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RESEARCH MEMORANDUM

AN ANALYSIS OF A PISTON-TYPE GAS-GENERATOR ENGINE

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NATIONAL ADVISORY COMMITTEE FOR AERONAUTICS

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SUMMARY

An analysis was made of a compound engine consisting of a compressor, a two-stroke compression-ignition engine that furnished the power for the compressor, and a turbine that was driven by the gases from the engine and produced the net useful work of the cycle. This form of power plant is called a gas-generator engine.

The analysis indicates that an engine of this type should be capable of operating with a brake specific fuel consumption of 0.32 pound per horsepower-hour and should have an installed specific weight comparable to that of a typical turbine-propeller engine. This performance should be obtained with limiting values of cylinder pressure and turbine-inlet temperature that are compatible with reliable engine operation.

INTRODUCTION

The high gas temperatures necessary for the attainment of low specific fuel consumption in conventional heat-power cycles can at present be achieved only by the use of some form of the reciprocating-type internal-combustion engine. Likewise, in order to obtain low specific engine weight, expansion of the gases to low pressures must be accomplished by the use of a gas turbine. These facts make the combination of a reciprocating engine and a gas turbine a desirable arrangement.

In the usual form of such a compound engine (references 1 to 5), the greatest portion of the work of the cycle is usually performed by the reciprocating engine. The high specific weight of the reciprocating engine and the relatively low specific weight of the gas-turbine component makes it advantageous, however, for the gas turbine to perform as much of the work of the cycle as is consistent with the attainment of high over-all efficiency. By this division of work, the reciprocating-engine component performs only the work accomplished during that portion of the cycle that involves high pressures and high temperatures; consequently, this component can then be made relatively small and light.

An analysis of an engine having a division of work such that the power of the reciprocating engine is just sufficient to drive the compressor is reported herein. Under these conditions, the turbine furnishes the entire net output of the complete engine. This combination of a reciprocating-type internal-combustion engine that drives its own supercharging compressor and generates gas for further expansion through a turbine will be called a piston-type gas generator in this report. The combination of a piston-type gas generator and a gas turbine will be called a gas-generator engine. A diagrammatic sketch of such a power plant is shown in figure 1. Because the primary purpose of the reciprocating element in this engine is to produce hot gases for the turbine, this component will be designated a piston-type burner.

SELECTION OF CYCLE

The conventional compound engine consists of a compressor, a piston engine, and a turbine, all geared to a single shaft (references 1 to 5). Such an engine usually operates with a fuel mixture at or near the stoichiometric mixture and is limited in its performance by (a) a maximum allowable peak cylinder pressure, and (b) a maximum allowable turbine-inlet temperature. In order to establish relations that will form the basis for modifications to the cycle of the conventional compound engine and thus obtain a lower specific power-plant weight, the effect of operation at a leaner mixture ratio will first be considered. This change will permit higher manifold pressures, with resultant higher air flows, for a given limiting maximum cylinder pressure. The net result of the two changes is that the increased air flow and the reduced mixture ratio so counteract one another that the power output of the piston-engine component remains approximately constant. At the same time, the higher air flow results in a larger turbine output and compressor input; as a result, the output of the entire engine increases by the amount of the difference between the increase in compressor input and the increase in turbine output.

It is generally recognized that the turbine and the aircraft compressor are characteristically light devices for high power ratings and that in the compound engine the weight of the piston-engine component is the predominating factor in fixing the specific weight of the power plant. Because the increase in power resulting from reducing the mixture ratio in the compound engine is obtained without any change in the weight of the piston-engine component, the net result should be a reduction in the specific weight of the power plant.

Altering the division of work between the piston-engine and turbine components causes the characteristics of the cycle to approach more closely those of the Brayton cycle; therefore, reducing the mixture ratio will lead to higher specific fuel consumptions.

Because of the manner in which the specific weight and the specific fuel consumption vary, the proper division of work between the piston engine and the turbine depends upon the service for which the aircraft is intended and must be arrived at from an aircraft-range analysis. Inasmuch as such data suitable for selecting the proper division of work are lacking, any decision regarding the division of work can only be arbitrary. For the purposes of this report, the division of work selected was that at which the piston-engine component delivers just enough power to operate the compressor and the net useful work of the cycle is derived from the turbine. Analysis has shown that this division of work will permit sizeable reductions in specific weight over that of the conventional compound engine and at the same time will retain the low fuel consumption obtainable with this engine. In addition, the separation of the compressor and turbine drives allows more flexibility in the construction and operation of such an engine.

METHODS OF ANALYSIS

Combustion in gas generator. - The analysis shows that the inlet-gas pressures and temperatures and the fuel-air ratios used with the piston-type burner are such that compression ignition with fuel injection is the most practical combustion process for this cycle. With such a process, the maximum burner pressure attained depends primarily upon the rapidity of combustion and some definite rate of combustion-pressure rise must be assumed. For present purposes, the rate of burning was assumed to be equal to that producing the maximum rate of pressure rise occurring in a spark-ignition engine. This burning rate requires the use of efficiencies that are compatible with the Otto cycle and are somewhat higher than those encountered in the conventional Diesel cycle at a given expansion ratio. It should be pointed out, however, that considerable combustion pressure is developed when the burning rate approaches constant-volume combustion. It can be shown that in a pressure-limited engine, the efficiency of the Otto cycle is less than that of the Diesel cycle because of the higher compression and expansion ratios permitted by the Diesel cycle. The choice of the high burning rate used in this analysis therefore leads to conservative results. Compression ignition permits the use of the two-stroke cycle with a consequent reduction in specific weight. Furthermore, the analysis indicates that the fuel-air ratios are sufficiently lean for the simple piston-ported loop-scavenged cylinder to perform satisfactorily, although this scavenging process is inferior to that of the uniflow cylinder.

Performance data. - The symbols and equations used in obtaining the performance data for the gas-generator engine are given in appendixes A and B, respectively. No convenient expression relating the burner-inlet pressure, the burner-expansion ratio, and the mixture ratio to the limits of burner-exhaust temperature and of peak burner pressure could be obtained. Because of this limitation, it was necessary to set up a series of curves for each operating condition of the gas generator and to use a graphical solution to locate the desired operating point.

In order to have the gas generator operate with the performance given by this analysis at more than one condition, it is necessary that the compressor and the turbine operate over a range of mass-flow rates and pressure ratios without appreciable changes in efficiency. Although several means by which this condition may be approached are available, the magnitude of the problem does not permit their discussion here. For this reason, the efficiency of the compressor and the turbine was assumed to remain constant over a range of pressure ratios and mass flows and a variable-area turbine nozzle was assumed.

The primary variables used in the analysis and the ranges through which these variables were investigated are given in the following table:

Variable	Basic value	Range investigated
Peak burner pressure, lb/sq in.	1600	1200 - 1600
Turbine-inlet temperature, $^{\circ}$R	2260	1960 - 2260
Altitude, ft	20,000	Sea-level - 50,000
Burner-scavenge ratio	1.0	0.7 - 1.3
Compressor efficiency	0.85	0.7 - 1.0
Burner thermal efficiency	[a]Standard	0.60 - 1.05 × standard

[a]Standard burner thermal efficiency is that defined by equations (7) and (8) of appendix B.

The basic values of the variables are the values that were maintained constant when the effect of one of the variables was being investigated. Because the effect of turbine efficiency was obvious, it was not included as a variable. Consideration of jet thrust also was omitted in order to avoid an additional variable.

The basic values of the limits in the preceding table (peak burner pressure and turbine-inlet temperature) were chosen, after an examination of a wide variety of data taken from both piston-type and turbine-type engines, to be compatible with reliable engine operation at the present time. In this connection it should be pointed out that the turbine-inlet temperature selected, 2260° R, is somewhat below the measured peak gas temperatures in current turbine-type engines. Temperature gradients in the gas stream in these engines cause wide differences to exist between the peak and the mean gas temperatures. Because of the manner in which the hot gases are produced in the gas generator, temperature gradients should not be a problem in operation of this engine and operation should be satisfactory at the chosen temperature.

Weight data. - The estimated weight of the gas-generator engine was obtained by scaling component parts of various current reciprocating, turbine-propeller, and turbojet engines to match the conditions of the gas-generator engine, and by taking their assembled weight as that of one possible configuration of the gas-generator engine. A résumé of the specific-weight calculations is given in appendix C.

RESULTS AND DISCUSSION

The results of the analysis of the gas-generator engine described herein are presented in plots showing the effect of compressor-pressure ratio, altitude, engine operating limits, scavenge ratio, and component efficiency on the basic performance of the gas-generator engine. An analysis and breakdown of the specific weight of the engine is also presented.

Performance of Gas-Generator Engine

Basic performance data. - The performance of the gas-generator engine for three values of the burner-expansion ratio is shown in figure 2. Superimposed on this plot are lines of constant peak burner pressure equal to 1600 pounds per square inch and lines of constant turbine-inlet temperature equal to 2260° R; the intersection of these two curves locates the optimum operating point from a power-output standpoint for the chosen engine operating conditions and limits. At any other point, the engine may be limited either by peak burner pressure or by turbine-inlet temperature and losses in engine efficiency and, or, power output result. The operating range of the engine is shown by the shaded area on the curves.

