merlin employed four valves in the head (in an arrangement somewhat similar to Fig. 19, p. 57). On this basis, the valves of an 18-in. bore, 330-rpm, nuclear gas engine could be opened halfway in 12 deg of crank rotation. Since the valve overlap should be small to avoid dilution of the hot return gas by the discharge flow, only about 50 deg of crank rotation would be available for the entire discharge valve opening and closing events. At first glance it would appear from this that adequate time would be available. A closer inspection, however, discloses other problems. The force required of the valve springs to hold the rocker arm against the cam during the central portion of the valve opening event may be difficult to provide and hence the cam profile might be different from that of the Merlin in this region. There is also the question of reliability. While essentially no trouble was experienced with the Merlin intake valves in service, the much greater operating life required for marine applications might prove to be a major problem. (The Merlin exhaust rocker arm pads showed some signs of pitting after long service, but the exhaust valve opening loads were about 3 times greater than for the intake valves.)

As a point of interest, data are also given in Table I for a Nordberg diesel engine recently developed for the Navy. Note that this second set of data indicates that the time required to open the valves of the hypothetical nuclear engine would be half again as great as the value derived from the Merlin. The incentives to open the valves rapidly are much greater in the nuclear engine than in a diesel; however, and hence may justify more expensive valve gear.

Scavenging Losses

The intake and exhaust ports at the base of the cylinder barrel for scavenging and cooling the gas will probably be timed so that the exhaust port will open approximately 20% of the stroke and the intake

TABLE I

Comparative Data on the Crank Angles Required for Operating Poppet Valves

<u>Engine Designation</u>	<u>Rolls-Royce Merlin</u>	<u>Hypothetical Nuclear Gas Engine with Valve Gear Similar to Merlin</u>	<u>Nordberg FMD-1116-HSC</u>	<u>Hypothetical Nuclear Gas Engine with Valve Gear Similar to Nordberg</u>
Bore, in.	5.4	18	11	18
Piston speed, fpm	3000	1200	1800	1200
Crankshaft rpm	3000	330	900	330
Crank angle for half-opening, deg (from middle of ramp)	60	12	46	18.7
Valve diameter, in.	1.8	1.6	4.9	6
Valve lift, in.	0.59	1.77	1.06	1.77
Number of valves per cylinder	4	4	2	4
Valve running clearance, in.	0.010	0.020		0.020
Valve velocity (maximum), fps	33		8.3	
Valve acceleration (maximum) ft/sec ²	23,200	23,200		

port approximately 12% of the stroke from bottom center, or at 60 and 45 deg of crank rotation from bottom center, respectively. These port timings are consistent with conventional practice for two-cycle diesel engines, and should permit a scavenging ratio of between 1.5 and 2. With a scavenging ratio of this order it is estimated that the average gas temperature in the cylinder at the beginning of compression will be about 30°F above the intake port temperature.

The rounding of the indicator diagram corners associated with the exhaust blowdown process at the end of the power stroke will not entail an appreciable loss in indicator diagram area relative to the ideal because, as indicated in Fig. 9, the loss in diagram area immediately prior to the ideal port opening event is roughly balanced by the gain in indicator diagram area during the period immediately following the ideal port opening event because of the appreciable period required for the exhaust blowdown process.

There is some loss associated with the expulsion of gas from the cylinder as the piston moves upward from the point where it closes the intake port until it closes the exhaust port. A preliminary estimate indicates that 3 or 4% of the cylinder charge will be expelled through the exhaust port during this interval. Thus for a given effective pressure in the cylinder at the initiation of compression it will be necessary for the low-pressure gas system to operate at a pressure 3% higher than for the idealized case where this loss would not occur. If this is done no appreciable loss in imep or thermal efficiency will occur.

Mechanical Friction

The loss in power caused by mechanical friction in the engine depends on piston speed. Data on large internal combustion engines indicate that, at a piston speed of 1200 fpm, this loss will be approximately 5% of the indicated power developed in the cylinder. It increases as approximately the 1.7 power of the piston speed so that it is about 12% of the brake horsepower at a piston speed of 3600 fpm, for example. About half of this mechanical friction is in the pistons and rings while the balance is in the connecting rod and main bearings and in miscellaneous smaller items such as the valve gear. A further loss of 1 to 3%

is entailed by the power required to drive engine accessories such as the lube oil pump, jacket water pump, etc.

Over-all Effects of Losses

Preliminary estimates of the above losses as calculated for the typical cycles of Figs. 8 and 9 are summarized in Table II. The basic cycle calculations were carried out using the data for nitrogen given in the "Gas Tables" by Keenan and Kaye. These tables include the effects of variations in specific heat with temperature. The results of a typical calculation are presented in Table III. Note that an allowance was made in Table III for dilution of the gas returning to the cylinder by the residual gas in the clearance volume, which was assumed to be 1% of the total piston displacement.

It is evident from Table II that the greatest losses are those associated with pumping and with charge dilution by residual gas. In view of this, the effects of the principal engine design parameters on engine output and over-all thermal efficiency have been calculated with allowance for the pumping power losses and dilution since these effects are easily calculated. Allowance for the heat losses to the cylinder walls and rounding of the indicator diagram corners by the valving and porting processes are much more difficult to estimate, and have been omitted because their effects should be relatively small. Figure 10 shows the effects of compression ratio on both the imep and the thermal efficiency for both nitrogen and helium. Figure 11 shows the effects on the same factors of the gas temperature at the outlet of the reactor, while Fig. 12 shows the effects of the gas temperature in the cylinder at the beginning of the compression stroke. Note that the nuclear engine is more sensitive to the initial temperature at the beginning of compression than is an internal combustion engine because of the much smaller peak temperature in the cycle. Appendix A gives the details of the calculations on which Figs. 10, 11, and 12 were based.

TABLE II

Summary of Estimated Losses Caused by Differences
between the Ideal and the Actual Cycle

Reactor Gas Outlet Temperature = 1100°F

<u>Principal Factors</u>	<u>Nitrogen (Comp. Ratio = 4.8:1)</u>			<u>Helium (Comp. Ratio = 2.5:1)</u>		
	<u>Magnitude of Loss in mep(%)</u>	<u>Resulting Performance mep (psi)</u>	<u>Thermal Eff.(%)</u>	<u>Magnitude of Loss in mep(%)</u>	<u>Resulting Performance mep (psi)</u>	<u>Thermal Eff.(%)</u>
Ideal cycle imep and thermal efficiency		246	40.0		309	35.3
Less:						
Pumping losses	13	214	34.8	3	300	34.3
Cylinder charge dilution by residual gases	16	178	36.0*	14	262	36*
Portion of stroke required for scavenging	14	156	36.0	14	230	36
Heat losses to cylinder walls	5	149	34.2	5	218	34.2
Noninstantaneous valve operation	1	147	33.9	1	216	33.9
Mechanical friction in engine	10	132	30.5	10	194	30.5
Power to accessory drives	1.7	130	30.0	1.7	192	30.0
Estimated actual cycle bmep and efficiency		130	30.0		192	30.0

* Note: There is an apparent anomaly here in that the thermal efficiency for some compression ratios actually increases as a result of residual gas dilution because the dilution leads to an increase in the expansion ratio. In these cases the net effect happens to give a small increase in thermal efficiency.

TABLE III

Typical Thermodynamic Calculation for a Nuclear Gas Engine Cycle
with Allowances for Residual Gas Dilution
and Pumping Losses

Working fluid = N_2

Compression ratio = 4.8:1

Reactor outlet temperature = 1100°F

Initial specific volume = 37.05 ft³/lb-mol

Initial density = 1.322 lb/ft³

Clearance volume at top center = 1% of initial cylinder volume

Volumetric ratio of residual to total charge = 0.01/0.2085 = 0.048

Drop in temperature of returning charge = 0.048 x 485 = 23.5°F

Charge temperature at beginning of expansion = 1560 - 23.5 = 1536°R

Point in Cycle	Temp. (°R)	Enthalpy (Btu/lb-mol)	Relative Pressure (P_r)	Relative Volume (V_r)	Pressure psia	Ratio of Gas Volume to Initial Volume
1. Begin Comp.	580	4028.7	1.3200	4715	168.5	1
2. End Comp.	1075	7515	11.740	983	1500	0.2085
3. Begin Exp.	1536	10,921	44.20	372.9	1456	0.3065
4. End Exp.	990	6908	8.732	1217	287	1

Assume that pressure drop in low-pressure loop is 14% of total pressure:

$$\text{Net work} = (h_3 - h_4) - (h_2 - h_1) + 0.86 (P_4 - P_1) V_1 (144/778)$$

$$\text{Net work} = 4013 - 3486 + 102 \times 37.05 \times 0.1852 = 1227 \text{ Btu/lb-mol}$$

$$\text{Thermal efficiency} = 1227/3406 = 36.0\%$$

$$\text{imep} = (1227 \times 778)/(144 \times 37.05 \times 1.143) = 156.3 \text{ psi}$$

(factor of 1.143 allows for 1/8 of stroke lost during scavenging)

FIGURE 10. EFFECTS OF COMPRESSION RATIO ON THE THERMAL EFFICIENCY AND IMEP OF THE NUCLEAR GAS ENGINE CYCLE WITH ALLOWANCES FOR PUMPING LOSSES AND RESIDUAL GAS.

FIGURE II. EFFECTS OF GAS TEMPERATURE AT THE REACTOR
OUTLET ON THE THERMAL EFFICIENCY AND IMEP
OF THE NUCLEAR GAS ENGINE CYCLE WITH
ALLOWANCES FOR PUMPING LOSSES AND RESIDUAL GAS.

FIG. 12. GAS TEMPERATURE AT BEGINNING OF COMPRESSION - °F

433 039

TYPICAL NUCLEAR GAS ENGINE INSTALLATION

In evaluating the possibilities of the nuclear gas engine it is instructive to consider a power plant installation designed for the N.S. Savannah. This approach makes possible a direct comparison between the nuclear gas engine and a representative pressurized-water reactor power plant. The layout envisioned is presented in Figs. 13, 14, and 15. The working fluid chosen was nitrogen because of its ready availability. The design was prepared on the premise that the plant should be as simple and straightforward as possible with a minimum of novelties other than the nuclear gas engines. The reactor design problem was greatly simplified by choosing one of a series of small reactor cores studied recently in connection with advanced design work for the gas-cooled reactor program. The objective of that work was to reduce the fixed costs associated with the reactor, fuel, and shield through the use of smaller cores and higher system pressures. Data for the major components of the power plant are presented in Table IV.

Reactor Design

In general the reactor design follows closely that of the GCR-2 presented in ORNL-2500. The fuel elements would be the same as those presented in ORNL-2500 except that their length would be 24 in. instead of 40 in. as shown in Fig. 16. The limiting stainless steel capsule temperature would be the same as in the GCR-2; i.e., 1500°F, but the much higher heat transfer coefficient available at 1500 psi permits increasing the gas outlet temperature 100°F. It should be noted that a development program for these fuel elements is well underway at the time of writing in connection with the gas-cooled reactor program. The fuel charging operations would be carried out in essentially the same fashion as indicated in the GCR-2 report, although an additional locking device would be required to hold the fuel element securely in place against the accelerations that might be experienced aboard ship. The control rod drive mechanisms would also differ from those of the GCR-2 in that a rack-and-pinion control rod drive would be used instead of a winch arrangement.

GAS COOLED REACTOR FOR A NUCLEAR GAS ENGINE PLANT
SCALE 1/2"=1'

FIG. 13. SECTION THROUGH THE PROPOSED GAS-COOLED REACTOR

433 042

INBOARD PROFILE

Ref. Maritime Administration Design PZ-M-MA40a

Fig. 14

LAYOUT NUCLEAR GAS ENGINE
IN MARINE CARGO SHIP
Mk. 2

Scale - 1/4" = 1'-0"

Drawn - 8/20/58

By - J. J. Tudor

L-X-4 Rev. A

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Ref. Maritime Administration Design P2-N1-MA40a

Fig. 15

LAYOUT NUCLEAR GAS ENGINE
IN MARINE CARGO SHIP

Mk. 2

Scale ~ 1/4" = 1'-0"

Drawn ~ 8/20/58

L-X-3 Rev. A

TABLE IV

Specifications for Proposed Nuclear Gas Engine Power Plant - NGE-1

Normal Power for Propulsion	20,000 shp
Maximum Power for Propulsion	22,000 shp
Power to Electrical System	1500 kw
Reactor Power (max)	60 Mw (thermal)
Working Fluid	N ₂
Over-all Thermal Efficiency	30 %
Reactor Data:	
Fuel element	UO ₂ in SS capsules
Moderator	graphite, grade TSF
Lattice pitch	8.0 in.
Core diameter	8.5 ft
Core length	10.0 ft
Side reflector thickness	1.5 ft
End reflector thickness	2.0 ft
Fuel channel diameter	2.75 in.
No. fuel elements required	4655 capsules 665 assemblies
No. fuel channels	133
Pressure vessel I.D.	12 ft 8 in.
Pressure vessel O.D.	14 ft
Pressure vessel height	23 ft
Shield O.D.	26 ft
Shield height	32 ft
Engine Data:	
No. of main propulsion engines	4
Bore	15.5 in.
Stroke	22 in.
No. of cylinders	12
rpm	335
Compression ratio	4.8
bhp per engine	5,500
bmepp	130 psi

TABLE IV (continued)

Piston speed	1230 ft/min
Piston displacement (per engine)	49,800 in. ³ (28.9 ft ³)
No. of nuclear gas engine generator sets	1
Bore	15.5 in.
Stroke	22 in.
No. of cylinders	6
bhp	2,500
Gas System Data:	
Total gas system volume	1800 ft ³
high-pressure system	1200 ft ³
reactor	20 ft ³
low-pressure system	600 ft ³
Volume of free gas at STP	50,000 ft ³
Gas flow through reactor at full power	125.5 ft ³ /sec 465 lb/sec
Gas velocity through reactor at full power (mean)	80 ft/sec
Gas velocity through ducts (mean)	75 ft/sec
Gas temperature at reactor inlet	600°F
Gas temperature at reactor outlet	1100°F
Gas temperature at cooler inlet	470°F
Gas temperature at cooler outlet	90°F
Pressure at cylinder discharge port	1,500 psi
Pressure at cylinder return port	1,456 psi
Pressure at cooler inlet	157 psi
Pressure at cooler outlet	152 psi
Pressure at cylinder inlet port	165 psi
Gas density at cylinder discharge port	3.7 lb/ft ³
Gas density at cooler outlet	0.723 lb/ft ³

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Seven control rods would be required, one in each of the fuel charge tubes. The control rod drives would be attached to the flange at the top of these tubes just outside the shield. As in the GCR-2 the actuating mechanism would operate at the system pressure to avoid the need for seals.

