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A THEORETICAL ANALYSIS OF THE PERFORMANCE OF A DIESEL ENGINE-COMPRESSOR-TURBINE COMBINATION FOR AIRCRAFT

By Eldon W. Hall

Aircraft Engine Research Laboratory
Cleveland, Ohio

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ADVANCE CONFIDENTIAL REPORT

A THEORETICAL ANALYSIS OF THE PERFORMANCE OF A DIESEL

ENGINE-COMPRESSOR-TURBINE COMBINATION FOR AIRCRAFT

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SUMMARY

A theoretical study was made of a Diesel engine-compressor-turbine combination for aircraft application. The performance characteristics of the compressor and the turbine chosen for the analysis are believed to be attainable on present aircraft equipment. Maximum cylinder pressure, maximum exhaust-gas temperature, and maximum engine speed were limited to values now obtained on spark-ignition engines. The analysis indicates favorable performance characteristics for the proposed power plant for the following combinations of conditions:

(a) Extremely high engine inlot-manifold and exhaust-manifold pressures

(b) Very lean fuel-air ratio

(c) Low engine compression ratio compared with conventional Diesel engines to reduce the maximum cylinder pressure to a safe value

(d) Intercooling between the compressor and the engine

(e) Use of turbine power in excess of that required to drive the compressor

The results indicate that it would be possible to obtain much more power per unit displacement volume with this combination than with a spark-ignition engine of the same maximum cylinder pressure and engine exhaust-gas temperature. When the proposed combination was assumed to operate under these conditions, the computed specific fuel consumption at maximum power was lower than that now obtainable with spark-ignition engines operating under a lean-cruise condition.

Because of the internal cooling provided by the large quantity of excess air in the proposed system, adequate engine cooling would be considerably easier to obtain than for a conventional Diesel engine. The weight per horsepower output of the combination was calculated to be less than that of a conventional spark-ignition aircraft engine.

INTRODUCTION

Considerable work is being done on the combination of the spark-ignition engine with a turbine and a compressor to increase the power-plant efficiency and performance.

The occurrence of fuel knock prevents a spark-ignition engine from operating at the high inlet-manifold and exhaust-manifold pressures and the lean mixtures that are necessary to obtain maximum performance. The Diesel engine is not limited by this fuel-knock condition and, when operated in conjunction with turbine and compressor, presents the possibility of considerably better performance than an equivalent system incorporating a spark-ignition engine.

The main objections to the application of the Diesel engine to airplanes have been the high specific weight and the large amount of external cooling required. Combinations of Diesel engine, compressor, and gas turbine have been proposed for marine use; one system already being successfully used is described in reference 1. The present analysis, made at the Aircraft Engine Research Laboratory of the NACA during the summer of 1943, is the study of a combination of similar components to be applied specifically to aircraft and reveals how the system would eliminate most of the serious difficulties involved in the use of a Diesel engine alone.

The purpose of the analysis in this report is to show the performance that may be possible with a current aircraft spark-ignition engine modified to operate on the Diesel cycle in combination with a compressor and a turbine while adhering to the current limitations on maximum cylinder pressure, engine speed, and exhaust-gas temperature. Curves are presented that show the performance for various engine inlet and exhaust pressures with and without the compressor and turbine geared to the engine shaft. The equations used in the analysis are derived in the appendix.

The values for the weights and the efficiencies of the components of the system used in the analysis are characteristic of current practice. The system may be built around components that have already reached advanced stages of development for aircraft use, and their adaptation to the proposed system should be a relatively simple problem.

ANALYSIS

In the first part of the proposed cycle the air is compressed in a supercharger, intercooled, and introduced into the engine at high pressure. In the engine it is further compressed to the maximum cylinder pressure; the fuel is introduced, burned at constant pressure, and the mixture is then expanded. The compression ratio of the engine is so computed as to provide the desired maximum cylinder pressure at the end of the compression stroke. At the end of the expansion stroke the gases are exhausted at a high back pressure into the turbine and thence to the atmosphere. The amount of fuel introduced is limited by the condition that the exhaust-gas temperature does not exceed a value considered safe for turbine operation. For additional power the exhaust gas could be discharged rearwardly from the turbine to obtain jet thrust.