At the optimum operating point, the brake specific fuel consumption is about 0.32 pound per horsepower-hour. This low value of fuel consumption is obtained principally by the use of the high expansion ratio of the gases (from the mean combustion pressure to ambient-air pressure). The specific output (net work output of gas-generator power plant) is 0.052 Btu per cycle per cubic inch of cylinder volume or, for a mean piston speed of 2600 feet per minute, a stroke of 6 inches, and a compression ratio of 4.2, the output is 4.18 brake horsepower per cubic inch of piston displacement. The high power output is attributed to the large air capacity of the gas-generator engine in comparison to that of the conventional reciprocating engine. This power output is obtained despite the fact that the brake specific air consumption is approximately twice that of the reciprocating engine. At the optimum point, the mixture ratio of the engine is about 0.032 pound of fuel per pound of air.

Effect of altitude. - The performance of the gas-generator engine is shown in figure 3 as a function of altitude. As the altitude of operation is increased, simultaneous adjustments must be made in manifold pressure and burner-expansion ratio in order to maintain the prescribed limits of peak burner pressure and turbine-inlet temperature. At the same time, however, the brake specific air consumption decreases because the cycle efficiency and the burner-mixture ratio increase. The net result of these variations is that the power output of the gas-generator engine is substantially constant up to an altitude of about 30,000 feet; above this altitude, the power output decreases. The power drops about 30 percent when the altitude is increased from 30,000 to 50,000 feet. This variation in power output is partly caused by the manner in which the ambient temperature varies with altitude.

The low fuel consumptions shown in figure 3 are a direct result of the high cycle efficiencies attainable by a complete-expansion engine. Because the gas-generator engine is pressure-limited, the expansion ratio increases with altitude and thus the fuel consumption decreases. The brake specific fuel consumption is 0.32 pound per horsepower-hour at 20,000 feet and decreases 24 percent in an ascent from sea level to 50,000 feet.

Figure 3 also shows that an effective change in expansion ratio is necessary for operation at the selected limits at different altitudes. This change is necessary in order to preserve the balance between the peak burner pressure and the turbine-inlet temperature. Such a change can be effectively brought about merely by varying the time of fuel injection. The resultant changes in the pressure-volume relation are shown by the typical burner indicator cards presented in figure 4.

Effect of engine operating limits. - The effect of varying the peak burner pressure and the turbine-inlet temperature on the performance of the gas-generator engine is shown in figure 5.

Reducing the turbine-inlet temperature from 2260° to 1960° R at constant peak burner pressure reduces the power output about 20 percent. A small decrease in the specific fuel consumption is also noted; this decrease is a result of operating the engine at a higher mean combustion pressure. In order to obtain the limiting cylinder pressure at reduced turbine-inlet temperature, the manifold pressure must be reduced and the burner-expansion ratio raised. The net result is that the specific air-flow rate (air-flow rate through piston-type burner, lb/(cycle)(cu in.) burner volume) is reduced and the brake specific air consumption is raised, whereas the over-all expansion ratio is little changed. The selection of a limiting turbine-inlet temperature therefore has a primary effect on the specific weight of the engine.

Reducing the peak burner pressure at constant turbine-inlet temperature requires that both the manifold pressure and the burner-expansion ratio be lowered. The net effects of these changes are that brake specific air consumption is changed only a small amount, but the air-flow rate and therefore the power output are again appreciably reduced. Lowering the peak burner pressure results in reduced over-all expansion ratios and consequently in higher fuel consumptions; therefore, the selection of a limiting peak burner pressure is of importance in determining both the specific weight and the specific fuel consumption of the engine.

Effect of change in scavenge ratio. - Exhaust-gas dilution (at low values of scavenge ratio) and low combustion efficiency (at high values of scavenge ratio) distinctly limit the range of scavenge ratio over which the gas generator can operate. This limitation is a natural result of the gas-generator mode of operation. The work of the burner, which is equal to the work of the compressor, must increase in proportion to the quantity of air handled. The burner, however, is limited in the quantity of air it can accommodate; therefore, when the scavenge ratio is increased, the mixture ratio in the burner must be enriched in order to preserve the balance between compressor and burner powers.

Figure 6 shows the effect of change in scavenge ratio on the performance of the gas-generator engine. As the scavenge ratio increases, the power output undergoes a large increase, which may provide a convenient means of controlling power output of this engine. This change in power is almost directly proportional to the quantity of air flowing through the engine, because the changes in the brake specific air consumption are small; as a result, power output is approximately proportional to the product of manifold density and scavenge ratio.

The rise in the fuel-consumption curve is caused by the changes in the burner pressure drop and in burner mixture ratio.

Calculations indicate that with a scavenge ratio of 1.3, the burner mixture ratio is in excess of 0.06. At a scavenge ratio of 0.7, the burner mixture ratio is much leaner but approximately 35 percent of the charge consists of exhaust gases from the previous cycle.

Effect of change in component efficiency. - In the gas-generator engine the effect of change in turbine efficiency is clearly apparent. Because the turbine delivers all the net brake output of the engine, the brake specific fuel consumption of the engine is inversely proportional and the specific output of the engine is directly proportional to turbine efficiency.

The effect of changes in compressor efficiency is illustrated in figure 7. This figure indicates that such changes result in large changes in the specific output, but affect the brake specific fuel consumption only slightly. The magnitude of the loss resulting from reductions in compressor efficiency at any operating condition is dependent to a large extent on the brake specific air consumption of the engine. At conditions under which the brake specific air consumption is low for the basic compressor efficiency, the effect of change in compressor efficiency becomes small.

The effect of changes in thermal efficiency of the piston-type burner is difficult to treat analytically because of the intimate relations existing between thermal efficiency, peak burner pressure, and rate of pressure rise during combustion. If it is assumed that part of the fuel burns at approximately constant volume at the beginning of the expansion process and the rest burns at the end of the expansion, the gas-generator-engine performance will vary as shown in figure 8. In this plot, the ratio of actual burner efficiency to burner efficiency as defined by equation (8) of appendix B is used as the abscissa. This condition of reduced burner efficiency would correspond to the use of over-rich mixture ratios in the burner, where some of the scavenging air is required to complete combustion.

Figure 8 shows that improvements in burner efficiency improve the specific output. A change in the burner efficiency must be compensated for by adjusting the burner-expansion ratio; however, increasing the expansion ratio requires the use of lower manifold pressures with a consequent reduction in air-flow rates and in specific output.

It should be pointed out that if the burner efficiency is altered by a change from constant-volume to constant-pressure combustion, the over-all performance of the gas-generator engine will be improved, inasmuch as both higher manifold pressures and higher expansion ratios can be used.

Specific Weight of Gas-Generator Engine

The installed specific weight of the gas-generator engine was estimated to be 1.41 pounds per horsepower at 20,000-foot altitude. (See appendix C.) The operating conditions upon which this value is based are the basic conditions used throughout this report.

A comparison of the specific weight of the gas-generator engine with three other current propeller engines is as follows:

Engine	Type	Specific weight (lb/hp) (a)
A	Liquid-cooled reciprocating	1.98
B	Air-cooled compound	2.39
C	Axial-flow turbine propeller	1.36
Gas generator	Axial-flow compressor, liquid-cooled burner	1.41

[a]All specific weights based on installed weight without propeller and on maximum continuous power at 20,000-foot altitude.

The value of specific weight for the turbine-propeller engine in the preceding table may appear somewhat higher than anticipated, but the values are nevertheless comparable because the component parts of this turbine-propeller engine were used in making up the gas-generator engine. The turbine-propeller engine has a second advantage in that the power rating is based upon turbine-inlet temperatures 100° F higher than those used for the gas-generator engine and also includes the effective shaft horsepower resulting from jet thrust.

The low specific weight of the gas generator is due to two factors: high specific air-flow rates and low specific air consumption. The specific weight of the reciprocating engine is high because of its low air-flow rate; whereas that of the turbine-propeller engine is higher than would be expected on this basis because of the inherently poor specific air consumption of this engine.

CONCLUSIONS

An analysis has been made of a gas-generator compound engine, in which the piston-engine element supplies only enough work to drive the compressor. This particular division of work between the turbine and piston-engine elements was selected in an attempt to retain the desirable fuel-consumption characteristics of the compound engine and at the same time to achieve a low specific power-plant weight.

The analysis shows that the gas-generator engine can attain a low specific fuel consumption because of the high peak burner temperatures permitted by the piston-type burner and because of the complete expansion of the working fluid in the burner and the turbine. This type of power plant has an inherently low specific weight because of the high air-flow rate through the burner, which is obtained by reducing the work output of the piston-type burner in relation to that of the turbine.

On the basis of the analysis presented, it may be concluded that the gas-generator engine is seemingly capable of operating at moderate altitude with a brake specific fuel consumption of the order of 0.32 pound per horsepower-hour; the specific weight with limiting values of cylinder pressure and gas temperature compatible with reliable engine operation is comparable with that of the turbine-propeller engine.