The reactor and shield layout has been designed to permit removal of the reactor and pressure vessel as an assembly in the event so drastic an operation might be required. The pressure vessel and reflector are sufficiently thick to provide enough shielding to permit this operation if some limitations are placed on the proximity of personnel to the vicinity of the reactor during the removal operation. Ordinarily all control rod and fuel element changes could be accomplished through the fuel charge tubes at the top of the reactor.

The reactor pressure vessel has been designed to have a uniform thickness of 8 in. over the cylindrical side and the hemispherical ends. This thickness was chosen to satisfy ASME pressure vessel practice for the cylindrical section. While the hemispherical heads might have been made thinner, this was not done partly because reinforcement would have been required around the penetrations at the top, and partly because it was desired to have the pressure vessel serve as the principal gamma shield for the reactor. Preliminary calculations indicate that no additional gamma shielding would be required except possibly some patches in the vicinity of the ducts. These calculations will have to be made more rigorously, but at most another 2 or 3 in. of iron in the form of slabs will suffice to provide adequate gamma shielding. Note that, as in the GCR-2, a layer of Pyrex glass has been provided just inside the pressure vessel so that the low-energy neutrons leaking from the reflector will be absorbed in boron thus inhibiting the production of secondary gammas in the pressure vessel.

The reactor pressure vessel has been designed to have an outer diameter of 14 ft so that it can be fabricated in the shop and shipped by rail. The vessel was designed without a large flanged closure because experience with the Calder Hall reactors has indicated that such a closure is not necessary. This should reduce both the cost and the weight very substantially and should eliminate a serious possible source of leakage.

It should also give a stronger, more reliable pressure vessel. Further, there is a very real question as to whether it would be possible to get effective sealing of the flanged joint in the hot zone if the flange were opened after some years of operation. In the design shown in Fig. 13, the central charge tube has been designed with an 18-in. internal diameter so that men and equipment can enter the vessel for the installation of the graphite, the thermal insulation, the boron curtain, and the lateral support structure. The design anticipates that the support grid for the graphite will be fabricated, stress-relieved, machined, and installed in the pressure vessel before welding the pressure vessel head in place. The vessel as a whole could then be stress-relieved after this weld had been completed, and the installation of the internal thermal insulation, the boron curtain, and the graphite could be carried out.

The graphite would be supported from a grid fabricated with radial webs. This grid would be centered on a large vertical pin attached to the pressure vessel on its centerline, as shown in Fig. 13. Note that two grids are employed, the lower one made integral with the vessel and designed to operate at the pressure vessel temperature, that is around 200°F, and the upper one designed to operate at the gas temperature entering the reactor, that is around 600°F. A layer of thermal insulation is designed to separate the two grids. The graphite blocks were designed in the same fashion as those for the GCR-2 with a central hole for the fuel channel in each 8-in. square block. The blocks surrounding the outer perimeter of the reflector would be machined so that the over-all assembly would have 152 faces. A Pyrex plate backed by thermal insulation and enclosed in sheet steel would bear against each face. These would be held against the graphite by a steel plate which would be supported by two thin-walled steel tubes, as shown in Fig. 17. The tubes were designed to act as springs so that, as the heated graphite expands relative to the cool steel pressure vessel, these tubes will be distorted into ellipses with their minor axes perpendicular to the wall of the pressure vessel. The design is made such that the diametral deflection in the tubes will be elastic, and will give a spring load at operating temperatures sufficient to take a lateral load of 5 g's without inducing any

PRESSURE VESSEL

152 3" O.D. TUBES WITH $\frac{3}{32}$ " WALL
THICKNESS SPACED .100" APART
BY HOOPS RIVETED TO EARS AT
UPPER AND LOWER ENDS

SCALE ~ 1" = 1"

GRAPHITE
BLOCK

$\frac{5}{8}$ " THICK PYREX PLATE
 $\frac{1}{4}$ " THICK STEEL PLATE
 $\frac{7}{8}$ " THICK INSULATING TILE
 $\frac{1}{2}$ " THICK STEEL PLATE

FIG. 17. HORIZONTAL CROSS-SECTION THRU THE REGION BETWEEN THE GRAPHITE REFLECTOR AND THE PRESSURE VESSEL

appreciable additional deflection. A larger load would cause the tubes to spread slightly farther at which point they would touch each other. This would provide mutual stiffening and permit the carrying of a very much higher lateral load without plastic deformation taking place in the tube. During the installation of the graphite it would be necessary to insert shims between these tubes and the pressure vessel wall to eliminate practically all of the clearance so that the tubes will firmly locate the graphite even in the cold condition to take care of loads that might occur if the reactor were moved when cold.

Gas Duct Design

An important feature in the design of the high-pressure gas system was the use of thermal insulation between the high-temperature gas and the steel pressure envelope. This was done to minimize thermal stresses in the pressure envelope, particularly where the high-pressure gas ducts penetrate the engine cylinder heads and the pressure vessel. It also eliminated the need for expansion joints in the ducts between the engines and the reactor. The ducts are designed to be water-cooled by tubes wrapped around their perimeters so that under normal conditions their temperatures will run 100-200°F. In the event of a failure in the water flow to these cooling jackets, thermal convection of the engine room air will keep the duct wall temperature below 300°F. The plastic strains induced by such a temperature cycle will not be serious for the relatively small number of such occurrences to be expected in the life of the plant. Note that, as shown in Fig. 18, concentric ducts have been used with the lower temperature gas flowing outward from the engine in the annulus and the high-temperature gas returning from the reactor through the central pipe. This helps to minimize heat losses, reduces the likelihood of gas leakage from the system, and minimizes the plastic strains associated with a failure in the duct cooling water supply.

While Figs. 17 and 18 indicate a ceramic type of thermal insulation, it might be easier to degas the system prior to startup if a reflective type of thermal insulation were used. Alternate layers of flat and finely corrugated sheet should be effective for this application. Although this would increase the cost, it would greatly reduce the amount

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FIG. 18. CROSS-SECTION THRU HIGH PRESSURE GAS DUCTS

of adsorbed gas -- particularly water vapor -- to be removed. On the other hand, unless special care were taken to prevent moisture absorption by the graphite between its manufacture and the startup, it would be a greater source of adsorbed gas than a ceramic insulator.

Engine Design

A number of large U. S. stationary and marine diesel engines might be modified to function as nuclear gas engines. While the modifications to the cylinder head and valve gear would entail a major redesign, the engine size and weight should be about the same for a given bore and stroke.

The Cooper-Bessemer LSV12 engine was chosen for this study because it seems well suited to the installation and data were conveniently available. These engines are currently built for marine applications with two, three, and four engines connected through electric couplings and a reduction gear to a single propeller shaft. The electrical couplings minimize the transmission of torsional vibration to the gear train and permit easy declutching of an engine from the gear in the event of service difficulties. On the basis of the analysis summarized in Table II, each nuclear engine of this size should deliver approximately 5500 hp. Table IV presents the engine data as estimated for this study.

The breathing problem has been mentioned as being vital to the performance of the engine as it will determine the power required to pump the charge in and out of the cylinder. Probably the most difficult engine design problems in this area lie in the cylinder head. The first and probably the most important premise on which the design was based was that the valve gallery should operate at the peak gas pressure of the system so that a single seal could be used between the low-pressure crankcase and the valve gallery. This would be a rotating seal on the tower shaft driving the cam mechanism. The layout shown in Fig. 19 follows this approach. The cylinder head envisioned would be formed by two long cylinders with a common web in the central 60-deg sextant. This arrangement will give a constant wall thickness for both the web and the drums and will facilitate fabrication of the unit from rolled steel plate. In view of the 100-atmosphere pressure which the cylinders must sustain, and in view of the stringent leak-tightness requirements,

FIG. 19. SECTION THROUGH THE CYLINDER HEAD AND VALVE GALLERY OF THE PROPOSED NUCLEAR GAS ENGINE.

it seems best to use welded plate construction in place of cast steel or aluminum. Each drum could be operated with an internal pressure equal to the pressure in its gas manifold so that slip joints could be employed in the internal manifold to accommodate differential thermal expansion. Gas leakage through these joints would be trivial and would have little effect on the performance of the system.

The valve gear in each head would have its own lubricating oil system with a small bypass bleed flow to the main oil system for cleanup and makeup purposes. The pressure difference between the two lobes of the cylinder head would run approximately 30 psi so that a single lubricating oil pump and system would serve to accommodate both sets of valve gear.

The excellent breathing performance and service records of engines built with four poppet valves in the head of each cylinder made this configuration a logical choice for the preliminary design. Heavy transverse webs on either side of each cylinder would act as reinforcements to carry the hoop stress which would normally be carried by the sector perforated for the valves. These transverse webs would carry the load in combined tension and bending. The maximum stress would be developed at the vertical center line just under the central web of the cylinder head and would appear in the outer fiber of the web. These webs could also be designed to carry the bearings for the camshaft.

Access to the valve gear would be obtained through ports or wells in the form of flanged cylindrical stubs located just above the valves in each cylinder head. Transverse reinforcing webs similar to those on either side of the valve ports would be required on either side of each opening. These stubs would be closed with flanges having a cylindrical pilot fitted with O-rings. The closure could be secured with a split ring type of closure clamp or a gun breech type of closure could be considered. The manifold in the cylinder head could be made with alternate layers of plain and corrugated sheet metal to provide thermal insulation between the hot gas inside the duct and the rest of the cylinder head to avoid the oil adsorption and contamination problems of materials such as MgO. Note that this duct would have extensions off to each valve port to minimize heat losses to the cylinder head in that zone. In assembling the cylinder head, the

duct assembly would have to be inserted first and the port liners piloted into suitable recesses at each valve port. The camshaft with its bearing brackets could then be inserted and the brackets attached to webs in the cylinder head. The valves, valve guides, and the valve spring assemblies could then be inserted through the vertical access ports and secured in place. A bored cylindrical bracket attached to the central web would receive each assembly. Note that the roofs of the port liners would be provided with cylindrical openings through which the poppet valves could pass. These holes would be closed and the port liner held in place by the lower portion of the valve guide assembly. The rocker arm assemblies would then be installed and the outboard ends pinned to brackets provided on the inside of the valve gallery. A pin could then be inserted to connect the rocker arm to the link projecting upward from the top of the poppet valve and guide assembly. The valve clearance would be regulated by adjusting the vertical position of the tappet in the rocker arm. An oil splash pan should be incorporated in the manifold duct assembly to keep oil off the hot manifold and direct its drainage to a central sump.

The poppet valves should be very carefully designed to minimize weight and thus permit as rapid acceleration as possible to minimize the time required for the valve opening and closing events. In view of the very high differential pressures that must be accommodated across the valve head, it seems best to use a valve head spherically dished outward from the cylinder to make the head as inherently strong as possible for a given weight. The edges of the head would be beaded to reduce the pressure drop across the valve seat when the valve is open, and to carry the extra hoop tension induced by the use of a rather flat seat angle, probably an included angle of about 120 deg. While a sharper angle would give lower stresses in the outer rim of the valve head, it would give a lower effective opening rate because the projected area of the opening between the valve head and the seat would increase less rapidly for a given lift curve than with the flatter seat angle during the initial phase of the lift. The valve shown in Fig. 19 has been designed as a multipiece valve in which the tubular stem would be welded or brazed to the forged head and to the valve tip. The valve tip has been designed to incorporate an eye to facilitate linking

it to the rocker arm. This is believed to give a more reliable design than might be obtained with a spherical joint. The use of a simple eye at the tip of the poppet valve stem has the disadvantage that the valves will not be able to rotate. However, the relatively thin flexible head proposed should adapt itself well to the seat so that valve rotation may not be important.