When the engine back pressure or the turbine inlet pressure is made equal to the engine inlet pressure, the turbine power is normally greater than the compressor power; for this case, the coupled turbine and compressor are connected by a gear-reduction unit to the engine shaft, and the excess power is added to the engine power for driving the propeller. If the engine back pressure is less than the engine inlet pressure and is of such value that the turbine and the compressor powers are equal, it would not be advantageous to gear the turbine and compressor to the engine shaft and these two components become a turbosupercharger. The loss in turbine power in this case is partly compensated by the increase in engine power that results from the reduction in back pressure. An optimum back pressure exists at which the net power is a maximum.

With additional engine modifications, the system can be operated on a two-stroke cycle. With this type of operation, the engine back pressure should be less than the engine inlet pressure to permit scavenging of the engine cylinders. Two-stroke-cycle operation, which is well suited for the Diesel cycle because no loss of fuel occurs during the scavenging process, further increases the power output obtainable with this system.

The cycle for the proposed system is shown diagrammatically in figure 1. The air is first compressed from points 1 to 2 in a turbosupercharger, intercooled from points 2 to 3, and introduced into the engine under high pressure at point 3. In the engine it is further compressed to the maximum cylinder pressure at point 4 where fuel is injected and combustion is started. The main part of the combustion process occurs from points 4 to 5 at constant pressure.

The rest of the combustion occurs during the polytropic expansion from points 5 to 6. At the end of the expansion stroke (point 6) the pressure is dropped to the engine back pressure at point 7, which may be either above or below the engine inlet pressure, depending on the power desired in the turbine. Part of the gases remaining in the cylinder are then exhausted from the engine to point 8 and the pressure of residual gases is changed to the engine inlet pressure (point 9) when the engine inlet valve is opened. The gases leaving the engine are irreversibly expanded to the turbine inlet pressure at point 10; the temperature at this state is insured safe for turbine operation by limiting the quantity of fuel introduced during the burning phase in the engine. The gases are then expanded through the turbine to the initial air pressure at point 11. The combustion during the expansion process from points 5 to 6 is undesirable but is introduced to simulate more closely actual engine performance.

In order to show more clearly the possibilities of the proposed system, an aircraft engine with a displacement of 1820 cubic inches and modified to operate as a Diesel engine in the suggested manner was considered. The conditions applied in the calculations of the performance of the system were (1) that the maximum cylinder pressure did not exceed a value considered safe for current spark-ignition engines and (2) that the exhaust-gas temperature did not exceed a safe value for turbine operation.

The following are the assumed operating conditions:

Maximum cylinder pressure p_4, pounds per square inch	1200
Maximum temperature at turbine inlet T_{10}, °F absolute	2260
Engine displacement V_d, cubic inches per cycle	1820
Engine speed N, rpm	2400
Compressor efficiency η_c, percent	70
Turbine efficiency η_t, percent	70
Engine mechanical efficiency η_e, percent	87.5
Reduction-gear efficiency η_g, percent	90
Intercooler effectiveness ϵ	0.60
Exponent n for compression in engine	1.42
Exponent n' for expansion in engine	1.20
Engine weight W_e, pounds	1375
Reciprocating-compressor weight (for air injection of fuel) W_{rc}, pounds	230
Specific weight of each auxiliary component:	
Turbine w_t, pound per horsepower	0.30
Compressor w_c, pound per horsepower	0.20
Reduction gears w_g, pound per horsepower	0.25

The values of the polytropic exponents n and n' were chosen after a study was made of indicator diagrams taken on a Diesel engine (unpublished data on the modified NACA universal test engine described in reference 2 taken at Langley Field, Va., in 1931). The expansion exponent of 1.20 indicates the existence of further combustion after injection is completed; in the calculation of the fuel consumption, the fuel burned during this process is added to the fuel burned during the constant-pressure process. A Diesel fuel, dodecane, was assumed to be used with an effective heating value of 19,450 × (1 - F/A) Btu per pound in the range under consideration when determined by the method outlined in the appendix, where F/A is the fuel-air ratio. This method accounts for the variation in the composition and the specific heats of the mixture throughout the cycle. The constant-volume lower heating value of dodecane was taken as 19,150 Btu per pound at 60° F.