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APPENDIX A

SYMBOLS

The following symbols are used in this report:

$c_{p,a}$ specific heat at constant pressure of compressor air, 0.24 Btu/(lb)(°R)

$c_{p,g}$ specific heat at constant pressure of turbine gases, 0.27 Btu/(lb)(°R)

e base of natural logarithms

H_a enthalpy of air entering compressor (taken as zero), Btu/lb

H_g enthalpy of gases entering turbine, Btu/lb

h_c lower heat of combustion of fuel, 18,500 Btu/lb fuel

K constant

p_a ambient-air pressure, lb/sq in. absolute

p_c burner-compression pressure, lb/sq in. absolute

p_e burner-discharge pressure, lb/sq in. absolute

p_m burner-inlet pressure, lb/sq in. absolute

Q_l heat loss from burner, Btu/lb fuel

R_e expansion ratio of fluid in burner

R_m over-all mixture ratio, lb fuel/lb air

$R_{m,b}$ mixture ratio in burner, lb fuel/lb air

R_p pressure ratio of compressor

R_s scavenging ratio (ratio of volume of air flowing through burner per cycle measured at burner-inlet conditions to volume of burner)

T_a ambient temperature, ^{o}R 824

T_c burner compression temperature, ^{o}R

T_g mean turbine-inlet temperature, ^{o}R

T_m burner-inlet temperature, ^{o}R

T_r temperature of residual gases in burner after blowdown, ^{o}R

T_s temperature in burner at end of scavenging, ^{o}R

W_c work of compressor, Btu/lb air

W_t work of turbine, Btu/lb air

γ ratio of specific heats of turbine gases

η_c adiabatic compressor efficiency

η_b burner thermal efficiency (actual)

η_{id} burner thermal efficiency (ideal)

η_r reduction-gear efficiency, 0.95

η_s scavenging efficiency (ratio of volume of air remaining in burner at end of scavenging process measured at inlet conditions to volume of burner)

η_t adiabatic turbine efficiency based on static downstream pressure, 0.85

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APPENDIX B

ANALYSIS OF GAS-GENERATOR PERFORMANCE

The following equations were used to estimate the performance of the gas-generator engine:

Compressor calculations. - The performance of the compressor on the basis of work done per pound of air handled can be calculated by simply assuming a value for the adiabatic efficiency. Thus,

$$p_m = R_p \left(p_a\right) \tag{1}$$

$$T_m = \frac{T_a}{\eta_c}\left[\left(R_p\right)^{0.283} - 1\right] + T_a \tag{2}$$

$$W_c = c_{p,a}\left(T_m - T_a\right) \tag{3}$$

Scavenging efficiency and scavenging ratio. - Data presented in reference 6 indicate that the scavenging ratio for a piston-ported, looped-scavenged cylinder can be represented quite accurately in the range under discussion by considering the engine ports as an equivalent orifice and using the orifice equation

$$R_s = K \sqrt{\left(1 - p_e/p_m\right) T_m}$$

In this equation, the constant K includes the area and the discharge coefficient of the equivalent orifice and a conversion factor to make the units consistent. Evaluating the constant K from the data of reference 3 for a mean piston speed of 1800 feet per minute and for the definition of scavenging ratio given results in the equation

$$R_s = 0.0910 \sqrt{\left(1 - p_e/p_m\right) T_m} \tag{4}$$

This equation is for port arrangement D (fig. 23, reference 6) with an inlet-opening time of 58° B.B.C. and an exhaust-opening time of 74° B.B.C. For a high-speed engine, the port areas can be so enlarged that the gas velocities through the ports are unchanged. Equation (4) will then apply also for piston speeds in excess of 1800 feet per minute.

Several types of scavenging process can occur in the piston-ported engine. In the case being considered, scavenging is assumed to occur through perfect mixing, where each element of incoming air mixes completely with the gases in the cylinder and an equivalent volume of the mixture then flows out of the cylinder. This assumption is based on the data in reference 7. For this process, reference 8 gives the equation

$$\eta_s = \left(1 - e^{-R_s}\right) \tag{5}$$

For this process, the temperature at the end of the scavenging process is given by the equation (reference 5)

$$T_s = \frac{T_m}{1 - \left(1 - \frac{T_m}{T_r}\right) e^{-R_s}} \tag{6}$$

The solution of this equation requires a knowledge of the temperature of the gases remaining in the cylinder at the end of the blowdown process. Reference 5 indicates, however, that at moderately high scavenging ratios the effect of change in this temperature on the final scavenging temperature is small; therefore, the temperature of the cylinder gases was assumed constant at 200° R.

Burner efficiency. - If the efficiency in the ideal constant-volume cycle is calculated with due regard for factors such as variations in the specific heats and chemical dissociation, then the calculated efficiency will closely agree with the efficiency of an actual engine. The agreement can be further improved by introducing empirical corrections for heat loss, combustion time, and other modifying factors.

In reference 9, such a cycle is treated and an empirical equation is developed for the efficiency of the ideal cycle as a function of the expansion ratio and the fuel-air ratio. This equation has been adopted as the basis of the work presented herein:

$$\eta_{id} = 1 - \left(\frac{1}{R_e}\right)^n$$

where

$$n = 0.3867 - \frac{6.5}{\frac{6.65}{R_{m,b}} - 35} - \frac{0.043}{R_e} \tag{7}$$

These efficiency equations were selected to be compatible with the assumed burning rate in the piston-type burner.

The following corrections were considered in order to make the calculated efficiencies more compatible with actual values; these losses were approximated from data presented in references 8 and 10:

Factor	Reduction in thermal efficiency
Heat transfer	5.0
Combustion time and variation	.5
Mechanical friction	2.0

Thus, the burner efficiency can be found from the equation

$$\eta_b = 0.925 - \left(\frac{1}{R_e}\right)^n \tag{8}$$

where n is calculated as in equation (7).

In applying this value of efficiency to the performance of the gas generator, it is necessary to satisfy the condition that

$$W_c = \eta_b R_m h_c \tag{9}$$

Because a simple relation between η_b and R_m does not exist, it becomes necessary to use a trial-and-error method of solution. The procedure for this case was as follows:

The burner efficiency was first approximated by the use of the equation

$$\eta_b' = 0.925 - \left(\frac{1}{R_e}\right)^{0.32} \tag{10}$$

(The use of a prime on the symbols indicates an approximation.) Using this value, the over-all fuel-air ratio was estimated by use of the relation

$$R_m' = \frac{W_c}{\eta_b' \, h_c} \tag{11}$$

This value of over-all mixture ratio was modified to represent approximately the mixture ratio existing in the burner cylinder as follows:

$$R_{m,b}' = \frac{R_s R_m'}{\eta_s} \tag{12}$$

Using this value of $R_{m,b}'$, the burner-efficiency calculations were repeated and the corrected mixture ratios were found from equations (9) and (12). One such approximation was usually sufficient.

Maximum burner pressure. - The calculation of the maximum burner pressure is necessarily complicated because the application of ideal-cycle relations for other than constant-pressure cycles indicates values of pressure that are much higher than those actually present. In this analysis, the calculated values of peak burner pressure were so modified by a factor determined from reference 11 as to make them more nearly representative of the actual values.

For the present purposes, it was assumed that the ratio between the ideal maximum burner pressure and the burner compression pressure was a function of the compression temperature and the mixture ratio in the burner. Actually, the compression pressure would also have a small effect on this quantity, but the errors resulting from neglecting this effect, particularly at lean mixtures, are within the accuracy of this report. Curves of the ideal combustion pressure ratio as a function of mixture ratio with compression temperature as a parameter were prepared and used in the analysis; the slope of the curves at the zero intercept can be found by the use of an air standard cycle, whereas the rich-mixture points can be evaluated by using the methods of reference 12.

Reference 11 shows that, in a spark-ignition engine at the fuel-air ratio at which the rate of pressure rise is nearly a maximum, the actual peak cylinder pressure is only 0.70 of the ideal value. Such a factor may be used to calculate the actual peak burner pressure; however, it obviously must be dependent upon the mixture ratio and must reach a limiting value of unity when the mixture ratio is zero. Determining the value of this factor at leaner mixtures from spark-ignition-engine data would be incorrect because of flame-speed effects. The factor C was therefore expressed as a linear function of the mixture ratio by means of the equation

$$C = -3.75\, R_{m,b} + 1.0 \tag{13}$$

The procedure for calculating the maximum burner pressure was as follows: The compression pressure and temperature were calculated by means of the equations

$$p_c = p_e \left(R_e\right)^{1.35} \tag{14}$$

$$T_c = T_s \left(R_e\right)^{0.35} \tag{15}$$

With these values, the ideal maximum burner pressure was found using the combustion-pressure-ratio curves. The actual pressure was then evaluated by multiplying the ideal pressure by the correction factor from equation (13).