The hollow valve construction should minimize the valve weight and hence the acceleration forces for opening and closing. Note that valve springs have been provided to close the valve. The valve operation near top center need not be at an extremely rapid rate as the piston velocity will be low during that interval. The discharge valve will have to open and the return valve will have to close very rapidly but in both cases the pressure differential across the valve will be in the same direction as the operating force and will assist in operating the valve. The cam will, of course, limit the closing velocity to a reasonable value. Hydraulic tappets might be employed to minimize the lost motion and the need for adjustment, and the much cleaner oil should make them more reliable than they have been in conventional engines. They would also aid in absorbing any shock associated with valve bounce under transient conditions where the discharge valves might blow open. However, the valve gear design proposed here is sufficiently compact so that differential thermal expansion may not present a serious valve clearance problem and hence the simpler and more reliable mechanically adjusted tappets may be best for this purpose.

It is important to minimize the volume in the space between the piston and the poppet valve heads to minimize the dilution effect of the residual gas remaining in the cylinder at top center as discussed previously in the section on cycle analysis. Both cylinder head cooling and cylinder head strength considerations coupled with fabrication problems will make it exceedingly difficult to keep this clearance volume sufficiently small so that the dilution effect of the residual charge will not cause a substantial loss in power output. It is essential to provide cooling for the top of the cylinder head, particularly around the valve seats, but cooling passages in this region tend to weaken the

cylinder head and increase the clearance volume. It may be that this cooling can best be done by making a copper filler block for the cylinder head and brazing it to the drums enclosing the valve gallery and gas manifolds. This copper block would then be cooled by a water jacket around its perimeter and a central passage between the two sets of valve seats, as indicated in Fig. 19. Cooling will also be required in the vicinity of the discharge and return gas ducts, although it may prove possible to get adequate cooling of much of the valve gallery pressure envelope by oil splash from the valve lube oil system.

The breathing problems associated with the cooling phases of the cycle also present some difficult engine design problems. The schematic diagram of a section through the scavenging ports in the cylinder wall in Fig. 20 shows the course of the gas flow through the cylinder, the manifolds, and the cooler. This arrangement envisions the use of a large cylindrical drum just outside the cylinder bank to house the cooler together with the exhaust and intake manifolds. A blower would be located at the end of the drum to draw the gas through the cooler and force it back into the engine cylinders. The cooler consists of 1-in. OD U-tubes wound with 2-in. OD helical aluminum fins 0.015 in. thick. These tubes would project inward toward the center from each end of the drum so that the bent length of each U would be half the length of the drum. The headers at the ends of the drum would be designed to direct the flow in such a way that essentially counterflow heat transfer conditions would prevail. The temperature rise in the sea water would be 15°F at full power. Further data on the cooler are given in Table IX (see pages 84-85).

Note that the loop scavenging arrangement of Fig. 20 envisions the use of intake ports at the sides of the cylinder to admit the gas and direct it in such a way that loop scavenging of the cylinder will be accomplished.

To minimize gas leakage from the system the crank case would be enclosed and maintained at perhaps 1 psi above atmospheric pressure. Gas blowing by the pistons into the case can be compressed and returned to the gas storage system. The shaft seals should not present a serious problem as the pressure differential across them would be small, the

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FIGURE 20. SCHEMATIC SECTION THRU CYLINDER INTAKE AND EXHAUST PORTS AND THE GAS COOLER.

rubbing velocity reasonable, they would be well lubricated, and a small amount of leakage through the seals would not be objectionable.

A considerable amount of auxiliary equipment is required for the engines in the form of the jacket cooling water and lubricating oil systems. These will be essentially similar to the corresponding systems in diesel engine installations. Figure 21 is a flow sheet showing the manner in which this equipment would be connected to the engines.

Nitrogen Supply and Storage System

The principal consideration in the design of the gas system is neutron activation of impurities in the gas passing through the reactor. In the Calder Hall plant this resulted in dose levels of about 75 mr/hr at the ducts leaving the reactor shield. About two-thirds of this was formed by neutron absorptions in the oxygen of the CO_2 . The remaining one-third came mainly from A^{41} . The argon concentration in the CO_2 ran approximately 6 ppm. Pure nitrogen would give activities in the gas stream outside the reactor shield only 1 or 2% of the values given by pure CO_2 , so that activation of the nitrogen itself does not present a problem. Commercial nitrogen, however, usually contains from 40 to 100 ppm of argon, and this argon will be the principal source of activity to be expected in the gas stream other than the possibility of fission products which might escape from leaks in the fuel capsules. Thus to avoid serious problems in routine operation and maintenance the argon concentration in the nitrogen should be kept below 10 ppm. While fission product activity could be much greater, it would not be unreasonable to tolerate the inconvenience that it would cause for a short time since the occasions would be quite rare.

In this connection it is important to stress the difference between the radiation doses tolerable for routine operation and those tolerable under relatively rare emergency conditions. The former of necessity must be about 0.1% of the latter. Since the incidence of fuel capsule leaks should be quite low, and since the proposed reactor design is such that the leaking fuel capsule can be removed readily by removing the five fuel elements in the offending channel, the estimated dose level of about 0.4 r/hr at the engines when a fuel capsule leaks should not represent a serious handicap. Engine operation could be continued until the vessel reached port. There is no apparent way in which the trace amounts of

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Legend
 Lube Oil Lines
 Raw Water Lines
 Jacket Water Lines

FIG. 21
 FLOW DIAGRAM
 COOLING SYSTEM & LUBE OIL SYSTEM
 FOR NUCLEAR GAS ENGINE POWERED
 MARINE CARGO SHIP
 Drawn - 9/1/58 By - J.J. Tudor Drawn - Na-L-X-6

fission product gases from the leaking fuel capsule could induce difficulties with the engines, but, in the unlikely event that engine trouble did arise from other factors during the trip to port, the gas system could be emptied to get rid of most of the activity at the engines, and repairs could then be made. While the inconvenience that this would entail could be tolerated on rare occasions, there is a strong incentive to keep the activity in the gas system sufficiently low so that the dose level under normal conditions will be below 6 mr/hr for all points outside of the reactor shield.

In view of the low activity level desired, the source of nitrogen best suited to this application appears to be a small nitrogen generator that makes use of refrigeration and fractional distillation, i.e., the same process as is used in large commercial air reduction plants. Unfortunately, good data are not available on the minimum argon content obtainable from such small plants, but the manufacturer of one promising piece of equipment believes that it would be possible to hold the argon concentration to the desired 10 ppm by reducing the through-put of a 1500 ft³/hr unit costing \$25,000 or by reprocessing the output a second time. This unit was chosen for the proposed installation with the thought that, if the press of circumstances required, it could be operated at its normal capacity to provide a system loading in two days. While the argon content of this loading would run about 100 ppm, it would serve temporarily, and the nitrogen generator could then be operated at reduced capacity to provide a second loading of much lower argon content in less than a week.

To provide a substantial degree of operating flexibility, it was decided to make use of nitrogen storage tanks having a capacity sufficient to hold 1.1 system volumes at 250 psi. These tanks could be the same as the standard air receivers used for starting in diesel engine installations. The system capacity was chosen so that it could accommodate many different situations. Under normal conditions it would permit operation of the reactor for 100 days with a leakage of 1% per day (i.e., about the level to be expected) with no resupply from the nitrogen generator. In practice, even if the supply dropped to half of the capacity and all of the nitrogen was lost from the reactor system, there would still be enough left in the storage tanks to recharge the reactor gas system to at least half of

its normal pressure. This would permit operation of the engines at half power. In the event of an emergency in which it was necessary to depressurize the reactor gas system and retain the gas discharged because of contamination, it would be possible to empty enough of the nitrogen storage tanks so that the bulk of the reactor gas system could be blown down into these tanks and the balance could then be pumped in. The gas could be retained in the tanks until the fission product activity had dropped to a sufficiently low level so that it could be released, or other provisions made for its disposal. For example, it could be vented at sea under favorable wind and weather conditions. A suitable stack would of course have to be provided for this purpose in the form of a pipe perhaps 2 in. in diameter.

An aluminum oxide dryer has been included in the nitrogen system to remove excess moisture. This is important because otherwise the water gas reaction over a period of years would result in a serious loss of weight in the graphite moderator. The projected system also includes a vacuum pump for evacuating the reactor system and the storage tanks prior to filling, as well as the related valves and equipment.

The principal use of the vacuum pump and drying equipment would occur during the initial startup when it would be necessary to degas the graphite and any ceramic thermal insulation exposed to the gas in the system.

Power Plant Operation and Control

The pressure level at each of several key points in the gas system will be controlled by a number of valves. Under normal conditions the principal pressure level control will be in the low-pressure cooling loop. A portion of the nitrogen storage tanks will be maintained at a pressure somewhat below the pressure in the low-pressure cooling loop while others will be maintained at a pressure somewhat above. If the pressure in the cooling loop drops too low, a valve will open and allow nitrogen from the higher pressure storage tanks to feed into the system, while if the pressure rises above the value chosen a relief valve will open and allow gas to blow down to the low-pressure nitrogen storage tanks. One of the nitrogen compressors will operate continuously to scavenge nitrogen from the crankcases and deliver it to the higher pressure nitrogen storage tanks and in this way the gas blown by the pistons

and rings can be scavenged to maintain the crankcase pressures approximately 1.0 psi above atmospheric pressure.

The pressure in the high-pressure portion of the gas system will be controlled by the effective compression ratio of the engines as established by the camshaft setting. The camshafts will be set so that the discharge valves in the cylinder heads will be opened at a pressure of approximately 1500 psi under normal conditions. A blowoff valve in the high-pressure system will be provided to prevent the system pressure from exceeding a selected value (e.g., 1,525 psia). If, through any circumstances, the camshaft fails to open the valves before the cylinder pressure exceeds approximately 1,515 psi, the valve springs will be sufficiently light so that the discharge valves will blow open. This will prove particularly important during fast reversal operations when the camshaft timing may be changed in fashions which otherwise might lead to excessive cylinder pressures during the transient. Stop valves will be provided at the gas-outlet end of the cylinder head to permit isolation of an engine from the rest of the gas system. In addition, a bypass valve between the gas-discharge and gas-return lines should be provided at each cylinder head to provide control of the temperature of the gas entering the cylinders from the return line and hence of the engine power output. This gas bypass valve should be operated in conjunction with control of the camshaft timing for the gas-return valves. The effect of this method of control can be seen readily by examining the indicator diagram in Fig. 9. As gas is allowed to bypass the reactor directly from the discharge to the return ports the temperatures of the return gas will drop so that the volume available per cycle will be less. The camshaft, therefore, will be advanced so that the valve cutoff point will occur earlier in the cycle and hence the indicator diagram will have a smaller net area. If a rapid deceleration is desired, this manner of control can be pushed to the point where the power input from the crankshaft to the gas on compression will actually exceed the power obtained from expansion so that exceptionally rapid deceleration could be achieved. Of course, if this were done, the volume of gas delivered to the high-pressure system by the engine would exceed the volume consumed by the

engine, and the high-pressure gas system blowoff valves would have to dump substantial quantities of gas to the coolers.

For startup following a shutdown in which the system had been open to the atmosphere it will be necessary to remove the bulk of the argon and oxygen in the system. This can best be done by evacuating the entire gas system including the engines. It will probably be necessary to make provisions for a temporary rubber dam where the crankshaft emerges from the crankcase because the usual shaft seal at that point will probably not be adequate to make it possible to reduce the pressure in the crankcase to a low level. Any other shafts projecting from the engine crankcase will have similar provisions. These can be in the form of discs close together, one of which would be integral with the shaft and the other integral with the crankcase. A strip of tape tightly drawn over the perimeters of these two discs will give a good closure for the evacuation. The gas system pressure should be reduced to as low a value as the vapor pressure of the lubricating oil will permit, probably less than 1 mm Hg. This will reduce the oxygen concentration to a level such that the loss in graphite weight for the many starts in the life of the reactor will be completely trivial. It will also reduce the argon concentration to a level lower than that of the nitrogen supply. The system can then be filled to atmospheric pressure with nitrogen and the high-pressure portion of the system filled to approximately 150 psi.

The first step in bringing the reactor critical will be to circulate gas by motoring the auxiliary engine coupled to the electric generator at a low speed using power from the diesel generator. The generator could be built with windings so that it could be operated as a motor or a separate motor could be coupled to it coaxially. The discharge camshaft timing should be set to give the minimum compression ratio obtainable and in that way minimize the power consumption. The power put into the gas by motoring the engine would warm up the system slowly. The reactor should be brought to criticality by gradual withdrawal of the control rods, and then the reactor power should be increased gradually. The auxiliary engine would begin to run under its own power when the reactor gas outlet temperature reached around 750°F. More nitrogen should then be admitted to the reactor system to raise the pressure. The speed of the

auxiliary engine should be increased with further increases in reactor gas outlet temperature, the generator coupled into the circuit with the diesel, and the load shifted to the nuclear gas engine. The reactor gas outlet temperature and pressure should then be up to the design values.

Before starting up the main propulsion engines the bypass valves at the engines should be opened a little to allow gas at normal compressor outlet temperature to bypass the reactor and thus warm up the ducts. The block valves to one of the engines can then be opened and the engine started.

The engine ought to be warmed up with no load, i.e., with the electric coupling to the reduction gear disengaged. After the engines had been warmed, they could be loaded by operation of the electric couplings, or the propulsion engines could be stopped, the couplings engaged, and the engines restarted under load.

Once the gas system has been brought to something approaching design conditions the engines should start easily.