The compressor and turbine efficiencies listed are characteristic of modern compressor and turbine performance. Higher efficiencies, when they are obtained, will yield greater net powers than stated in this report. The weights of the components are also representative of present equipment and are believed to be conservative.

The density of the fresh charge in the cylinder before it was mixed with the residual gas was assumed to be equal to the density in the inlet manifold. This assumption can be made inasmuch as the errors that are introduced will be compensated for by the air which is added during the fuel-injection period; this injected air was not considered in the computations. If liquid-fuel injection is used or if the engine operates with a low volumetric efficiency, the power output of the system will be reduced according to the decrease in air flow.

The method of analysis, in general, was to vary the inlet and the exhaust pressures of the engine and to compute in each case the powers of the various components, the fuel consumption, and the over-all weight, subject to the conditions listed in this section. The net power was taken as equal to the engine power plus the difference between the turbine and the compressor powers multiplied by the reduction-gear efficiency. The equations used in this computation are included in the appendix.

DISCUSSION

Figure 2 shows power, specific weight, specific fuel consumption, engine compression ratio, and fuel-air ratio at various engine inlet pressures for the illustrative case at sea level in which the turbine and the compressor are geared to the engine shaft and the engine inlet pressure is equal to the engine exhaust back pressure. A turbine and a compressor, each with an efficiency of 70 percent, are assumed to give the desired engine inlet pressure. In order that the maximum cylinder pressure of 1200 pounds per square inch will not be exceeded, the engine compression ratio must be decreased as the engine inlet pressure is increased. The loss in efficiency because of this reduction in the compression ratio of the engine is partly compensated for by the additional compression and expansion in the turbine and the compressor, and the net change in specific fuel consumption is small.

As the engine inlet and exhaust pressures are increased, the net power output increases in the range shown in figure 2. When the inlet pressure is increased above 4 atmospheres (all atmospheres are considered to be at sea level), the net power increases only slowly, whereas compressor and turbine powers increase more rapidly. From practical considerations, it does not appear feasible to operate at engine inlet pressures much higher than 4 atmospheres. If the turbine and the compressor efficiencies are increased considerably above 70 percent, the gain in net power and efficiency at pressures above 4 atmospheres may be sufficient to warrant the additional difficulty of operating at those higher engine inlet pressures.

At an engine inlet pressure of 4 atmospheres, the specific weight of the power plant is reduced to 1.00 pound per horsepower, which is 0.25 pound per horsepower less than that of the original spark-ignition engine around which the proposed system is assumed to be built.

The minimum specific fuel consumption is obtained at an engine inlet-manifold pressure of 1 atmosphere, which corresponds to no supercharging. The engine compression ratio in this case is 22 and the fuel-air ratio approaches the value for the theoretical mixture. It is very doubtful whether most of the fuel could be burned at constant pressure in practice and it would be necessary to reduce the compression ratio below that given to prevent exceeding the maximum allowable cylinder pressure. This reduction in compression ratio would be accompanied by a corresponding increase in specific fuel consumption.