<u>Turbine-inlet temperature.</u> - If the gas generator (compressor and burner) is isolated and treated as a steady-flow machine, the mean turbine-inlet temperature can be found by means of a simple heat balance

$$H_a + (h_c\, R_m) - Q_l = H_g\, (1 + R_m)$$

In order to evaluate this equation, it is first necessary to fix the heat loss Q_l. The heat loss was taken as being equal to 18 percent of the heat input (reference 10). Then, if the enthalpy of the entering air is taken as zero, because the enthalpy of the entering air and that of the burner gases is practically the same at temperature T_a, the quantity H_g can be expressed in terms of the turbine-inlet temperature. Thus,

$$(1 - 0.18)\, h_c\, R_m = c_{p,g}\, (T_g - T_a)\, (1 + R_m)$$

or

$$T_g = \frac{(1 - 0.18)\, h_c\, R_m + c_{p,g}\, T_a\, (1 + R_m)}{c_{p,g}\, (1 + R_m)} \tag{16}$$

The specific heat $c_{p,g}$ is the mean specific heat between the turbine-inlet and turbine-outlet temperatures. The value of this quantity was estimated from the data in reference 13 and was considered to be constant at 0.27 for this work.

Turbine calculations. - The turbine work per pound of air can be found from the conventional equation

$$W_t = \eta_t \, c_{p,g} \, T_g \left(1 + R_m\right) \left[1 - \left(\frac{p_a}{p_e}\right)^{\frac{\gamma-1}{\gamma}}\right] \tag{17}$$

Data from reference 13 was used to establish $c_{p,g} = 0.27$ and $\gamma = 1.34$.

Unit performance calculations. - It is desirable to express the output of the gas-generator power plant in terms that do not require reference to a specific burner size or speed. Such a term for the output is Btu per cycle per cubic inch of burner volume. In order to modify the turbine work to put it on this basis and also to consider the reduction-gear efficiency, the following equation was used:

$$\text{Brake output} = \frac{\eta_r \, W_t \, R_s}{1728} \, \frac{144 \, p_m}{53.3 \, T_m} \tag{18}$$

The brake thermal efficiency of the unit can best be expressed in terms of the brake specific fuel consumption

$$\text{bsfc} = \frac{2545 \, R_m}{\eta_r \, W_t} \tag{19}$$

APPENDIX C

ANALYSIS OF SPECIFIC WEIGHT OF GAS-GENERATOR ENGINE

The particular gas-generator engine selected for this analysis has a rating of 3000 brake horsepower at an altitude of 20,000 feet. The limitations assumed to apply for continuous engine operation were a peak burner pressure of 1600 pounds per square inch, a turbine-inlet temperature of 2260° F, and a mean piston speed of 2600 feet per minute.

Inspection of the performance data show that in the range of altitude from sea level to 20,000 feet the power output of the gas-generator engine is practically constant. Inasmuch as the compressor and the turbine pressure ratios increase with altitude, it is apparent that the conditions which will determine the weight of the engine that will operate over this altitude range will be those at 20,000 feet. Some consideration, however, should be given to increasing the power output for take-off purposes, which can most readily be accomplished by increasing the scavenge ratio (fig. 6) or the turbine-inlet temperature (fig. 5). Either method should require a larger compressor, turbine, and reduction gear. It is anticipated, however, that for short-period operation overloading of the reduction gear and relatively inefficient operation of the compressor and the turbine can be tolerated; as a result, the critical condition in determining the specific weight of the gas-generator engine is maximum continuous power at 20,000 feet.

The specific weight of the engine can be estimated by treating each component separately and then summing the component specific weights.

Compressor. - The performance data in figure 2 show that the selected gas-generator engine will require an air-flow rate of 8.0 pounds per second at a compressor-pressure ratio of 21 in order to deliver the required power. The nearest approach to such a compressor is that of an existing turbine-propeller engine; a 500-pound, 14-stage, axial-flow compressor handles 11.7 pounds of air per second at a compressor-pressure ratio of about 7. Analysis shows that 22 stages would be needed to reach a compressor-pressure ratio of 21. In order to scale this compressor to the size needed to accommodate the gas-generator engine, it was assumed that the weight of the compressor required would be directly proportional to the weight of air handled and to the number of stages. In addition, it was assumed that the compressor would be built in two parts in order to operate satisfactorily at the required pressure ratio, and that this construction would incur an additional 30-percent increase in weight. The specific weight of the gas-generator compressor would therefore be

$$\frac{500}{3000} \times \frac{8.0}{11.7} \times \frac{22}{14} \times 1.30 = 0.233 \text{ lb/bhp}$$

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Piston-type burner. - Examination of the data in figure 2 shows that the piston-type burner must have a cylinder volume of 1060 cubic inches for the required power output. The only two-stroke-cycle engine of this approximate size is the Junkers Jumo 207 engine (reference 14). This engine is of the two-stroke-cycle opposed-piston type, operates at a peak cylinder pressure of about 1600 pounds per square inch, has a cylinder volume of 1096 cubic inches, and weighs 1470 pounds, complete with all accessories. In considering the weight of this device for use in the gas-generator engine, 100 pounds were deducted from this weight figure inasmuch as the supercharger and the reduction gear are not needed. The specific weight of the piston-type burner therefore becomes

$$\frac{1370}{3000} = 0.457 \text{ lb/bhp}$$

Turbine. - In order to determine a reasonable weight for the turbine of the gas-generator engine, the turbines from several jet engines were weighed; the resulting data are shown in figure 9 as a function of gas-flow rate. For a mean gas-flow rate of 420 cubic feet per second, it appears, from figure 9, that the turbine would weigh 105 pounds. These data, however, are for pressure ratios requiring single-stage turbines, and a four-stage turbine would be required for the gas-generator engine. Because the weight of the turbine is roughly proportional to the number of stages, the specific weight of the gas-generator turbine is

$$\frac{105 \times 4}{3000} = 0.140 \text{ lb/bhp}$$

Reduction gear. - The only current turbine-propeller engine for which data are available has a reduction gear designed to reduce shaft speed from that of the turbine to that of the propeller and to handle a maximum of 2800 horsepower. This gear, including accessory drives, weighs 435 pounds. In adapting this gear to the gas-generator engine, it was assumed that 35 pounds could be eliminated by removing the accessory-drive gears (because these drives have already been provided on the piston-type burner) and that for a change from 2800 to 3000 horsepower the weight would be directly proportional to the horsepower. Thus, the specific weight of the reduction gear becomes

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$$\frac{400}{2800} = 0.143 \text{ lb/bhp}$$

<u>Compressor-drive gear.</u> - Analysis of the performance data shows that approximately 1940 horsepower is required to drive the compressor. No data are available for a gear having the required speed ratio for this application. Because the speed ratio here is about one-half of that required for the propeller-reduction gear, however, a single-stage reduction gear should be sufficient and the specific weight of the required gear can be estimated by assuming that it weighs, on a specific basis, 70 percent as much as the propeller reduction gear. In addition, the weight is assumed to be increased by 30 percent because the gear must have two output shafts. The specific weight of the compressor-drive gear for the gas-generator engine therefore becomes

$$0.143 \times \frac{1940}{3000} \times 0.70 \times 1.3 = 0.084 \text{ lb/bhp}$$

<u>Heat exchangers.</u> - The weight of the radiator and oil cooler was calculated by assuming that 16 percent of the heat of the fuel is rejected to the coolant and 2 percent to the oil. These figures result in heat-rejection rates of 5880 Btu per minute to the oil and 47,000 Btu per minute to the coolant. Data in reference 15 indicate that the oil cooler will reject approximately 2500 Btu per minute per square foot of frontal area per 100° F initial temperature difference, and that the corresponding heat-transfer rate for the radiator would be 6000 Btu per minute per square foot per 100° F; these figures are for coolers 12 inches deep. For a fluid temperature of 250° F and NACA standard air at 20,000 feet altitude, the necessary oil-cooler volume is therefore 5.6 cubic feet and that of the radiator 10.8 cubic feet. Wet weights for these heat exchangers were found to be 48 pounds per cubic foot for oil coolers and 54 pounds per cubic foot for radiators. On this basis, the specific weight of the oil cooler for the gas-generator engine becomes

$$\frac{5.6 \times 48}{3000} = 0.090 \text{ lb/bhp}$$

and that of the radiator

$$\frac{10.8 \times 54}{3000} = 0.194 \text{ lb/bhp}$$

Miscellaneous. - Under this heading is included the weight of coolant and oil in the gas-generator engine itself, reinforcing of the manifolding to withstand the high pressures, and other items not heretofore considered. This weight was estimated at 200 pounds, or

$$\frac{200}{3000} = 0.067 \text{ lb/bhp}$$

Summary. - The installed specific weight of the gas generator in pounds per brake horsepower-hour is the sum of the preceding items and is summarized as follows:

Compressor	0.233
Piston-type burner	.457
Turbine	.140
Reduction gear	.143
Compressor-drive gear	.084
Radiator	.194
Oil cooler	.090
Miscellaneous	.067
Installed specific weight	1.408