With engines of eight or more cylinders, it will probably suffice to open the block valves at the ends of the cylinder heads and the engine should start by itself because there will be enough cylinders so that the net torque on the crankshaft will be positive irrespective of the crank position. Engines of less than eight cylinders will probably have to be equipped with a conventional diesel engine air-starting valve system.

The steps in the startup can be summarized as follows:

- 1) Evacuate gas system and fill with N_2 to ten atmospheres.
- 2) Circulate gas through the reactor by motoring the auxiliary engine.
- 3) Withdraw the control rods to bring the reactor critical and then gradually increase the reactor power until the gas outlet temperature is about 750°F.
- 4) When the gas temperature is sufficiently high so that the auxiliary engine will operate at full speed under its own power it should be coupled into the line and the electrical load shifted from the diesel to the nuclear engine.
- 5) Nitrogen should be added to the gas system to bring the pressures up to design values.

- 6) The main propulsion engines should be started up and warmed up gradually with no load by opening first the bypass valves and then the stop valves at the engines.
- 7) The main propulsion engines can then be coupled to the reduction gear and propeller.
- 8) At low loads the camshaft for the gas return valves will be timed so that the return valves will close relatively early in the expansion stroke, and the bypass valve will permit a substantial amount of gas to bypass the reactor.
- 9) To increase the engine power the return camshaft timing should be retarded so that the gas return valves will close later in the expansion stroke. At the same time through a suitable linkage the opening of the bypass valve will be reduced. This linkage could be designed so that it would tend to give a small excess delivery of gas from the engine to the reactor at low powers. This excess would be handled automatically by the pressure relief valves in the high-pressure circuit, and would be blown down to the coolers.
- 10) The process outlined in (9) would be reversed to reduce the engine power output.

Reversal of the engines presents special problems. The first step in the reversal will be to decelerate the engine by advancing the timing of the return camshaft with the effects as indicated above. As the crankshaft comes to rest it will be necessary to reverse the timing of the discharge and return valve camshafts. The engine will then start to operate in the reverse direction. A sequence of events that seems reasonable thus is as follows:

- 1) Reduce power by changing the camshaft timing to advance the return valve closing event.
- 2) The control rods will automatically move in to reduce the reactor power and hold the reactor gas outlet temperature constant.
- 3) When the engine comes to a stop the discharge and return camshaft timing should be changed to reverse the direction of rotation.
- 4) The engine power can then be increased as indicated above.

It should be noted that the very large reservoir of high-

pressure gas in the reactor pressure vessel and duct system will be many times that used for conventional diesel engine starting operations. The time required for the reversal operation will probably run less than 30 sec. It might be as little as the 4 sec with which complete reversal can be accomplished with a reciprocating steam engine.

The principal problem involved during the reversal operation will probably be that of reactor temperature control. A preliminary estimate indicates that the negative temperature coefficient associated with the fuel will be sufficient to prevent a temperature overshoot in the fuel of more than about 100°F if the period during which the engines are completely stopped does not exceed about 5 sec. If an auxiliary engine is operating and driving an electric generator carrying at least 50% of its rated load there should be sufficient gas flow through the reactor to provide adequate cooling at all times.

Hazards Considerations

In general, the hazards considerations for this reactor are essentially similar to those for the GCR-2 except that the power density in the fuel elements would be approximately doubled. The principal effect of this latter factor will be to increase the temperature overshoot associated with an abrupt drop in the gas coolant flow rate. Another important difference is that, as laid out, the small core reactor will not cool by natural convection of the cooling gas. Several arrangements are being examined to provide for the removal of afterheat in the event of the abrupt stoppage of all five engines after extended operation at full power. The simplest of these approaches would be to reduce the thickness of the thermal insulation around the inside of the reactor pressure vessel sufficiently to permit the normal heat losses to the shield to run 2% of the full-power output of the reactor instead of 0.5%. This would be a crude but an effective means of providing for afterheat removal. Two other somewhat more complicated arrangements are under consideration which would not entail this substantial heat loss.

It is believed desirable to enclose the reactor and engine compartments so that, in the event of a fuel element failure, leakage of

radioactive material from the high-pressure gas system at the engine will be confined in the engine compartment. This is believed necessary because it appears difficult to make the engines absolutely leak-tight. While the activity that might escape in this way would normally be trivial, it might prove serious in the event of a major failure involving many fuel elements. The probability of such a failure is so small that the Calder Hall plants which leak from one to three tons of CO_2 (or 4% to 12% of the gas charge) per day per reactor have no provisions to prevent the dispersal of activity by this means in the event of a major fuel element failure. It should also be noted that the stainless steel and UO_2 fuel elements in this proposed nuclear power plant are much less likely to lead to a major fuel element failure than the CO_2 -magnesium-metallic uranium combination employed in the Calder Hall plants. While the probability of such an event is certainly very low it seems prudent to provide additional containment, particularly since in this case the incremental cost should not be great because the containment envelope need not withstand much pressure.

The total amount of high-pressure gas contained in the system is such that, in the event of a break in the pressure envelope, the increase in gas pressure in the engine compartment will run only about 9 psi. As can be seen from Table V, such a failure is very much less serious than it would be in a system containing large amounts of superheated water which would flash into steam and create a very serious explosion hazard.

TABLE V

Summary of Factors Affecting Design for Containment

	<u>N.S. Savannah</u>	<u>Nuclear Gas Engine</u>
Reactor cooling system pressure - psia	1,750	1,500
Free volume of gas released by rupture of reactor coolant system - ft^3	600,000	130,000
Volume of containment enclosure - ft^3	32,300	210,000
Pressure in containment enclosure following reactor cooling system rupture - psig	186	9

The relatively small volume of gas released in the event of a rupture in the reactor coolant system makes it possible to utilize the hull of the ship as the basic containment vessel. The additional special provisions for containment would consist mainly of two double-door air locks for personnel and two large pressure-tight hatches, one for the reactor compartment and one for the engine room. In addition, the hull, bulkheads, and the deck closing the top of the compartment should be made leak-tight by welding or calking. The engine room and reactor compartment ventilation systems would also require valves which would be closed automatically by monitrons if the activity level in the engine room exceeds the tolerance value.

An important consideration is that the reactor be designed so that no serious consequences will result if the ship were to sink in a harbor. Thus it is essential that the reactor be designed so that its reactivity will be reduced if it were flooded. While calculations have not been completed for the reactor core geometry proposed in Table IV, it is clear that if this is not satisfactory as laid out, the lattice pitch could be increased somewhat to insure a loss of reactivity upon flooding. This would entail a small increase in reactor core diameter, 1 ft at the most.

Cost and Weight of Power Plant Components

The more important items of equipment required for the proposed power plant are listed in Table VI together with the weight and estimated cost of each. In general, the costs were prepared using the same techniques, and, where applicable, the same basic data as were used for the GCR-2 reactor described in ORNL-2500.

The reactor vessel cost was based mainly on weight with allowance for design complexities. The material of construction was considered to be A-212 carbon steel. The estimate includes a large contingency factor.

The graphite moderator costs were derived from ORNL-2500 data which are believed to provide a sound and conservative estimating basis.

The costs of the control rods and drives have been derived from APPR data since it employs rod drives that are similar in design. The APPR rods have recently been quoted by Alco at \$15,000 per unit so that

TABLE VI

Equipment for the Proposed Nuclear Gas Engine Power Plant

<u>Item</u>	<u>Number Required</u>	<u>Description</u>	<u>Weight 1000 lb</u>	<u>Cost \$</u>
Reactor				
Pressure vessel	1	14' dia x 23' high x 8" thick; A-212 steel	320	960,000
Internal structure		A-212 carbon steel	12	65,000
Moderator		TSF - graphite	146	220,000
Control rods and drives	7	Alco units for APPR	4	120,000
Nuclear instrumentation		Counters, fuel capsule leak detection, etc.		300,000
Fuel fabrication		Modified GCR-2 fuel elements	13	300,000
Subtotal			495	1,965,000
Shielding and Containment				
Reactor shield tank	1	26" OD x 32' high x 3/8" thick carbon steel	40	20,000
Containment		2 personnel air locks and 2 large hatches	100	100,000
Fuel handling equipment		1 fuel charging machine with casks and fixtures	50	500,000
Subtotal			190	620,000
Gas System				
Ducts, valves, etc.		Carbon steel with stainless liners in hot zones	80	106,000
Vacuum pump	1	National Research Corp. NRC-400-5, 400 cfm	5.7	4,000
Compressors	3	Quincy D-325 5-hp, 23-cffa/min	1.3	2,200
N ₂ storage tanks	40	30" dia x 20' long air receivers	108	64,000
N ₂ generator, dryer, etc.	1	Joy Mfg., 1500 ft ³ /hr	10	25,000
Subtotal			205	201,200
N ₂ Coolers	9	Carbon steel shell, 70-30 Cu-Ni tubes with Al fins; 5040 ft of 20-ft tubes, 1" OD, 0.049" wall	42	105,000

TABLE VI (continued)

<u>Item</u>	<u>Number Required</u>	<u>Description</u>	<u>Weight 1000 lb</u>	<u>Cost \$</u>
Main Propulsion Engines	4	Same frame as Cooper-Bessemer LSV12	656	1,554,000
Reduction gear	1	Western Gear 3.0:1	70	455,000
Couplings	4	Elliott electric	102	284,000
Subtotal			828	2,293,000
Jacket Cooling Water System				
Jacket water cooler	4	Ross No. 1708 CP, 17" dia x 9'6" long	11	22,000
Raw water pump	6	Allis-Chalmers 5 x 4 SSL - 10 hp each	4	6,000
Fresh water pump	6	GD9-3767-463, 10 hp each	3	12,000
Temperature regulating valve	4	GD9-3768-155 G	1	4,100
N ₂ cooler water pumps	6	20 hp each	8	20,000
Lubricating Oil System				
Standby pumps	2	GD9-3767-251 4, 40 hp each	4	10,000
Cooler	4	Ross 1214 CP, 12" dia x 16' long	8	14,800
Strainer	4	Duplex with magnets, GC9-4188-32 1	6	9,600
Filter	4	GD9-3909-249 1, 32" dia x 54" high	3	7,400
Sump tanks	4	1000 gal each	6	4,200
Temperature regulating valve	4		0.6	2,800
Subtotal			55	113,000
Auxiliary Electric Power Plant				
1500-kw generator and diesel engine	1	Same engine frame as Cooper-Bessemer LS6 (complete with water and oil systems)	320	300,000
750-kw generator and diesel	1	Complete with auxiliaries	160	100,000
Electric switch gear		Complete with controls	8	40,000
Subtotal			488	440,000
Total Plant Components			2,276	5,737,000
Engineering, Installation, and Contingency				3,203,000
TOTAL PLANT				8,940,000

the estimate of \$120,000 for seven rods seems to be in order. The cost is naturally higher than that given for the rods for the GCR-2 because they were designed on a winch-cable principle.

A value of \$300,000 was assigned to the nuclear instrumentation. The comparable figure for the GCR-2 was \$500,000 exclusive of the burst slug detector system. Since there will be fewer control rods than in the GCR-2 and fewer temperatures to monitor as well as no blower controls, the reduction seems justifiable. A burst slug detection system will be necessary, but, in view of the small number of channels to be monitored, the expense should not be major.

The fuel fabrication costs for the Savannah and the nuclear gas engine are discussed in a later section on fuel costs. The values given for the gas turbines are from the referenced reports.

The shielding in this case is basically simple particularly since the design is such that the reflector and pressure vessel also serve as the gamma shield. The cost estimates were based on the quantity of steel required for the water tank.

It was assumed in estimating the costs chargeable to containment that in this design the reactor and propulsion machinery will be installed in compartments that will be separated from the remainder of the vessel so that the incremental cost of containment would be mainly that of the two personnel air locks and the two sealed hatches. There would also be an additional cost of about \$30,000 associated with leak-testing and sealing. The compartment walls would be part of the ship's structure and would be fabricated with conventional bulkheads. These will be adequate to confine any contaminated gas that might escape to the atmosphere surrounding the reactor and the propulsion machinery in the event of an accident.

The piping and valve system for the gas coolant has been estimated by using conventional piping costs and estimating methods comparable to those for the GCR-2. The principal uncertainty in this component is the duct length. If the length shown in the conceptual drawings is essentially correct the cost estimate should be reliable. It gives a weight of 40 tons and a cost of \$106,000.

Fuel handling could be a shore-based operation with one handling facility to service ten vessels. The equipment should be fabricable for \$500,000 and proration over ten vessels would give \$50,000 per unit. However, in this case the entire cost was charged off against a single ship.

The cost estimate for the nitrogen system has been developed on the assumption that nitrogen will be produced by liquefaction and fractional distillation. The moisture that might enter the system can be absorbed from the nitrogen in a dryer and filter arrangement, all of which are conventional equipment. Nitrogen compressor facilities for charging and discharging of the reactor are also included in the estimate which amounts to \$31,000 for the nitrogen generator, the dryer, filter, and the compressor system. An additional \$64,000 is included for nitrogen storage.

The cost estimate for the gas coolers was based on Wolverine Tru-Fin U-bend shell-and-tube heat exchangers weighing 21 tons. The tubes are bimetallic consisting of a cupro-nickel inner tube with an outer aluminum fin surface and are available as a conventional product of the Wolverine Tube Division, Calumet-Hecla Copper, Inc.