The fuel-air ratio decreases as the engine inlet pressure is increased and, at a pressure of 4 atmospheres, the system operates with considerable excess air. The clearance volume of the proposed engine with a high inlet pressure must be much greater, for a given maximum cylinder pressure, than the clearance volume of the conventional Diesel engine operating on a lower inlet pressure; consequently, the rate of volume change during expansion in the engine is lower for the same engine speed and constant pressure may be maintained with a lower rate of combustion than for a conventional Diesel engine. This lower required rate of combustion allows a more complete burning during the constant-pressure process that occurs during a greater portion of the expansion stroke and a more nearly adiabatic expansion during the rest of the stroke. It is therefore believed that, at an inlet pressure of 4 atmospheres, the large amount of excess air, the high air density, and the low required rate of combustion are favorable for the attainment of constant-pressure combustion. The combination of large powers and low specific fuel consumptions shown by figure 2 when compared with performance values of conventional aircraft engines indicates that, even if constant-pressure combustion is not completely achieved, the compound engine under discussion may still have attractive performance.

Although the elimination of the intercooler would result in considerable loss of maximum power, a gain in efficiency would be effected when the turbine and the compressor are geared to the engine shaft, as indicated by the following table calculated for engine inlet-manifold and exhaust-manifold pressures of 4 atmospheres for sea-level operation.

	With intercooler	Without intercooler
Net horsepower output	2054	1413
Engine horsepower	1779	1205
Compressor horsepower	616	465
Turbine horsepower	922	696
Specific fuel consumption, lb/bhp-hr	0.330	0.296
Specific weight, lb/bhp	1.00	1.39

In the weight calculations throughout the report, no allowance has been made for the weight of the intercooler. At an engine inlet pressure of 4 atmospheres at sea level, the weight of the intercooler is estimated at less than 0.04 pound per horsepower output.

At high altitudes the performance given in figure 2 could be obtained by an additional turbosupercharger and intercooler for maintaining sea-level atmospheric conditions at the main compressor inlet and the main turbine exhaust. The power in the exhaust gas issuing from the main turbine would be more than enough to operate the auxiliary turbosupercharger.

Operation with equal engine inlet-manifold and exhaust-manifold pressures does not necessarily represent an optimum condition. Figure 3 shows the performance characteristics of the system operating at an inlet pressure of 4 atmospheres when the exhaust back pressure is varied. At both sea level and 30,000 feet, little difference in net power output exists between the case in which turbine and compressor powers are equal and the case in which engine inlet and exhaust pressures are equal and the turbine and the compressor are geared to the engine shaft. Decreasing the back pressure to a value that results in equal turbine and compressor powers increases the specific fuel consumption but decreases the specific weight of the system.

In figure 4 is shown the performance when the turbosupercharger is not geared to the engine shaft and the engine inlet pressure is allowed to vary. Although the turbine power is lower than that shown in figure 2, this loss is partly balanced by a gain in engine power resulting from the reduced back pressure. This type of operation offers the practical advantage of eliminating the gearing between the turbosupercharger and the engine and the system becomes more flexible; the principal difference is that the specific fuel consumption is slightly higher.

Examination of the curves in figures 2 to 4 indicates that theoretically more than 2000 horsepower may be expected from an engine with a displacement of 1820 cubic inches when operated on a Diesel cycle in the proposed manner without exceeding the safe maximum cylinder pressure for current engines or the safe exhaust-gas temperature for current turbines. This power is obtained with a specific fuel consumption of 0.330 pound per horsepower-hour and a net specific weight of 1.00 pound per horsepower when the engine is operated at an inlet pressure of 4 atmospheres and with a fuel-air ratio of 0.037. The maintenance of safe cylinder temperatures is favored by the low fuel-air ratio.

If an excess amount of scavenging air is used for cooling the engine cylinder and for reducing the exhaust-gas temperatures for safe turbine operation, it may be possible to burn more fuel and generate more power in the engine. No appreciable reduction in efficiency will result because most of the power for compression of the excess scavenging air will be realized in the turbine. Even further increases in power and reductions in weight could be expected if the engine were operated on a two-stroke cycle to which this system readily lends itself.