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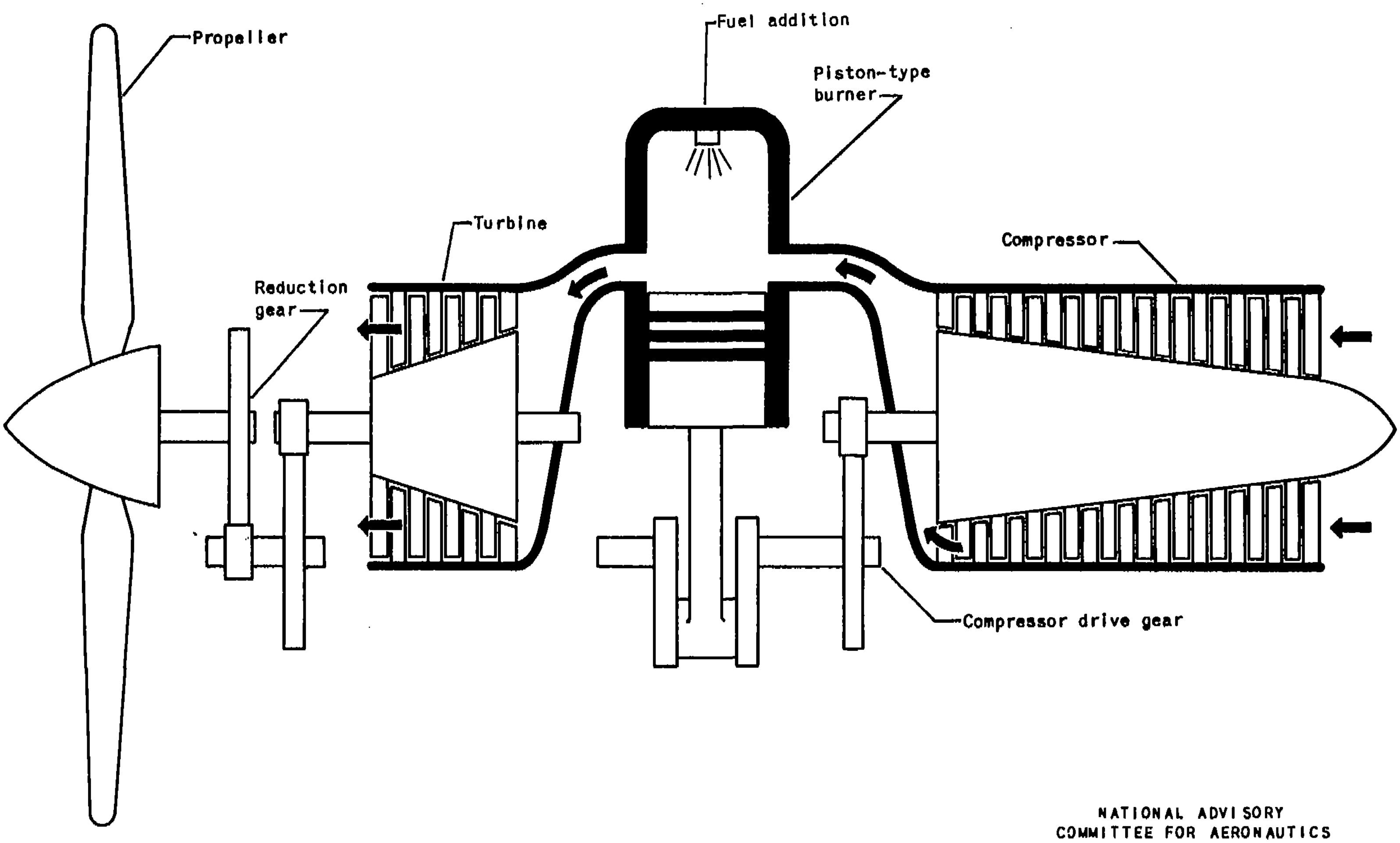

Figure 1. - Diagrammatic sketch of gas-generator engine.

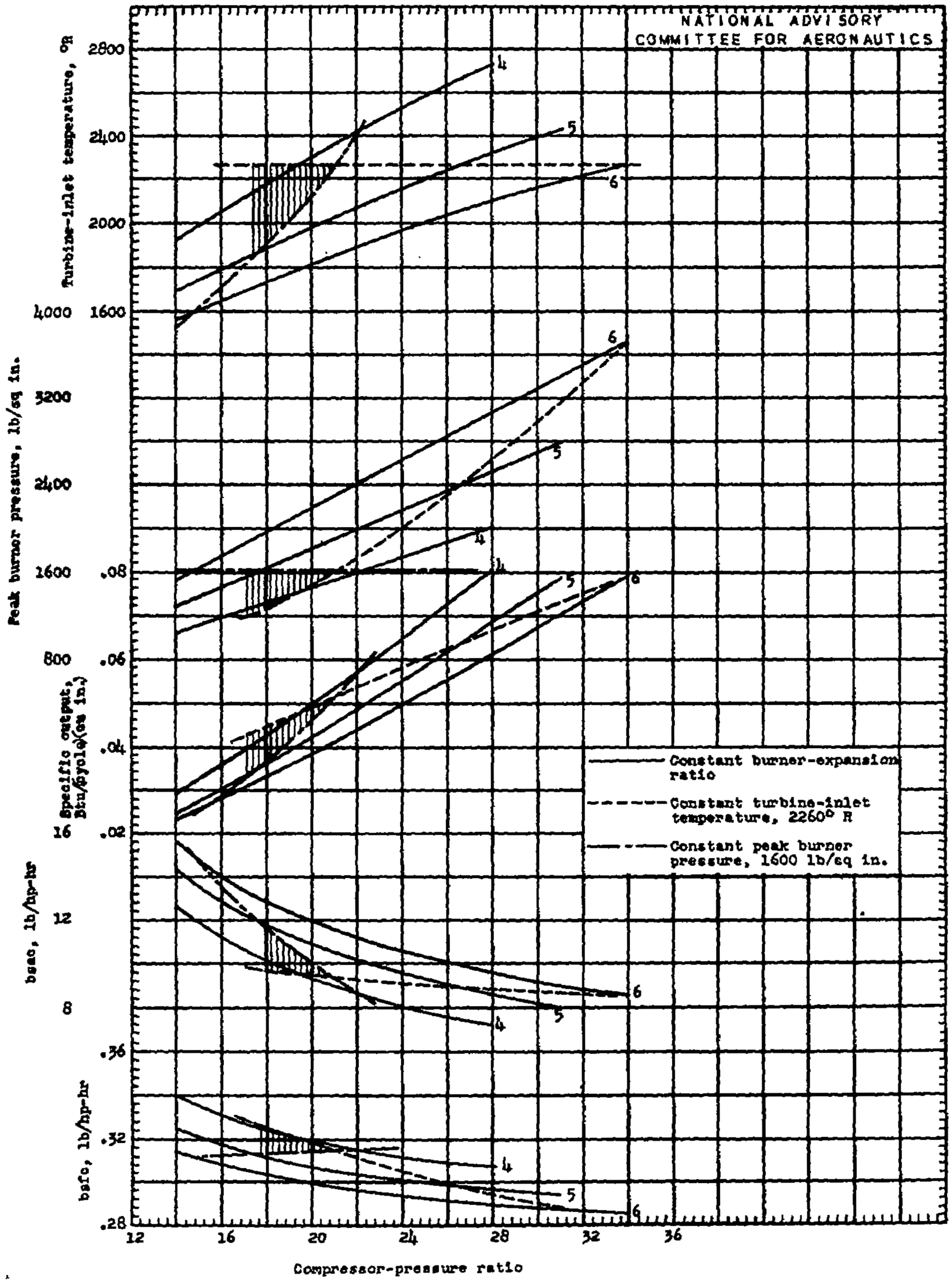

Figure 2. - Basic performance data calculated for gas-generator engine for three burner-expansion ratios. Altitude, 20,000 feet; compressor efficiency, 0.85; scavenge ratio, 1.00. Shaded area indicates permissible operating range of engine.