An estimate of \$440,000 has been established for the necessary electrical equipment to be included with the nuclear gas engine. The cost included a 1500-kw 480-v generator connected to one nuclear gas engine and a 750-kw diesel generator. Costs also include a 9-unit, 480-v switchboard; 100-kva, 480/120-208-v transformer; a 20-unit control center; 50-kw battery charging unit, 480/250-v capacity; a 250-v battery unit; 25-kw dca controlled generator, and the necessary conduit and cable for the installation.

The cost of the engines was estimated by doubling the cost of production diesel engines of the same size. This would not, of course, include the cost of the basic design and development work. The cost of the reduction gear and engine auxiliaries was taken as being the same as that for a diesel engine installation of the same power and speed.

Comparison of Cost and Weight Data for Representative Power Plants

A measure of the validity of the cost estimates for the nuclear gas engine can be gained through comparing these costs with similar data

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for other types of plant. This approach is also helpful in that it will serve to point up the relative advantages and disadvantages of each type from the cost standpoint. A comparison of the cost and weight of the major components for several different types of maritime nuclear power plant has been prepared and is presented in Tables VII and VIII. The data for the gas turbine plants were taken from preliminary design reports prepared by the Ford Instrument Company and General Atomic. The data for the conventional oil-fired steam plant are a bit old so that the costs are low, but they were the best available. No good detailed breakdown of prices is available for the Savannah. However, good weight and dimensional data for the Savannah components were available and were tabulated. The over-all contract prices for the major portion of the work involved a fixed price contract of \$9,872,000 to Babcock & Wilcox for the propulsion plant. Of this, DeLaval under a \$2,500,000 subcontract will furnish the turbine, condensers, feedwater heaters, reduction gear, the steam turbine generator units, and the lubricating oil systems. The New York Ship Building Corporation is responsible for design and installation of the containment vessel, the shielding around the containment vessel, the propeller, and the shafting. The contract price for this work is \$6,685,000. The cost of the components for the Savannah is not given in Tables VII and VIII, but it was estimated using the same techniques as were applied to the nuclear gas engine. This led to over-all costs that were consistent with the contract prices given above if, as in the nuclear gas engine, no allowances were included for engineering development costs (particularly for the fuel element). These costs were not included here because it was felt that they might be misconstrued, but sufficient data on the weight and size of the major components are included to make a similar cost estimate fairly easy.

Because of the importance of heat exchanger costs, an additional compilation of detailed data on the heat exchangers for these power plants has been prepared and is presented in Table IX. Unfortunately, the data in this table are not complete, but a majority of the more important factors affecting the cost of the heat exchangers are included and make possible some instructive comparisons.

TABLE VII

Weight and Cost Data for the Nuclear Portion of Typical Maritime Power Plants of 20,000 shp

	N.S. Savannah Pressurized-Water Reactor and Steam Turbine			Ford Instrument Company FICO No. 103 6-10-57 (unclassified) Helium-Cooled Reactor and Gas Turbine			General Atomic GA-87 4-8-57 (unclassified) CO ₂ -Cooled Reactor and Gas Turbine			Proposed Gas-Cooled Reactor and Nuclear Gas Engine		
	Description	Weight x 1000 lb	Cost x \$1000	Description	Weight x 1000 lb	Cost x \$1000	Description	Weight x 1000 lb	Cost x \$1000	Description	Weight x 1000 lb	Cost x \$1000
<u>Reactor</u>												
Pressure vessel	9'4"x27'6- 1/2" 6 SH clad of 304	285		11'4"x21'5" 316 SS	246	364	8'2"x15'8" Mn Mo SH clad	120	995	carbon steel	320	960
Internal & external structure		100			92	310		80			12	65
Moderator	water	60		C density: 20	40	300	ZrH & C	8.2	220		146	220
Control rods and drives	21 rods & drives	17		7 rods & drives		350	25 to 37 rods		1000	7 rods & drives	4	120
Nuclear instrumentation						436			250			300
Fuel fabrication	UO ₂ in SS cans	16		UO ₂ in SS cans	6		sintered UO ₂ SS plates	2	150	UO ₂ in SS cans	13	300
Subtotal		529			384			914			495	1965
<u>Shielding and Containment</u>												
Reactor shield	water T & Pb	130				175		1140	115	water & steel	1000 40	20
Secondary shield	Pb, concrete polyethylene	1322 1655 107										
Fuel handling equipment						170					50	500
Miscellaneous aux. equipment												

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TABLE VII (continued)

	Description	Weight x 1000 lb	Cost x \$1000	Description	Weight x 1000 lb	Cost x \$1000	Description	Weight x 1000 lb	Cost x \$1000	Description	Weight x 1000 lb	Cost x \$1000
Containment	35'x50.5'x 2-1/2" WT	242			bolted hatch compartment		Not specified				100	100
Provision for afterheat						75	Pb-Bi system		100			
Subtotal		3456									1190	620
<u>Primary Fluid System</u>												
Valves		34		see working fluid system			see working fluid system			see working fluid system		
Piping		36		same			same			same		
Pumps	1100 hp	30		same			same			same		
Storage tanks				same			same			same		
Fluid in system												
Purification equipment	ion ex- change	36										
Subtotal		136										
<u>Total Nuclear Plant Components</u>		4121	6759*			2450			2830		1685	2585

*Estimated at ORNL

TABLE VIII

Weight and Cost Data for the Propulsion Machinery and Over-All Plant for Typical Maritime Power Plants of 20,000 shp

	N.S. Savannah PWR B&W Pressurized Water Reactor and Steam Turbine			Ford Instrument Company FIC No. 103 6-10-57 (unclassified) Helium-Cooled Reactor and Gas Turbine			General Atomic GA-87 4-8-57 (unclassified) CO ₂ -Cooled Reactor and Gas Turbine			Proposed Gas-Cooled Reactor and Nuclear Gas Engine			Typical Conventional Tanker Oil-Fired Steam Boiler and Turbine *		
	Description	Weight x 1000 lb	Cost x \$1000	Description	Weight x 1000 lb	Cost x \$1000	Description	Weight x 1000 lb	Cost x \$1000	Description	Weight x 1000 lb	Cost x \$1000	Description	Weight x 1000 lb	Cost x \$1000
<u>Working Fluid System</u>															
Piping, valves, etc.					80	500			473	carbon steel SS in hot zone	81	106		170	150
Pumps										1-400 cfm 3-23 cfm	7	6		78	30
Storage tanks					182					45,000 ft ³	108	64	feed water	82	140
System fluid	steam			helium	1.5		CO ₂			N ₂	4	2	steam		
Purification equip.		3-4								N ₂ generator CO ₂ abs. dryer	10	25	misc.	224	
Subtotal					264				900		210	203		554	320
<u>Heat Exchangers</u>															
Heaters	steam generator	129.6		recuperator	304								boiler	484	375
Regenerators	feed water heaters	38.4		heat ex- changer	250								feed water heaters	32	50
Coolers	condensers	140		precooler	73					gas coolers	42	105	condensers	136	40
Subtotal		308			637						42	105		652	465
<u>Prime Mover and Aux.</u>															
Prime mover	steam turbine			turbine compressor	135			74	2580	4 12-cyl engines	758	1838	steam turbine	120	1370
Reduction gear				54:1	175		60:1			3.6:1	70	455		220	300
Lube oil system											28	50		60	40
Auxiliaries										jacket water system	28	63			
Propeller & shafting	fixed pitch			controllable pitch	prop. only 132		controllable pitch		prop. only 98	fixed pitch	240	250		240	250
Subtotal					442						1123	2656		640	1960

* Data from FICO No. 103.

TABLE VIII (continued)

	Description	Weight x 1000 lb	Cost x \$1000	Description	Weight x 1000 lb	Cost x \$1000	Description	Weight x 1000 lb	Cost x \$1000	Description	Weight x 1000 lb	Cost x \$1000	Description	Weight x 1000 lb	Cost x \$1000
Auxiliary Electric Power															
Electric generators	2 1500-kw steam, 2 750-kw diesel							535	1 1500-kw NGE, 1 750-kw diesel	320 160	300 100				250
Switchgear & controls							180			8	40				200
Aux. power source	750-hp elec.			diesel	26		1100	1710	750-hp motor	12	25			174	
Miscellaneous													stacks & fan	60	170
Subtotal										500	455			234	620
Total Propulsion Mach. Components		2580			1140	4150		6300		1875	3419				
TOTAL NUCLEAR PLANT COMP.		4121				2450		2830		1685	2585				
TOTAL FOR COMPONENTS						6600		9130			6004				
DESIGN, INSTALLATION, & CONTINGENCY	60% of comp		6198*	48.3% of comp		3185	57.8% of comp	5280	56% of comp		3360				
TOTAL POWER PLANT		8180 dry	15,557		3066	9785		3600	14,410	3560	9364			2080 dry	3335
FUEL INVENTORY	4.1% enrich	7038 Kg	3880							3% enrich	5700 Kg	2136			

*Estimated at ORNL

TABLE IX

Heat Exchanger Data for 20,000-shp Maritime Nuclear Power Plants

Item	Heat Source			Regenerator			Cooler						
	Savannah Boiler	Oil-Fired Plant Boiler	Savannah Feed-Water Htrs.	Ford Instr. Recuperator	Gen. Atomic Regenerator	Oil-Fired Plant Feed-Water Htrs.	Savannah Conden.	Ford Instr. Precooler	Ford Instr. Heat Exchg.	Gen. Atomic Precooler	Gen. Atomic Intercooler	NGE Conden.	Oil-Fired Plant Conden.
Working fluid peak temp. $^{\circ}\text{F}$	465	850		1150	1300								
Hot fluid in - $^{\circ}\text{F}$	520			835	909	206	92	349		269	285	420	91
Hot fluid out - $^{\circ}\text{F}$	497			349	269	206		90	75	100	100	90	91
Hot fluid pres. - psia	1750			303	292	13	0.6	300		290		155	0.7
Hot fluid heat trans. coef., h, Btu/hr-ft 2 - $^{\circ}\text{F}$												150	
Hot fluid flow rate - ft 3 /sec							32,800	796		146			16,000
Mean hot fluid dens. lb/ft 3			3-stage				0.00156	0.165		1.855			
Cool fluid in - $^{\circ}\text{F}$	347	281	85	295	236	120	75	75	60	80		75	75
Cool fluid out - $^{\circ}\text{F}$	465	850	347	781	777	196	84		75	90		85	85.5
Cool fluid pres. - psia	490	600	1850	600	2000	115							
Cool fluid heat trans. coef., h, Btu/hr.													
Cool fluid flow rate	238,000 lb/hr	180,000 lb/hr				180,000 lb/hr	36,000 gpm	6100 gpm					25,000 gpm
LMTD	45			54	72		11.7			93.5		64	
Surface area - ft 2 (mean)	8100	16,700			13,230	650	25,000			4920	6550	15,000	19,500
Heat load - Btu/hrx10 $^{-6}$	214				81.8		156			39.6	56.2	96.3	
Heat loading - Btu/hr-ft 2 - $^{\circ}\text{F}$ x10 $^{-6}$	586				85.8		530			86.1		100	
Shell material	304 SS clad SAE 212	P.C. Steel				P.C. Steel						Steel	Steel
Tube material	304 SS	P.C. Steel				Admiralty	70-30 Cu-Ni			Cu-Ni	Cu-Ni	Al fins on Cu-Ni	Al Brass

Item	Heat Source		Regenerator				Cooler						
	Savannah Boiler	Oil-Fired Plant Boiler	Savannah Feed-Water Htrs.	Ford Instr. Recuperator	Gen. Atomic Regenerator	Oil-Fired Plant Feed-Water Htrs.	Savannah Conden.	Ford Instr. Precooler	Ford Instr. Heat Exchg.	Gen. Atomic Precooler	Gen. Atomic Intercooler	NGE Conden.	Oil Fired Plant Conden.
Tube OD - in.	3/4	1.25			3/8	3/4	3/4			5/8	5/8	1	3/4
Tube wall - in.	0.072	0.109			0.035	0.049	0.049			18 BWG 0.049	0.049	0.049	0.049
Tube length - ft	25.4				40	9	20			19	24	20	14
Total footage of tubing - ft	41,200	(equiv) 51,000			288,000	3,000	127,200			38,000	48,000	3040	100,000
No. of tubes	1624				7,200	170	6,360			2,000	2,000	252	7,400
Total wt - lb (dry) (wet)	79,600	310,000 332,000		304,000		5,260	99,500 142,500	73,000	60,000			84,600	120,000
Wt of shell & headers - lb	58,160	241,000				4,160	53,000					70,000	56,000
Wt of tubes - lb	21,500	69,000			63,600	1,100	46,500			11,450	14,500	14,600	44,000
Total cost - \$1000												105	
Cost - \$/ft ²												7.00	
Cost - \$/lb												2.50	
No. of units	2	1			12		1 large 1 small			1	1	9	
Shape and size	U-tube 40"x40'	15'x21'x28'			Cyl 27"x20'	Cyl 21"x12'	30'x40'x 12'			Cyl 40"x19'	Cyl 40"x24'	Cyl 18"x20'	19'x11'x14'

In analyzing the cost data of Tables VII and VIII, it is helpful to look at the subtotals for the major components. For example, the cost of the reactor in each case appears to be consistent. The pressure vessels in all cases weigh about the same and differ mainly in the complexity of detail designs and in the materials of construction. Note that the pressure vessel for the nuclear gas engine is made of cheaper steel than the others and does not require a large closure. Thus it involves a substantially less complex design than the other three, so that definitely the estimate for the ORNL design is high relative to the others. Similarly, the price of the control rods for the nuclear gas engine reactor should be substantially less than that for the others because the amount of excess reactivity required at startup is smaller and hence fewer rods are required. This is made possible by the lower fuel enrichment and higher conversion ratio possible in the ORNL core which is relatively free of neutron poisons. As will be discussed later, the fuel element fabrication costs for the ORNL core should be lower than for the Savannah. Certainly a major factor is that most of the development costs should already have been paid for by the gas-cooled reactor program.