For part-load operation several methods of varying the power output are possible. The power may be reduced by decreasing the engine speed, the engine inlet-manifold pressure, or the fuel-air ratio. These methods require that one or more of the following items be adjusted: the turbine-nozzle area, the compressor Q/N (volume flow per rpm) for efficient operation, or the relation between the turbine, the compressor, and the engine speeds. Additional study is required to reveal the method of power control that provides maximum efficiency without excessive mechanical complication.

For fairly high airplane speeds, a large amount of additional power may be available from exhaust jet propulsion. Because of the high turbine-nozzle-box pressure and the increased amount of exhaust gases handled with the proposed system, a larger proportion of the power may be realized from jet thrust than from a conventional turbosupercharged engine. This added power would further reduce the specific fuel consumption.

The proposed power plant, in general, retains the desirable characteristics of a Diesel plant without the main objectionable features — high specific weight and large amount of external cooling — commonly associated with aircraft Diesel engines. The mechanical features of the proposed system will undoubtedly need some detailed study but, inasmuch as the system generally lends itself to the use of aircraft equipment that has already reached advanced stages of development, the problems of adaptation should be relatively simple.

CONCLUSIONS

In the proposed Diesel engine-compressor-turbine combination, based on an engine with a displacement of 1620 cubic inches, a maximum cylinder pressure of 1200 pounds per square inch, a maximum exhaust-gas temperature of 1800° F, an engine speed of 2400 rpm, equipped with an auxiliary turbine and compressor with efficiencies of 70 percent each, and with conservative assumptions regarding other operating conditions, the following performance characteristics may be obtained:

1. With an engine inlet pressure of 4 sea-level atmospheres, the net power output of the system would be approximately 2000 horsepower at a lower specific fuel consumption than is now obtained in the lean-cruise condition on the conventional spark-ignition engine.

2. With four-stroke-cycle engine operation, the specific weight of the system would be reduced to 1 pound per horsepower with an engine inlet pressure of 4 sea-level atmospheres.

3. Increasing the engine inlet pressure would increase both the compressor and the turbine powers but, if this inlet pressure were increased beyond 4 sea-level atmospheres, only a small increase in net power would result at the expense of higher specific fuel consumption.

4. The fuel-air ratio required to provide 2000 net horsepower at an engine inlet pressure of 4 sea-level atmospheres is 0.037. Because of the large amount of excess air at this mixture the problem of cooling should be simpler than for the conventional Diesel engine at the same specific power.

5. Decreasing the engine back pressure below the inlet pressure to a value that makes the turbine and the compressor powers equal would increase the specific fuel consumption and reduce the turbine power and the over-all specific weight of the system, but the net power output would remain substantially the same.

6. The analysis shows, in general, that it would be advantageous to add a turbine and a compressor to a Diesel engine for handling large quantities of excess air above that needed for combustion even though a reduction in the engine compression ratio would be necessary to avoid excessive cylinder pressure. Because of the additional pressure ratios across the turbine and the compressor, the resulting system would have a high over-all compression ratio. The proposed system thus would have a high net efficiency but would be capable of handling larger quantities of air than a given conventional engine, which would result in a higher specific power output.

Aircraft Engine Research Laboratory,
National Advisory Committee for Aeronautics,
Cleveland, Ohio.

APPENDIX - DERIVATION OF EQUATIONS APPLICABLE TO THE PROPOSED CYCLE

Symbols

B ratio of volume of gases in the engine after combustion to volume before combustion for the same temperature and pressure, $\frac{R'M_T}{R\ M_e}$

c specific fuel consumption, lb/bhp-hr

C fuel consumption, lb/sec

c_p specific heat at constant pressure before combustion, 7.72 Btu/(°F)(slug)

c_p' specific heat at constant pressure after combustion, 9.60 Btu/(°F)(slug)

c_v' specific heat at constant volume after combustion, Btu/(°F)(slug)