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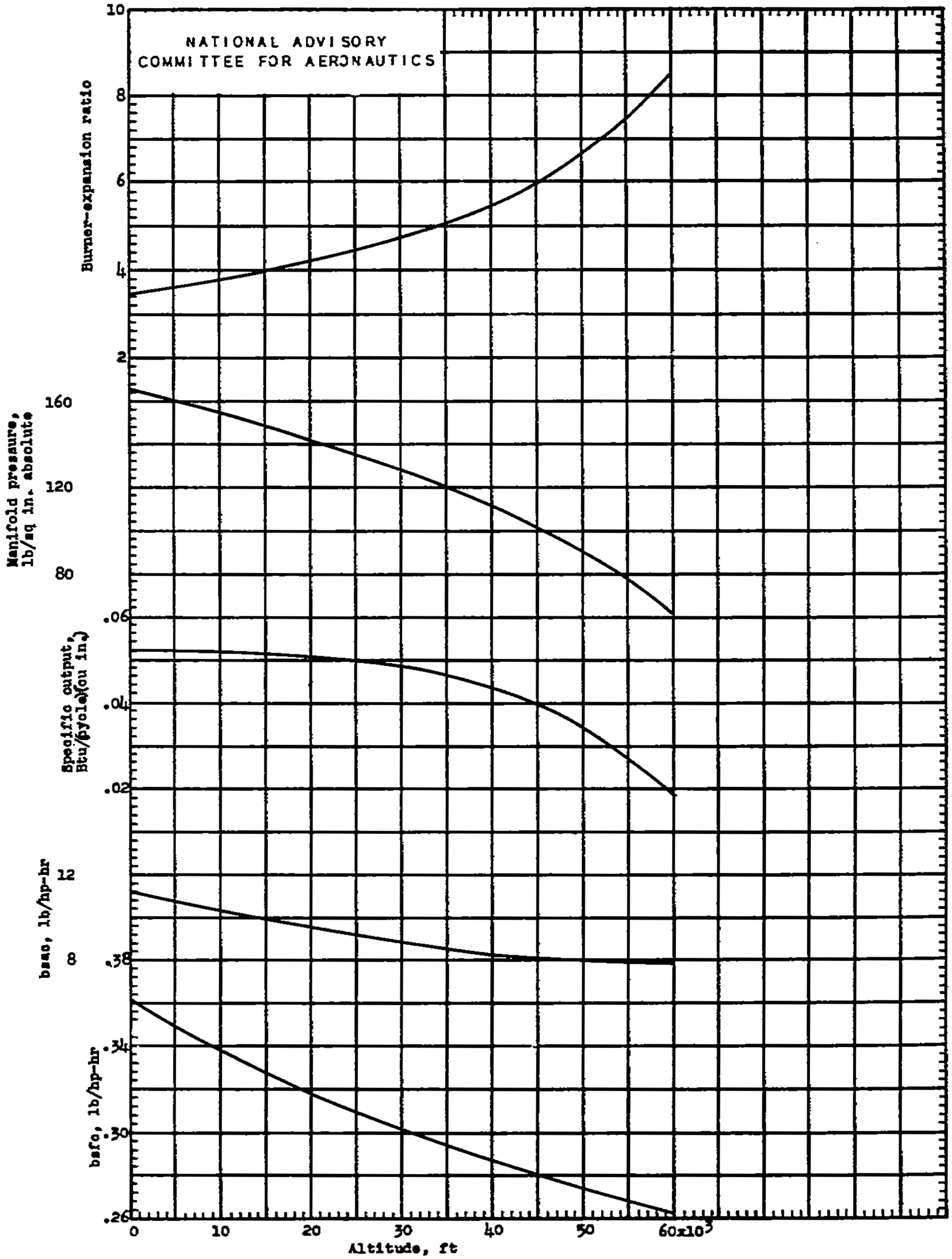

Figure 3. - Effect of change in altitude on calculated performance of gas-generator engine. Peak burner pressure, 1600 pounds per square inch; turbine-inlet temperature, 2260° R; compressor efficiency, 0.85; scavenge ratio, 1.00.

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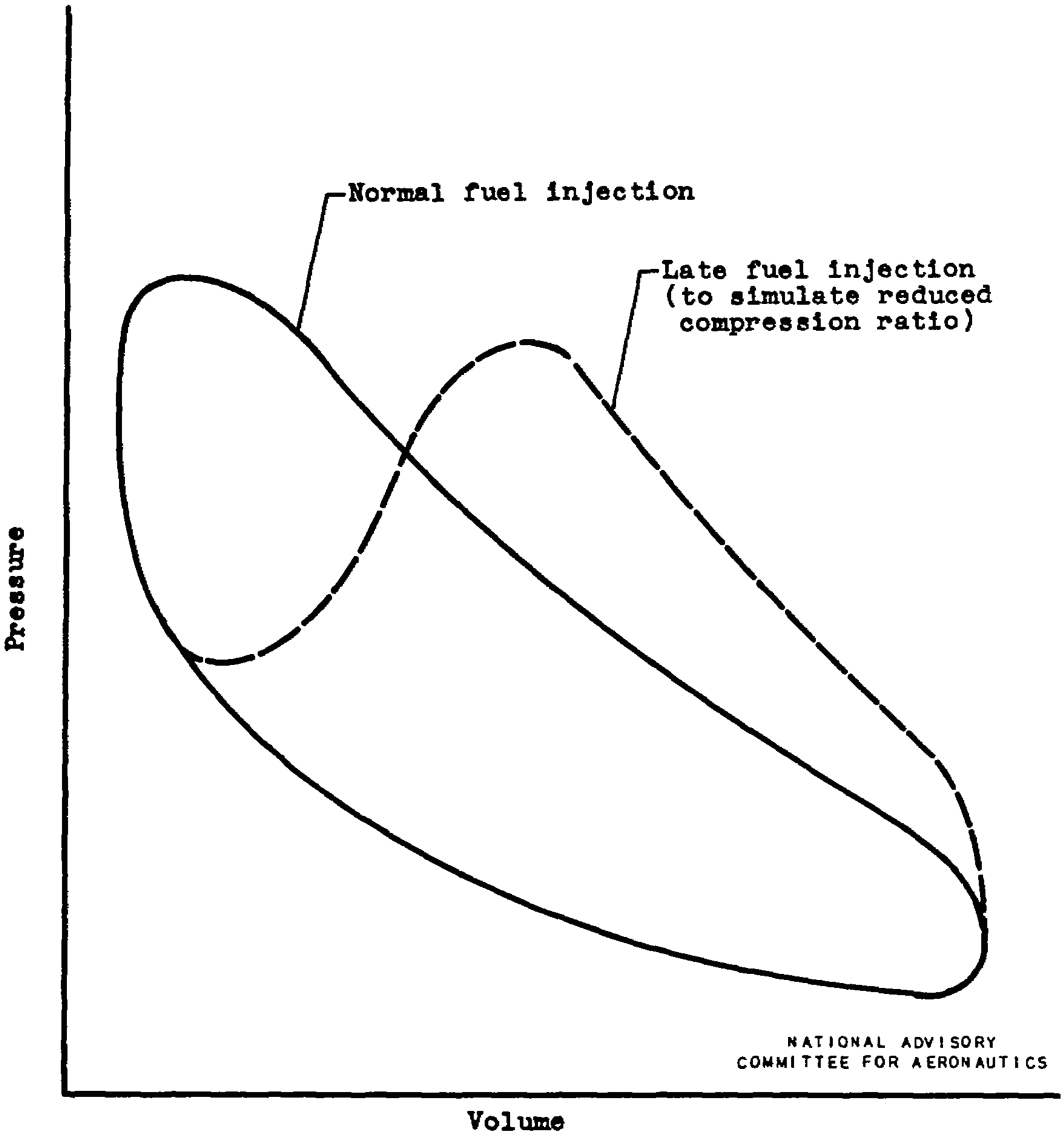

Figure 4. - Representative pressure-volume indicator card for combustion process in gas-generator engine.

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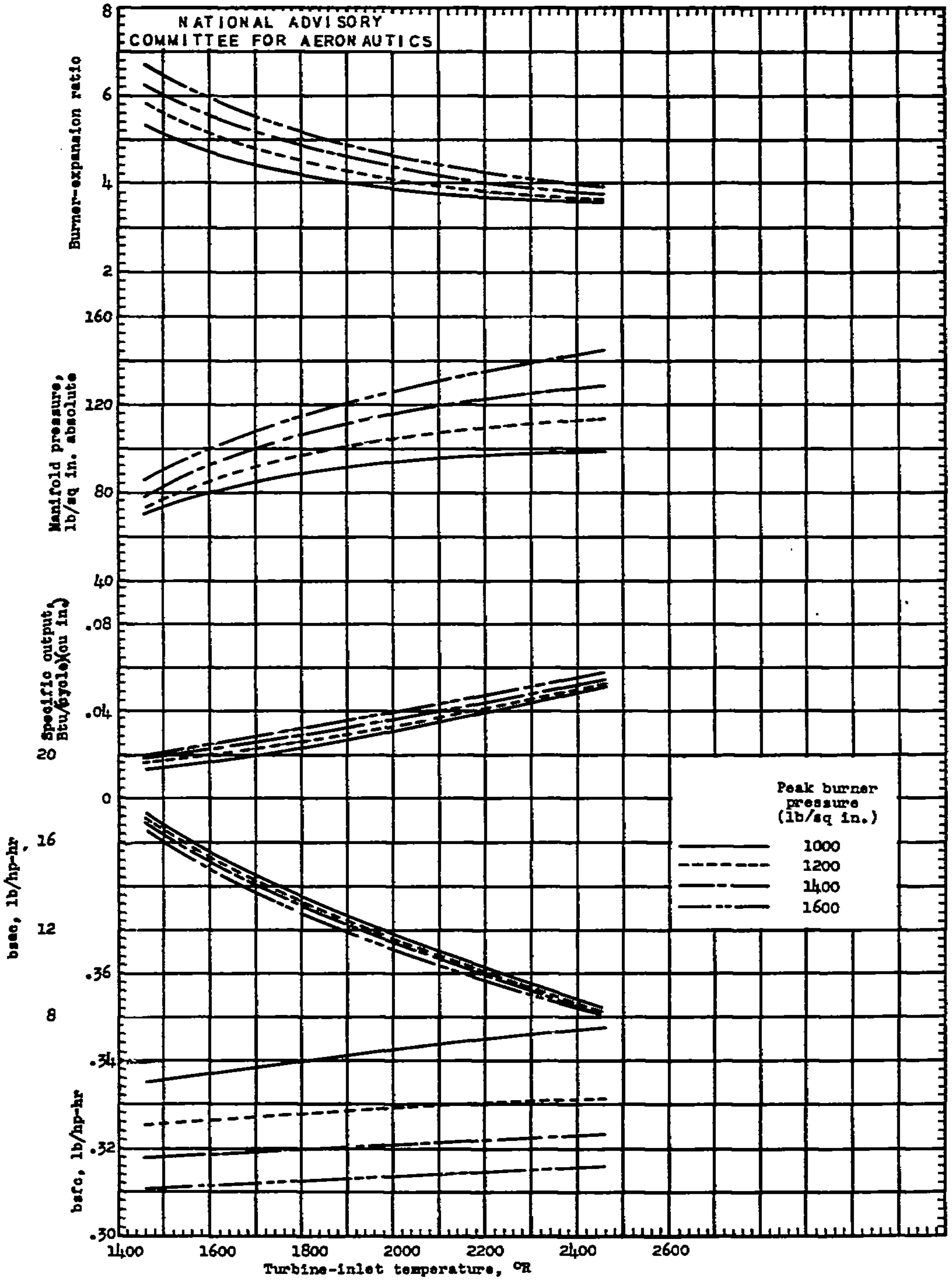