The cost of the shielding for the ORNL reactor is much less than that for the pressurized water reactor because no secondary shield is required. This comes about because of the large difference in activation of the reactor coolants. Activation of the oxygen in the water to form N^{16} makes necessary a heavy secondary shield around the steam generators and the reactor coolant system of pressurized-water reactors. Such an arrangement is not required for a gas-cooled reactor employing a gas such as carbon dioxide, helium, or nitrogen, as has been demonstrated by the British experience with the Calder Hall reactors. This reduced shielding requirement produces a marked reduction in the gas-cooled reactor shielding costs relative to the Savannah. Similarly, the containment provisions for the gas-cooled reactors are much less than those for pressurized-water reactors because of the reduction in the quantity of gas that may be released in the event of a severe rupture in the reactor coolant system. This was discussed previously in the section on hazards. A third major item of expense in the pressurized-water reactor is the extra complication

of the primary coolant system. The extra pumps, storage tanks, instrumentation and related equipment have no counterpart in the gas turbine or nuclear gas engine plants.

The cost of the equipment and piping associated with the working fluid system seems to be much the same for all five power plants. Similarly, the cost of the heat exchangers runs about the same for them all except that the nuclear gas engine with its small heat exchanger requirements inherently involves much lower heat exchanger costs.

The cost of the propulsion machinery for the various types of plant differs somewhat. It is to be expected the pressurized-water plant involves a more expensive turbine than the oil-fired plant because of the lower pressure and temperature steam available. The cost of the nuclear gas engine as entered in the table is about 25% higher than the cost of the turbines. It should be remembered that this cost has been estimated only very roughly by doubling the cost of diesel engines of the same bore, stroke, and number of cylinders. It is very possible that the actual cost of these engines would be much less. In fact, if a fair number of engines were produced, they should not be much more expensive than conventional diesel engines of the same size.

In an effort to improve the cost comparison between the nuclear gas engine and the Savannah power plant provisions were made in Table VIII for the propeller and shafting and a take-home electric motor for use in an emergency. These items are not included in Table VI or in Figs. 14 and 15. Note also that the estimate for the controllable pitch propeller for the General Atomics plant seems low relative to that for the fixed pitch propellers of the nuclear gas engine and the Savannah.

The cost of the installation depends on the number of pieces of equipment, their size, and the length, size, and weight of the piping required to make the connections. A comparison of the layouts presented in Figs. 13 and 14 with those for the steam or gas turbine plants indicates that there should be less need for large, high-pressure pipes, and there will be fewer connections and valves for these pipes than in the gas turbine or steam turbine installations. As a result it is reasonable to expect that the installation costs for the nuclear gas engine would run substantially less than for the pressurized-water or gas turbine plants.

Several approaches were considered for estimating the costs of the design of the nuclear plant, the design of the complete power plant installation, and the installation, leak-testing, and inspection of the installed equipment. This figure should also include provisions for contingencies. A value running about 50 to 60% of the cost of the components would be consistent with common practice. In compiling the costs given in Table VIII the dollar entries for the gas turbine plants were taken from the reference reports while the entry for the Savannah was simply the difference between the component costs as estimated in compiling this report and the sum of the contract prices for the complete plant. Note that all three values for the engineering, installation, and contingency, as expected, are close to 50% of the cost of the components. A value close to the median percentage of the three cases was then used to obtain the corresponding cost for the nuclear gas engine plant.

In reviewing the cost data of Tables VII and VIII it should be remembered that the intent was not to effect an exact comparison because that is not possible. Rather, the intent was to gain some insight into the probable relative amounts of the major cost items to establish whether or not the nuclear gas engine shows promise of giving substantially lower over-all costs than the pressurized-water reactor. It appears from Tables VII and VIII that the design premises are, in fact, sound, and that by using a gas that is subject to relatively little neutron activation as both the reactor coolant and the thermodynamic cycle working fluid it is possible to eliminate the high-temperature heat exchangers, the extra pumps, tanks, and other equipment required for the reactor coolant system, much of the shielding, and much of the containment requirements. This eliminates or reduces the corresponding costs.

In comparing the costs of the nuclear gas engine with the gas turbine the comparison must be drawn primarily with the General Atomic design because too few cost data were given in the Ford Instrument Company proposal to assess the costs properly. It appears that many of the costs will be similar, but that the cost of the high-temperature heat exchangers and much of the piping required for the gas turbine will be eliminated. Even more important, the lower gas temperature acceptable

with the nuclear gas engine will permit the use of simpler fuel elements, less surface area for the stainless steel fuel capsules (or cladding), and hence a lower fuel enrichment. All of these factors will lead to lower costs for the nuclear gas engine as compared to the gas turbine plants.

Fuel Costs

The fuel elements for the Savannah are very similar to those for the proposed nuclear gas engine in that both make use of UO_2 in stainless steel capsules. In view of this similarity, an effort has been made to estimate the fuel costs for both the Savannah and the proposed nuclear gas engine on the same bases. The results are presented in Table X. The key data on enrichment, burnup, etc., for the Savannah were taken from a Babcock & Wilcox report. These items are marked with an asterisk to distinguish them from the values estimated by the writer.

The core life at normal power was calculated from the thermal efficiency of the plant assuming an output of 20,000 shp plus 1500 bhp to the generator supplying electric power for the ship's services. The fuel reprocessing cost was computed as in ORNL-2500 assuming a charge of \$12.40/kg U plus a fixed charge of \$100,000 per loading for plant clean-up on the unfavorable assumption that the preceding and following batches fed to the reprocessing plant will be of a different enrichment.

While the estimated fuel fabrication cost for the Savannah has not been included in Table VII where it has little effect on the over-all costs, it also is not included in Table X because of the many uncertainties associated with it. However, the greater weight of uranium, the larger number of capsules, the much greater number of UO_2 pellets, and the greater complexity of the fuel element assemblies will all tend to make the fuel fabrication costs higher for the Savannah. In particular, note that each fuel element in the Savannah consists of a large number of fuel capsules brazed together -- a design feature which is likely to increase fuel costs substantially because it is difficult to get virtually perfect brazing of all of a thousand or so joints in such an assembly. Since there are 32 fuel elements, each assembly of this sort that must be rejected in production leads to an increase of about 3% in the average fuel element fabrication cost for the core. The

TABLE X

Fuel Cost Data

	<u>N.S. Savannah</u>	<u>Nuclear Gas Engine</u>
Number of fuel elements	32*	665
Number of fuel capsules	5,248*	4,655
Number of capsules per element	164*	7
Number of UO ₂ pellets per loading	693,000*	200,000
Fuel capsule OD - in.	0.50*	0.80
Fuel capsule length - in.	68*	22.5
Fuel capsule wall thickness - in.		0.032
Estimated fabrication cost per loading - \$		300,000
Initial enrichment - %	4.10*	3.00
Final enrichment - %	3.27*	2.09
Initial conversion ratio**	0.41	0.65
Life of fuel loading at normal power - days	880	890
Burnup - Mwd/tonne U	7,760*	8,350
Bhp hours per loading	455 x 10 ⁶	460 x 10 ⁶
Initial loading - Kg U	7,038.5*	5,700
Initial loading - Kg U ²³⁵	288.5*	171
Weight of U ²³⁵ consumed - Kg	58.8*	51.9
Weight of Pu formed - Kg		40.2
Final weight of Pu - Kg	20.9*	30.2
Initial value of U - \$/Kg	551.70	375.50
Final value of U - \$/Kg	418.40	233.85
Initial value of U - \$/loading	3,883,140	2,140,350
Final value of U - \$/loading	2,915,909	1,312,950
Value of U consumed - \$/loading	967,231	827,400
Pu credit at \$12/g - \$/loading	250,800	362,500
Net cost of fissionable material consumed - \$/loading	716,431	464,900

* Data from BAW-1044, Rev. 1, 5-9-58, p. 9.

** Definitions for conversion ratio vary. These values are for the ratio of the fissionable material formed to that burned.

TABLE X (continued)

	<u>N.S. Savannah</u>	<u>Nuclear Gas Engine</u>
Fuel rental charge - \$/loading*	535,000	298,000
Fuel reprocessing cost - \$/loading	186,548	170,000
Cost of fuel consumed - mills/bhp-hr	1.57	1.01
Fuel rental cost - mills/bhp-hr	1.18	0.64
Fuel reprocessing cost - mills/bhp-hr	0.41	0.37
Fuel costs exclusive of fabrication - mills/bhp-hr	3.16	2.03
Fuel fabrication cost - mills/bhp-hr		0.65
Total fuel cost - mills/bhp-hr		2.68

* Fuel rental charge calculated at 4% of the initial value of the fuel for the life of the loading assuming that the reactor operates at the equivalent of normal power 70% of the time.

cumulative effects of these factors seem likely to make the fuel fabrication cost for the Savannah higher by a factor of two or more than for the nuclear gas engine with no allowance for the development and tooling costs. In the case of the nuclear gas engine these costs will be covered by the GCR program now underway and hence an allowance of only \$125,000 was included for the incremental expense for the special fuel support requirements of this marine application. To make the estimates of Table VII comparable, no provisions for developmental work were included in estimating the Savannah fuel costs but allowances were made for tooling.

From Table X it appears that, if the fabrication costs are omitted, the costs of the fuel for the nuclear gas engine will run only about 60% of the corresponding costs for the pressurized water reactor. This arises largely from the lower enrichment and better conversion ratio inherent in the gas-cooled reactor because it is not handicapped by neutron captures in the coolant. The higher thermal efficiency arising from the higher temperature thermodynamic cycle is also a factor.

RANGE OF APPLICATION AND DEVELOPMENT POTENTIAL

Propeller Requirements

Probably the first step in appraising the applicability of the nuclear gas engine to the propulsion of merchant ships is to establish the basic requirements imposed by propeller considerations. Operating experience and cost data indicate that it is best where practicable to put all of the power into a single screw. However, usual harbor channel depths coupled with propeller fabrication, shafting, and support problems are such that propellers for merchant ships are normally not made in diameters greater than approximately 25 ft. With this size of propeller, efficiency considerations lead to the conclusion that the power input per propeller shaft ought not exceed approximately 20,000 shp. This can be deduced from Fig. 22 which shows the propeller diameter and rpm as functions of power input and vessel speed for a constant propulsive efficiency of 70% for representative vessel speeds. Thus it appears that attention should be centered on the preliminary design of engines well suited to supplying 20,000 hp to a single propeller shaft because this high power application is of the greatest interest. Smaller outputs down to about 3000 hp are also of interest for many applications and should also be considered.

Engine Size, Weight, and Cost

The limited use of large reciprocating engines indicates that a logical next step is to establish the relations between the power output of reciprocating engines and their size, weight, and cost. The effects of the various parameters determining the power output of an engine can be deduced readily from the well-known formula for calculating the power output of a reciprocating engine, namely:

$$\frac{PLAN}{33,000} = \text{BHP}$$

The first item, P, is the mean effective pressure in pounds per square inch. It depends on the allowable peak pressure in the cylinder and on the thermodynamic cycle upon which the engine operates. For practical purposes it can be considered as independent of the engine size. L is the length of the piston stroke in feet, and in general is equal to or

FIGURE 22. PROPELLER DIAMETER AND RPM AS A FUNCTION OF SHAFT HORSEPOWER AND VESSEL SPEED FOR A PROPELLER EFFICIENCY OF 70 %.

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somewhat greater than the cylinder bore. A is the area of the piston in square inches. N is the total number of power strokes per minute from all of the pistons in the engine. In a two-stroke cycle engine it will be equal to the rpm times the number of cylinders.

It can be seen from the above that the power output of an engine can be increased in several ways. In practice it has been found that, for a given peak pressure in the cylinder and a given basic thermodynamic cycle, the mean effective pressure in the cylinder will be defined within fairly close limits. Further, it has been found best for long engine life and high reliability to limit the piston speed to about 1200 fpm. This means that the engine rpm will vary inversely with the piston stroke. Since the stroke is normally either equal to or somewhat greater than the cylinder bore, this in turn means that the engine rpm will be approximately inversely proportional to the cylinder bore. Thus the principal means of increasing the engine output are to increase either the number of cylinders or the cylinder bore, or both. If this is done for a given piston speed, the power will increase directly as the square of the cylinder bore.