F/A fuel-air ratio

h effective heat value of fuel, Btu/lb

i intercooler effectiveness

J Joule's constant, 778 ft-lb/Btu

M mass flow of air through the system, slugs/sec

M_e mass flow of residual gases and induction air in the engine, slugs/sec

M_f mass flow of fuel, slugs/sec

M_r mass flow of residual clearance gases, slugs/sec

M_T total mass flow of mixture and fuel in the engine, slugs/sec

n exponent for polytropic compression in the engine

n' exponent for polytropic expansion in the engine

N engine speed, rpm

p pressure, lb/sq in. absolute

P horsepower

R gas constant before combustion, 1718 ft-lb/(°F)(slug)

R' gas constant after combustion, 1724 ft-lb/(°F)(slug)

r engine compression ratio

T temperature, °F absolute

v specific volume, cu ft/slug

V volume, cu in./cycle

w specific weight, lb/bhp

W weight, lb

γ ratio of specific heats before combustion, 1.40

γ' ratio of specific heats after combustion, 1.30

η efficiency, percent

η_e engine mechanical efficiency, percent

Subscripts:

1 to 11 (points of the cycle (fig. 1)):

1 atmospheric air

2 after compression in compressor

3_i after intercooling

3 mixture before compression in the engine

4 after compression in the engine

5 end of constant-pressure combustion

6 end of expansion

7 in cylinder after blowdown

8 in cylinder before intake valve is opened

9 in cylinder after intake valve is opened

10 turbine inlet

11 turbine exhaust

a during constant-pressure combustion

b during polytropic expansion

c compressor

d displacement

e engine

f fuel

g reduction gears

N net

r residual gases

rc reciprocating compressor

t turbine

T total

The prime indicates the value during and after combustion.

Equations

The compressor power for a pressure ratio of p_2/p_1 is given by

$$\frac{P_c}{M} = \frac{Jc_pT_1}{550\eta_c}\left[\left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} - 1\right], \text{(bhp)(sec)/(slug)} \qquad (1)$$

and the temperature after compression by

$$T_2 = T_1 + \frac{550}{Jc_p}\left(\frac{P_c}{M}\right), \ ^{\circ}\text{F absolute} \qquad (2)$$

The temperature after intercooling with an intercooler of effectiveness i is

$$T_{3_i} = T_2 - i\,(T_2 - T_1), \ ^{\circ}\text{F absolute} \qquad (3)$$

For a maximum temperature at the turbine inlet T_{10} the formula for the turbine power is

$$\frac{P_t}{M} = \frac{\eta_t J c_p{}' T_{10}}{550}\left[1 - \left(\frac{p_1}{p_7}\right)^{\frac{\gamma'-1}{\gamma'}}\right], \ (\text{bhp})(\text{sec})/(\text{slug}) \qquad (4)$$

where p_7 is the engine exhaust back pressure. When the turbine and compressor powers are equal, the engine back pressure can be found from equation (4) by setting $P_t/M = P_c/M$ (equation (1)).

The clearance volume V_4 is related to the total volume V_3 or $V_4 + V_d$ by the equation

$$\frac{V_3}{V_4} = \left(\frac{p_4}{p_2}\right)^{1/n} = \frac{V_4 + V_d}{V_4} = 1 + \frac{V_d}{V_4} \qquad (5)$$

from which

$$V_4 = \frac{V_d}{(p_4/p_2)^{1/n} - 1}, \ \text{cu in./cycle} \qquad (6)$$

The residual gases left in the cylinder are compressed from pressure p_7 to p_2 by the fresh charge after the inlet valve is opened. The volume of the residual gas after this compression is given by the adiabatic formula

$$V_9 = V_4\left(\frac{p_7}{p_2}\right)^{1/\gamma'}, \ \text{cu in./cycle} \qquad (7)$$

where $V_4 = V_8$.