Figure 5. - Effect of engine operating limits on calculated performance of gas-generator engine. Altitude, 20,000 feet; compressor efficiency, 0.85; scavenge ratio, 1.00.

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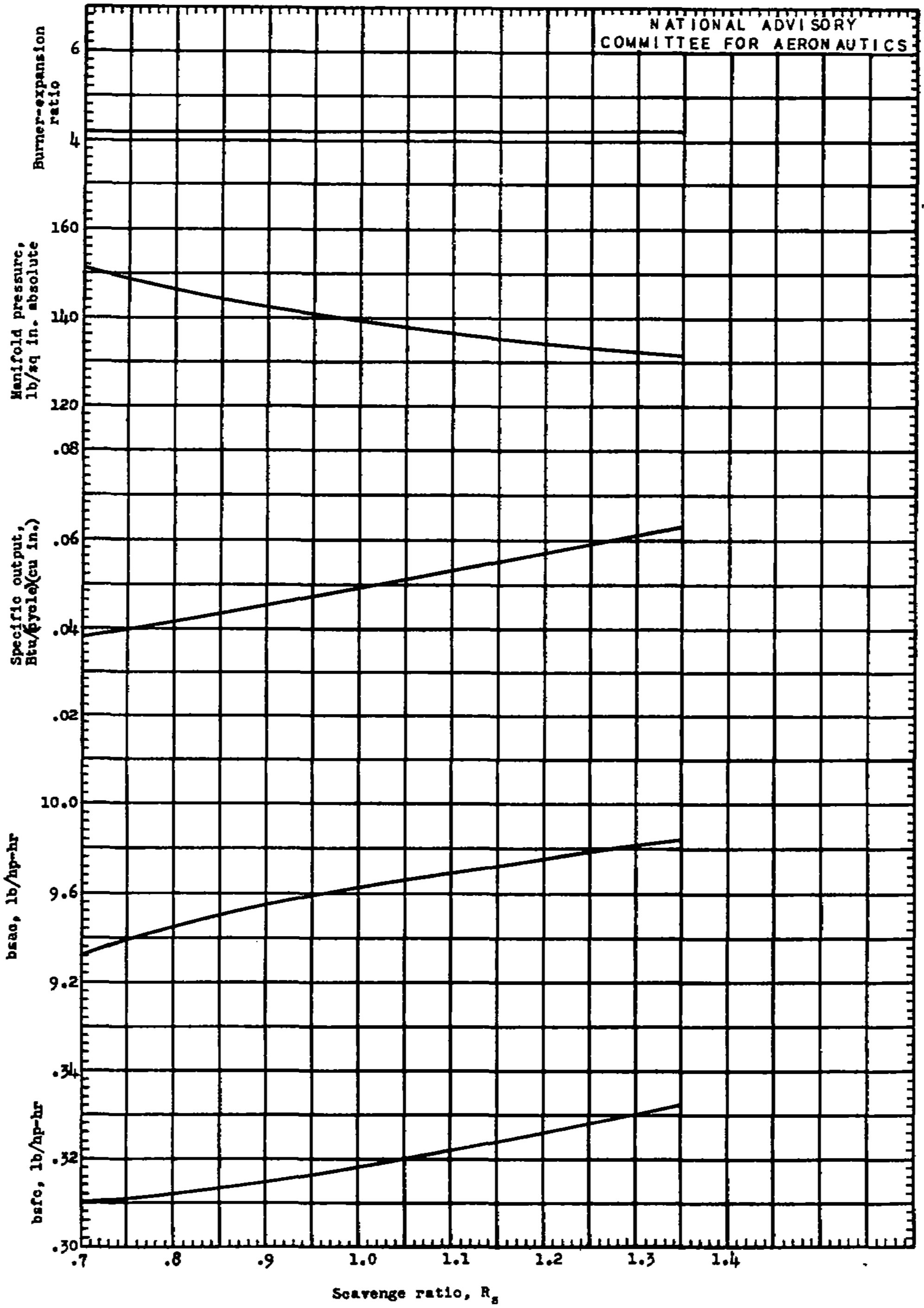

Figure 6. - Effect of change in scavenge ratio on calculated performance of gas-generator engine. Altitude, 20,000 feet; peak burner pressure, 1600 pounds per square inch; turbine-inlet temperature, 2260° R; compressor efficiency, 0.85.

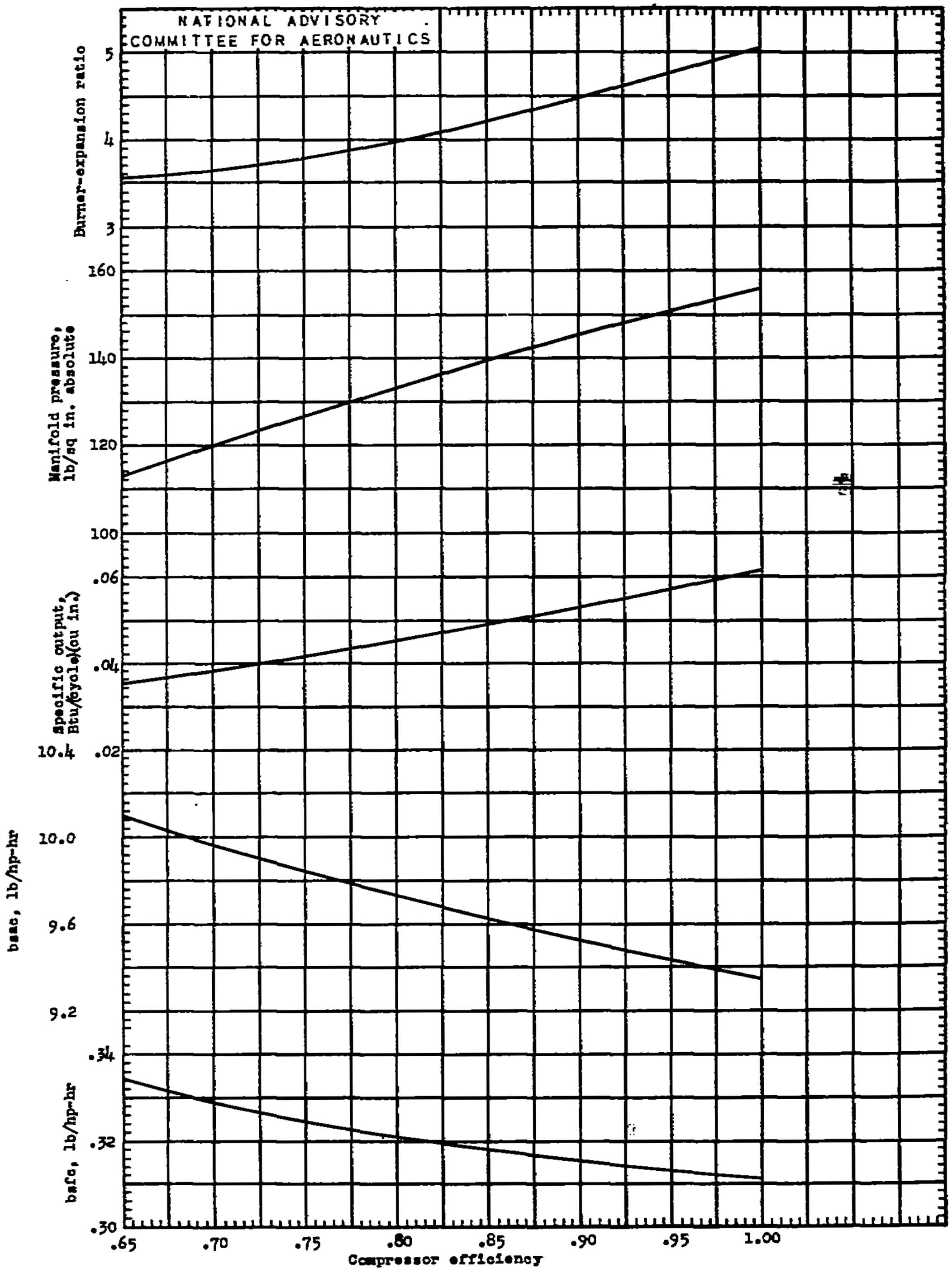

Figure 7. - Effect of change in compressor efficiency on calculated performance of gas-generator engine. Altitude, 20,000 feet; peak burner pressure, 1600 pounds per square inch; turbine-inlet temperature, 2260° R; scavenge ratio, 1.00.