The engine weight, cost, and power increase approximately in direct proportion to the number of cylinders. The effects of an increase in bore are more complex. The engine weight will increase with cylinder bore in a fashion which can be readily deduced from strength considerations. For a constant peak pressure in the cylinder, the cylinder wall thickness will have to increase directly with the cylinder bore while the cylinder weight will increase as the cube. At the same time, the connecting rod and crankshaft cross sectional areas will have to be directly proportional to the area of the piston in order to keep the stresses constant irrespective of the cylinder bore. Thus the weight of these parts will increase as the cube of the cylinder bore because their lengths will increase directly with the bore. Figure 23 shows data for many different marine engines from five different manufacturers. In all cases but one the data plot within 20% of the mean curve which has the slope of the ideal cube curve deduced above. In all cases the entire engine weight was divided by the number of cylinders to give the values plotted. Note that, as the number

FIG. 23. EFFECT OF CYLINDER BORE ON ENGINE WEIGHT PER CYLINDER FOR MARINE DIESEL ENGINES. 433 091

of cylinders is increased, the weight per cylinder tends to be reduced somewhat because of the reduced proportion of weight required for such items as engine accessories.

In view of the above discussion indicating that the engine weight will vary as the cube of the cylinder diameter while the power will vary as the square of the diameter for a constant mean effective pressure and piston speed, it follows that the engine weight in pounds per horsepower will vary as the $3/2$ power of the cylinder bore. Thus the higher the power of the engine the greater its weight per horsepower if the number of cylinders and piston speed are kept constant.

Engine cost in dollars per pound and in dollars per horsepower has been investigated but it has been difficult to correlate the data because engine costs for a given power output are more a function of the number of engines of a given model produced per year than of any other factor. In general, for developed engines of which 20 or more are produced per year, it appears that the engine cost is somewhere between \$1 and \$4 per lb. The engines costing more per pound are the higher output engines which are highly supercharged and operate at the higher piston speeds so that the higher costs per pound are partially offset by the higher power outputs. As a result, the cost per horsepower tends to run from \$50 to \$75 per horsepower with the higher value prevailing for the very large engines having bores of the order of 30 in., or for the very high performance engines designed for operation at piston speeds in excess of 1400 fpm.

In considering the above it is clear that it is desirable to make the cylinder bore as small as possible and yet obtain the required power output. The largest marine engines in production are 16-cylinder V-engines. Aircraft engines of 28 and even 42 cylinders have been built, but substantial service difficulties have been experienced because of the large number of cylinders requiring attention. Thus it would appear best to consider 16 as the largest practicable number of cylinders. Experience with multiple engine units indicates that two engines coupled to a reduction gear have an important advantage over a single engine in that they

give increased reliability. Three engines coupled to a single reduction gear begin to present problems in that the number of auxiliaries such as oil coolers, jacket water coolers, etc., requiring attention becomes substantial, while four engines seem to represent the upper limit of the amount of such complication which operators regard as acceptable. Discussions with operators indicate that a propulsion unit employing two or at the most three engines would represent the best compromise in their eyes.

Figure 24 has been prepared to show the maximum power obtainable from a 16-cylinder engine presuming a 130-bmep and a 1200-fpm piston speed with a cylinder bore-stroke ratio of 5:6. Note that the engine rpm and the weight as calculated from Fig. 23 are also plotted on the same coordinates. If the speeds and powers of Fig. 24 are compared to those indicated by Fig. 22 it immediately becomes apparent that the engine rpm given by the 1200-fpm piston speeds leads to an engine rpm which is three times that required to give a propeller efficiency of 70%. This in turn means that a reduction gear will be required between the engine and propeller shaft if the full output potential of the engine is to be utilized. If the bore-stroke ratio were kept constant and the engine speed reduced to the desired propeller speed so that the reduction gear could be eliminated, the engine cost and weight would be increased by a factor of approximately two. In practice it is usual to increase the stroke to values as great as double the bore. This helps materially to cut the engine weight and cost per horsepower but still leads to a less favorable power plant weight than can be obtained through the use of a reduction gear. The cost of a reduction gear for coupling the higher speed engines to the propeller shaft will run approximately \$20/hp (or of the order of 1/3 the cost of the engines) while its weight will run 3 or 4 lb/hp (or less than 1/10 the weight of the engines). Hence the cost and weight of the gear is much more than offset by the lower cost and weight of the higher speed engines. In view of this it appears from Fig. 24 that two engines of about 18-1/2-in. bore or three engines of approximately 16-in. bore would provide a promising power plant of 20,000 hp.

FIGURE 24

EFFECTS OF ENGINE POWER OUTPUT ON THE CYLINDER BORE, RPM, AND ENGINE WEIGHT FOR 16 CYLINDER NUCLEAR GAS ENGINES OPERATING AT A BMEP OF 130 PSI AND A PISTON SPEED OF 1200 FPM.

Comparison with the Gas Turbine

The reactor requirements of the nuclear gas engine and a nuclear gas turbine have much in common. In both cases the highest achievable gas temperature leaving the reactor is desirable, but in both cases the same basic materials limitations will apply. A reactor which will give good performance with a gas turbine should also give good performance with a nuclear gas engine. The reverse is not necessarily the case because the nuclear gas engine should be able to operate well on nitrogen whereas the gas turbine is likely to give difficulty (except in small sizes) because the pumping losses with nitrogen impose a much greater penalty on over-all performance so that the larger nuclear gas turbines are probably limited to operation with CO_2 or helium. This in turn limits the materials of which the reactor core can be fabricated. CO_2 may not be used with graphite at temperatures in excess of about 1000°F . The ready availability of nitrogen as compared to helium is a major advantage of this gas and hence of a thermodynamic cycle which will permit its use.

The fuel costs will be related to the degree to which the gas temperature leaving the reactor must approach the peak allowable fuel element temperature. If the latter is taken as 1500°F for the stainless steel fuel element capsules, the only way the gas temperature can be made to approach the fuel element temperature more closely is to provide increased fuel element surface area. Thus on the basis of a consistent set of conditions the amount of stainless steel and hence the amount of neutron-absorbing material will be twice as great for a reactor designed to give a 1300°F gas outlet temperature as for one designed to give an 1100°F gas outlet temperature. At the same time, for the same size of reactor core, the gas passages will have to be cut to a much smaller equivalent diameter to provide the greater surface area in the same volume. This in turn will lead to much closer tolerance requirements on the fuel element dimensions to maintain the same relative hot spot allowances. As a matter of fact, the tolerance problem becomes progressively more difficult as the fuel element structure becomes more finely divided because the difficulty of maintaining dimensional

tolerances tends to increase rapidly as the tolerance range is reduced. Thus the cost of fabrication increases more rapidly than linearly with the amount of surface area provided. The amount of enrichment required and the conversion ratio are also adversely affected as the percentage of stainless steel in the core is increased.

The pumping losses should be essentially similar for the gas turbine and the nuclear gas engine. The pressure losses in the valve ports may be somewhat higher in the nuclear gas engine than the corresponding losses in the gas turbine. On the other hand, elimination of the regenerators and the pressure losses that they entail should more than offset this. Further, eddy losses during the compression and expansion process should be very small in the nuclear gas engine whereas in the gas turbine they will run of the order of 10% of the work of compression or expansion. On the other hand, the mechanical efficiency of the nuclear gas engine will be lower than that of the gas turbine. In the balance, the lower sensitivity of the nuclear gas engine to losses during the compression, expansion and flow processes should lead to a better over-all thermal efficiency at low peak gas temperatures than can be obtained with the gas turbine, and this can be accomplished without regenerators and the expense, complication, and pressure loss that they entail.

Other advantages and disadvantages of the nuclear gas engine relative to the gas turbine are concerned with the design and operating characteristics of the machinery. The nuclear gas turbine will be less inclined to give difficulty with oil entrainment in the gas stream delivered to the reactor although oil seal leakage in the compressor could present a problem. Achieving a leak-tight system in a gas turbine should be somewhat easier than with a nuclear gas engine with its substantial requirements for access ports for servicing the valve gear and other equipment.

The inherent characteristics of the nuclear gas engine should make it a much more flexible plant to operate than a gas turbine. The difficult matching problem between the compressor and the turbine is eliminated and the speed range over which the engine will operate well is continuous from zero to full speed as compared to the relatively high degree of care that must be exercised to avoid difficulties with

surging in the compressor for the gas turbine. The availability of a very large torque at low speeds for reversing, and the ability to reverse rapidly, are also major advantages of the nuclear gas engine. These will greatly ease the reactor control problem for conditions involving a rapid reversal from full speed ahead to full speed astern.

The cost of developing a new reciprocating engine is high but probably less than the cost of developing a new gas turbine. Some indication of this is given by the fact that Nordberg recently developed a new light-weight diesel having a normal output of 3600 hp with a maximum output of 4000 hp. This engine was designed to effect a 35% reduction in weight over conventional engines of the same output. The total cost of designing, building, and extensive testing of the engine was about \$1,200,000. While a nuclear gas engine would represent a much greater departure from usual practice, this gives some indication of the cost of designing and building a first unit. The cost of testing would, of course, be higher for the nuclear gas engine because it would probably be necessary to make use of a special large electrical or gas-fired heat source.

Development Potential

The development potential of a new type of power plant is an important consideration. The bmepp of the nuclear gas engine might be increased over the 130 psi generally used in this report. This might be accomplished by any of several methods. The most effective approach would be to increase the gas temperature leaving the reactor as indicated in Fig. 10. Going from an 1100°F to a 1300°F reactor outlet temperature should increase the imep in the engine by approximately 35% with little change in over-all thermal efficiency.

As indicated in Table II the output of the engine might be increased by approximately 50% by using helium in the place of nitrogen. If helium were used and the reactor gas outlet temperature increased to 1300°F, the output of the power plant might be practically doubled over the values given for the proposed plant for the N.S. Savannah.

A third factor that might serve to increase the output of the engine would be an exhaust turbine-driven blower employed to provide

a portion of the pumping power for the cooling phase of the cycle. This would probably permit an increase in bhp of around 5% and a similar improvement in thermal efficiency. The mechanical friction in the engine might be reduced by operating the crankcase at a pressure approximately equal to the pressures in the cooling loop. This would reduce the piston and piston ring friction as well as the connecting rod and main bearing loads and might increase the mechanical efficiency of the engine by several per cent which would increase both the output and the thermal efficiency by the same factor.

The weight of the reactor and shield assembly and the fuel investment could be reduced markedly by using beryllium oxide as a moderator or by reducing the power. The core diameter will vary approximately as the square root of the reactor power output, while the use of beryllium oxide will permit the construction of reactor cores of about 70% of the diameter of comparable graphite-moderated cores. Thus it appears possible to reduce the reactor pressure vessel diameter approximately in half if the power output were to be reduced to perhaps 10,000 shp and beryllium oxide were substituted for graphite. Still further reductions in diameter could be achieved through the use of more highly enriched fuel. It also appears possible to effect a very substantial reduction in the weight of the reciprocating engines as was discussed previously. While it is not clear what advantages would accrue from these weight reductions, it is important to note the possibilities because they may prove advantageous for some applications.

In reviewing the previous sections it appears that the nuclear gas engine should be applicable to marine propulsion in units from perhaps 3000 to 30,000 hp. As indicated in Fig. 7, it should have particular advantages over the pressurized-water steam turbine type of plant in the smaller units such as would be required for electric power generation or in the smaller vessels requiring power plants of 3000 to 10,000 shp.

DEVELOPMENT PROBLEMS AND RECOMMENDED PROGRAM

The preliminary study covered by this report indicates that the nuclear gas engine is a very promising type of power plant for marine propulsion. At the same time, this preliminary study has disclosed and discussed many difficult development problems. Those affecting feasibility have been summarized in approximate order of importance in Table XI.

A careful appraisal of the problems of Table XI indicates that it would be best to consider dividing the development program into three phases. Neither of the latter two would be initiated unless it was clearly justified by its predecessor.

Phase I

The first phase would be concerned with the greatest uncertainties both in the technical feasibility for the proposed nuclear gas engine and the cost of the equipment required. These uncertainties appear to be associated mainly with the performance obtainable from the engine. This in turn is believed to be primarily dependent on the breathing losses which would actually be experienced and the residual gas volume that would be implicit in a design that would provide adequate port flow passage area. Therefore, the major emphasis in the first phase of the development program should be placed on the first four items in Table XI; i.e., the design of the valves and ports for both the high-pressure and the low-pressure phases of the cycle to determine whether the relatively low losses estimated in this preliminary study can actually be achieved. This first phase study should be made by an engine manufacturer with a staff having much experience in the problems of engine breathing, valve dynamics, engine structural design, and production. Concurrently with this intensive study of the special engine design problems imposed by the breathing requirements of a nuclear gas engine, attention should also be directed toward the disclosure of other areas which might have strong adverse effects on feasibility, particularly those which might induce substantial differences between the ideal and the actual indicator diagrams. Thus the first phase study should not have as its objective the design of a

TABLE XI

Problems in the Development of a Gas-Cooled Reactor
Coupled to a Reciprocating Engine

1. Engine breathing losses for the high-pressure pumping loop may prove to be excessive. These losses depend on:
 - a. Practicable valve accelerations and valve gear design.
 - b. The port flow passage area available. This depends also on cylinder head structural, fabrication, and cooling considerations.
 - c. Pressure losses in the gas discharge and return manifolds in the cylinder head.
2. Engine breathing losses for the low-pressure pumping loop may prove to be excessive. These losses depend on:
 - a. Port area available in the cylinder wall.
 - b. Port configuration for good scavenging of the cylinder.
 - c. Scavenging gas flow ratio required.
 - d. Power available from an exhaust turbine-driven blower for a practicable installation.
 - e. Manifold attachment details giving good leak-tightness characteristics.
3. Residual gas dilution of the charge returning to the cylinder from the reactor may be excessive. This is closely related to item 1-b.
4. In the design of the cylinder head and valve gallery it will be difficult to provide good isolation of the hot gas manifolds from the cylinder head, the valve gear, and the valve gear lube oil while yet accommodating differential thermal expansion between the cylinder and the manifold.
5. The fuel element, while similar to that in the GCR, will differ in some respects and will require testing. It will have to be secured against vertical and lateral accelerations and hence will present special problems in the design of the supporting devices and the charging machine.
6. Lubricating oil may be swept from the cylinder to the reactor and undergo some cracking followed by polymerization. This might lead to troubles with deposits on the valves, pistons, and cylinder walls, and fuel elements.
7. Provision for afterheat removal may introduce substantial complications.
8. Operation and control will present many problems. The transients associated with emergencies involving abrupt engine stoppage or reversals from full speed ahead to full speed astern deserve particular attention.
9. Startup may pose special problems.
10. Evolving an engine well adapted to maintaining a high degree of leak-tightness will require new solutions to old and stubborn problems.

complete power plant, but should instead define the practicable output and efficiency of the basic cycle. This work should be directed toward cylinders intended for use in engines of approximately 3000 and 10,000 bhp, i.e., cylinder diameters of about 12 in. and 20 in., respectively.

Once the cycle losses had been estimated and preliminary cylinder designs prepared, the approximate size, weight, and operating characteristics of these engines could be estimated, but detailed engine designs should not be attempted at this stage. These two power outputs would cover the range of interest because two 10,000-hp engines could be readily coupled to provide a 20,000-hp plant while a single 3000-hp engine would be of a good size for an auxiliary power plant in large vessels or a main propulsion power plant for small vessels. With rough over-all weight and performance data for these two engine sizes, it will be apparent approximately what can be expected for engines of intermediate output.

During phase I no time should be expended on the many details which, while important in a final design, will not have any great bearing on the point in question, that of feasibility. For example, no effort should be expended during phase I on the design of the reactor and its immediately associated equipment as it is believed that the work currently underway for other programs will provide the information most needed, namely, further experimental data on fuel elements consisting of UO_2 in stainless steel capsules. This work is so vital to these other programs -- e.g., the GCR, PWR and the Savannah -- that it will certainly be carried out, and there should be no need to provide funds for such work in the development of the nuclear gas engine. The balance of the reactor development problems are much less formidable. They, too, are being attacked in other programs and should not require any special funding in a development program for a nuclear gas engine.

Phase II

The second phase of the program should involve two parallel efforts. The first should be the design, construction, and operation of a single cylinder or small multicylinder engine to demonstrate feasibility and to assist in the detailed investigation of the principal losses to be expected in the cycle. Coupled with this, the engine design studies initiated in phase I should be extended, and over-all power plant layouts and evaluations should be carried out for several types of vessel. This should include the preliminary design of the reactor and gas system. Particular attention should be paid to the last six items of Table XI. Note that operation of a single cylinder test engine will serve not only to demonstrate feasibility and provide much information relative to the first four items in Table XI, but it will also give a valuable insight into the problems indicated in items 6, 8, and 9. In particular, since the principal damage to lubricating oil vapor in the gas passing through the high-temperature phase of the cycle will be from thermal effects, the tendency for deposits to form on the fuel elements, valves, pistons, etc., could be investigated nicely in a single-cylinder engine test. Since service life and maintenance problems cannot be predicted well except from test experience, and since these factors are so important in the evaluation of the power plant, the small-scale engine test work will be vital to the evolution of the design.

Phase III

The third phase of the program would be the detail design, construction, and testing of a complete power plant of 3000 to 5000 hp. This would be large enough to give a good full-scale demonstration of feasibility and yet should be small enough to keep the costs to a reasonable level.

APPENDIX A

TABLE XII

Effects of Reactor Outlet Gas Temperature on the
Nuclear Gas Engine Cycle with No Allowances
for Pumping Losses or Residual Gas

Working fluid = N_2
 Compression ratio = 5:1
 Peak gas pressure = 1500 psia
 Initial specific volume = 36.33 ft³/lb-mol
 Portion of stroke lost during scavenging = 1/8
 Clearance volume = 0%

Section A

	<u>T</u>	<u>h</u>	<u>P_r</u>	<u>V_r</u>	<u>P</u>	<u>V/V₁</u>
1.	540	3150.3	1.0281	5637	158.7	1
2.	1020	7121	9.712	1127	1500	0.2
3.	1360	11,104	46.90	356.9	1500	0.306
4.	1006	7021	9.197	1168	294.5	1

Net work = 1626 Btu/lb-mol.
 Thermal efficiency (η) = 39.85%
 imep = 211.5 psi

Section B

	<u>T</u>	<u>h</u>	<u>P_r</u>	<u>V_r</u>	<u>P</u>	<u>V/V₁</u>
1.	540	3150.3	1.0281	5637	158.7	1
2.	1020	7121	9.712	1127	1500	0.2
3.	1460	10,347	36.43	430.1	1500	0.2862
4.	912	6355	6.469	1502	266.5	1.0

Net work = 1336 Btu/lb-mol
 η = 41.4%
 imep = 173.5 psi

TABLE XIII (continued)

Section C

	<u>T</u>	<u>h</u>	<u>P_r</u>	<u>V_r</u>	<u>P</u>	<u>V/V₁</u>
1.	540	3150.3	1.0281	5637	158.7	1
2.	1020	7121	9.712	1127	1500	0.2
3.	1560	11,104	46.90	356.9	1500	0.306
4.	1006	7021	9.197	1168	294.5	1

Net work = 1626 Btu/lb-mol

η = 39.85%

imep = 211.5 psi

Section D

	<u>T</u>	<u>h</u>	<u>P_r</u>	<u>V_r</u>	<u>P</u>	<u>V/V₁</u>
1.	540	3150.3	1.0281	5637	158.7	1
2.	1020	7121	9.712	1127	1500	0.2
3.	1660	11,870	59.6	298.8	1500	0.3255
4.	1101	7702	12.8	922	322	1.0

Net work = 1895 Btu/lb-mol

η = 39.9%

imep = 246.5 psi

Section E

	<u>T</u>	<u>h</u>	<u>P_r</u>	<u>V_r</u>	<u>P</u>	<u>V/V₁</u>
1.	540	3750.3	1.0281	5637	158.7	1
2.	1020	7121	9.712	1127	1500	0.2
3.	1760	12,644	74.87	252.3	1500	0.345
4.	1160	8129	15.479	804.1	310	1

Net work = 2164 Btu/lb-mol

η = 39.2%

imep = 280.7 psi

Section F (Compression ratio = 4.8:1)

	<u>T</u>	<u>h</u>	<u>P_r</u>	<u>V_r</u>	<u>P</u>	<u>V/V₁</u>
1.	540	3750.3	1.0281	5637	170	1
2.	1001	6985	9.076	1183	1500	0.2082
3.	1560	11,104	46.9	356.9	1500	0.3245
4.	1030	7192	10.059	1100	316	1

Net work = 1659 Btu/lb-mol

η = 40.2%

imep = 215.5 psi

TABLE XIII

Effects of Reactor Outlet Gas Temperature on the
Nuclear Gas Engine Cycle with Allowances
for Pumping Losses and Residual Gas

Working fluid = N_2
Compression ratio = 5:1
Peak gas pressure = 1500 psia
Initial specific volume = 38.5 ft³/lb-mol
Portion of stroke lost during scavenging = 1/8
Clearance volume = 1%

Section A

	<u>T</u>	<u>h</u>	<u>P_r</u>	<u>V_r</u>	<u>P</u>	<u>V/V₁</u>
1.	570	3959.1	1.2422	4924	159.2	1
2.	1074	7510	11.7	985	1500	0.2
3.	1345	9481	26.75	539.6	1456	0.258
4.	802	5575	4.116	2090	224	1

Net work = 717.5 Btu/lb-mol

η = 36.25%

imep = 88 psi

Section B

	<u>T</u>	<u>h</u>	<u>P_r</u>	<u>V_r</u>	<u>P</u>	<u>V/V₁</u>
1.	570	3959.1	1.2422	4924	159.2	1
2.	1074	7510	11.7	965	1500	0.2
3.	1535	10,914	44.09	373.5	1456	0.2945
4.	975	6795	8.245	1268	272	1

Net work = 1268 Btu/lb-mol

η = 37.3%

imep = 155.8 psi

Section C

	<u>T</u>	<u>h</u>	<u>P_r</u>	<u>V_r</u>	<u>P</u>	<u>V/V₁</u>
1.	570	3959.1	1.2422	4924	159.2	1
2.	1074	7510	11.7	985	1500	0.2
3.	1725	12,372	69.29	267.5	1456	0.331
4.	1158	8113	15.383	808	323	1

Net work = 1772 Btu/lb-mol

η = 36.4%

imep = 217.7 psi

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TABLE XIV

Effects of Compression Ratio on the Nuclear Gas Engine Cycle
with Allowances for Pumping Losses and Residual Gas

Working fluid = N_2
Reactor outlet temperature = $1100^\circ F$
Initial gas temperature = $110^\circ F$
Peak gas pressure = 1500 psia
Portion of stroke lost during scavenging = $1/8$

Section A

Compression ratio = 3:1			Initial specific volume = 18.99		
<u>T</u>	<u>h</u>	<u>P_r</u>	<u>V_r</u>	<u>P</u>	<u>V/V₁</u>
1. 570	3959.1	1.2422	4924	323	1
2. 881	6137	5.763	1641.3	1500	0.333
3. 1538	10,937	44.42	371.5	1456	0.600
4. 1279	8986	22.15	619.5	726	1

Net work = 1073 Btu/lb-mol
 η = 22.35%
imep = 267 psi

Section B

Compression ratio = 4:1			Initial specific volume = 28.25		
<u>T</u>	<u>h</u>	<u>P_r</u>	<u>V_r</u>	<u>P</u>	<u>V/V₁</u>
1. 570	3959.1	1.2422	4924	217	1
2. 986	6877	8.596	1231	1500	0.25
3. 1537	10,929	44.31	372.3	1456	0.402
4. 1101	7702	12.801	923	421	1

Net work = 1265 Btu/lb-mol
 η = 31.2%
imep = 211.5 psi

Section C

Compression ratio = 6:1			Initial specific volume = 49.5		
<u>T</u>	<u>h</u>	<u>P_r</u>	<u>V_r</u>	<u>P</u>	<u>V/V₁</u>
1. 570	3959.1	1.2422	4924	124	1
2. 1150	8063	15.046	821	1500	0.167
3. 1535	10,914	44.09	373.5	1456	0.224
4. 876	6099	5.637	1668	186.2	1

Net work = 1169 Btu/lb-mol
 η = 41.0%
imep = 111.6 psi

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TABLE XV

Effects of Compression Ratio on the Nuclear Gas Engine Cycle
with Allowances for Pumping Losses and Residual Gas

Working fluid = He
Reactor outlet temperature = 1100°F
Initial gas temperature = 110°F
Peak gas pressure = 1500 psia
Portion of stroke lost during scavenging = 1/8

Section A

Compression ratio = 1:75 Initial specific volume = 2.54

	<u>T</u>	<u>P</u>	<u>V/V₁</u>
1.	570	599	1
2.	1.447	1500	1
3.	825	1500	0.571
4.	1547	1490	1.08

Net work = 101.4 Btu/lb-mol
 η = 11.24%
imep = 189 psi

Section B

Compression ratio = 2:0 Initial specific volume = 3.22

	<u>T</u>	<u>h</u>	<u>P</u>	<u>V/V₁</u>
1.	570	0	476	1
2.	900	412	1500	0.5
3.	1547	1219	1490	0.862
4.	1403	1040	1169	1

Net work = 178 Btu/lb-mol
 η = 22.0%
imep = 261 psi

Section C

Compression ratio = 2.5 Initial specific volume = 4.04

	<u>T</u>	<u>P</u>	<u>V/V₁</u>
1.	570	377	1
2.	988	1500	0.435
3.	1547	1490	0.686
4.	1205	798	1

Net work = 217 Btu/lb-mol
 η = 31.1%
imep = 254 psi

TABLE XV (continued)

Section D

Compression ratio = 3:1 Initial specific volume = 4.65

	<u>T</u>	<u>h</u>	<u>P</u>	<u>v/v₁</u>
1.	570	0	329	1
2.	1042	589	1500	0.40
3.	1547	1219	1490	1
4.	1100	661	635	1

Net work = 227 Btu/lb-mol

η = 36.0%

imep = 230 psi

Section E

Compression ratio = 3:5 Initial specific volume = 6.31

	<u>T</u>	<u>h</u>	<u>P</u>	<u>v/v₁</u>
1.	570	0	243	1
2.	1175	755	1500	0.333
3.	1550	1223	1490	0.443
4.	907	421	386	1

Net work = 211 Btu/lb-mol

η = 45.0%

imep = 158.1 psi

Section F

Compression ratio = 3:5 Initial specific volume = 8.12

	<u>T</u>	<u>P</u>	<u>v/v₁</u>
1.	570	187.5	1
2.	1303	1500	0.2856
3.	1550	1490	0.3415
4.	760	254.5	1

Net work = 168.3 Btu/lb-mol

η = 54.5%

imep = 98.1 psi

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