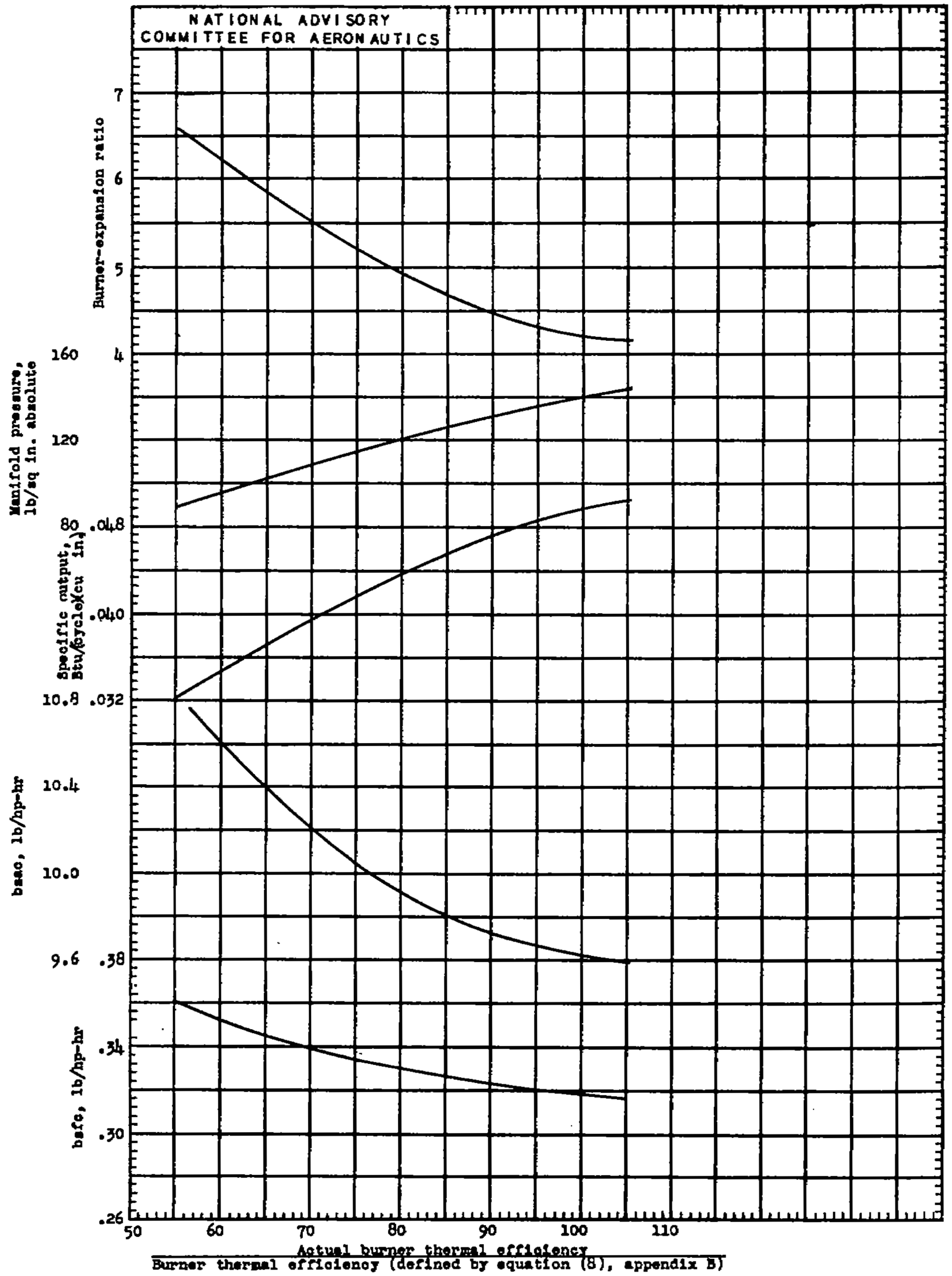

Figure 8. - Effect of change in burner thermal efficiency on calculated performance of gas-generator engine. Altitude, 20,000 feet; peak burner pressure, 1600 pounds per square inch; turbine-inlet temperature, 2260° R; scavenge ratio, 1.00; compressor efficiency, 0.85.

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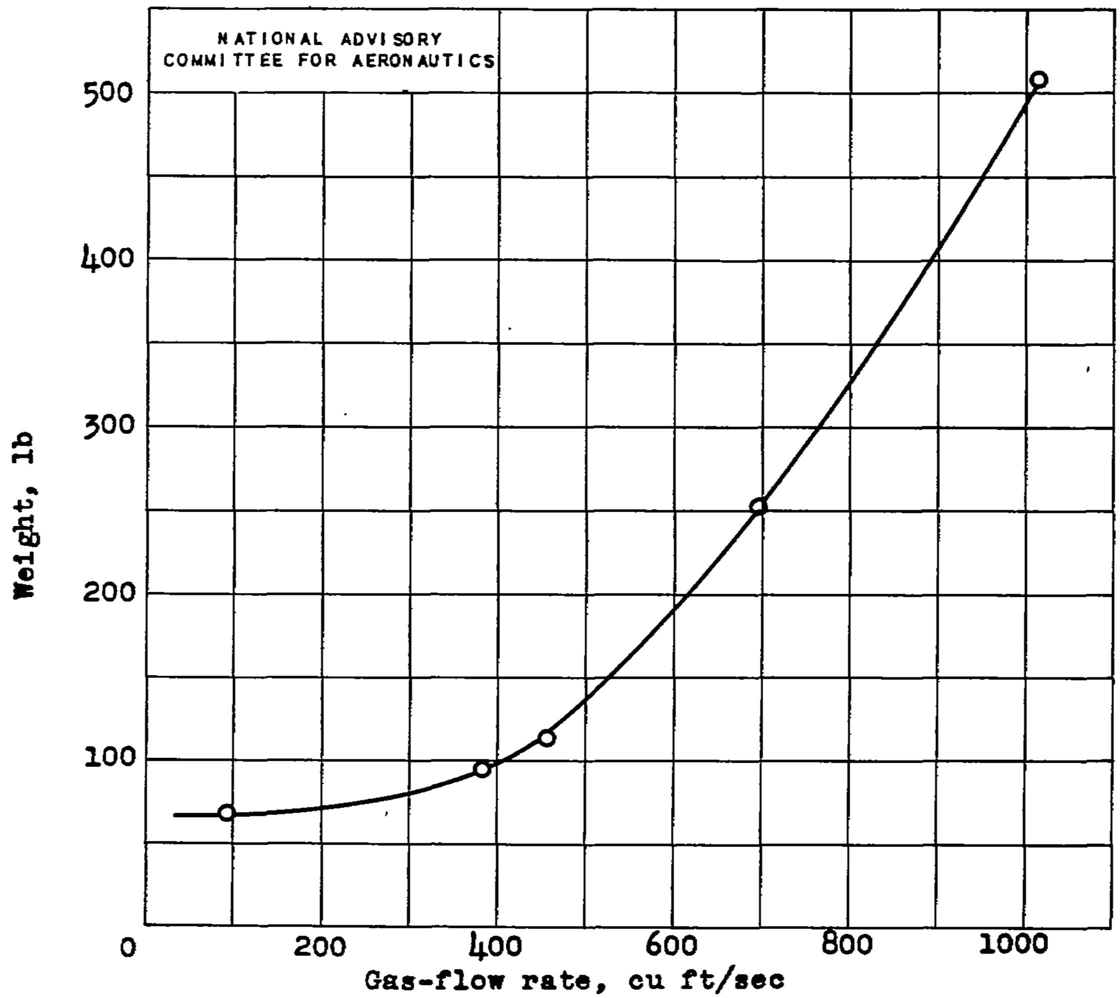

Figure 9. - Turbine weight as function of gas-flow rate.