During the intake stroke the mass of air drawn into the cylinder for four-stroke operation is given by

$$M = \frac{(V_3 - V_9)\ p_2 N/2}{60 \times 12 \times R \times T_{3_i}}\text{, slugs/sec} \tag{8}$$

The initial mass during the engine cycle is

$$M_e = M + M_r\text{, slugs/sec} \tag{9}$$

and the total mass at the end of the expansion stroke is

$$M_T = M + M_r + M_f = M(1 + F/A) + M_r\text{, slugs/sec} \tag{10}$$

where the mass of residual gases M_r is

$$M_r = M_T\left(\frac{V_4}{V_3}\right)\left(\frac{p_7}{p_6}\right)^{1/\gamma'} = \frac{M\left(1 + \frac{F}{A}\right)\left(\frac{p_7}{p_6}\right)^{1/\gamma'}}{\frac{V_d}{V_4} + 1 - \left(\frac{p_7}{p_6}\right)^{1/\gamma'}}\text{, slugs/sec} \tag{11}$$

When the energy of the gases before blowdown and the energy of the gases after blowdown are equated

$$T_6 + \frac{144 v_6 p_7}{Jc_v'} = T_{10} + \frac{144 v_{10} p_7}{Jc_v'}$$

or

$$T_6 + \frac{144 v_6 (p_7 - p_6)}{Jc_p'} = T_{10}$$

Then

$$T_{10} = T_6 - \frac{144(p_6 - p_7)v_6}{Jc_p'} = T_6 - \frac{(p_6 - p_7)R'T_6}{p_6 Jc_p'} \tag{12}$$

Since $V_6 = V_3$ (13)

$$T_6 = \frac{T_3}{B}\left(\frac{p_6}{p_2}\right)\text{, °F absolute} \tag{14}$$

where

$$B = \frac{M_T}{M_e} \times \frac{R'}{R} = \frac{M(1 + F/A) + M_r}{M + M_r} \times \frac{R'}{R}$$

For a very close approximation (applicable to lean mixtures)

$$B = 1 + F/A$$

Solution of equations (12) and (14) for p_6 gives

$$p_6 = p_2 \gamma' \left[\frac{BT_{10}}{T_3} - \frac{\gamma' - 1}{\gamma'}\left(\frac{p_7}{p_2}\right)\right], \text{ lb/sq in.} \qquad (15)$$

Because the gases remaining in the cylinder at the end of the expansion stroke are adiabatically expanded from p_6 to p_2,

$$T_9 = T_6\left(\frac{p_2}{p_6}\right)^{\frac{\gamma'-1}{\gamma'}}, \text{ }^{\circ}\text{F absolute} \qquad (16)$$

The temperature of the mixture before compression T_3 is about

$$T_3 = \frac{MT_{3_i} + M_rT_9}{M_e}, \text{ }^{\circ}\text{F absolute} \qquad (17)$$

For an approximate value of T_3 equations (10), (11), (14), (16), and (17) give

$$T_3 = \frac{T_{3_i}}{1 + \frac{M_r}{M} - \frac{R(p_7/p_2)^{1/\gamma'}}{R'\left(\frac{V_d}{V_4} + 0.8\right)}}, \text{ }^{\circ}\text{F absolute} \qquad (18)$$

where M_r/M may be assumed to be $0.05p_7/p_2$. The temperature after compression T_4 will be given by

$$T_4 = T_3\left(\frac{p_4}{p_2}\right)^{\frac{n-1}{n}}, \text{ }^{\circ}\text{F absolute} \qquad (19)$$

Nearly all of the fuel is burned during constant-pressure expansion so that the temperature T_5 before polytropic expansion to give T_6 at the end of the stroke can be given by

$$T_5 = T_6\left(\frac{p_4}{p_6}\right)^{\frac{n'-1}{n'}}, \ ^{\circ}\text{F absolute} \tag{20}$$

The fuel consumption during the constant-pressure expansion is

$$C_a = \frac{M_T c_p'}{h}(T_5 - T_4), \text{ lb/sec} \tag{21}$$

and during the polytropic expansion

$$C_b = \frac{M_T}{h}\left(\frac{c_p'}{\gamma'}\right)\left(\frac{\gamma' - n'}{n' - 1}\right)(T_5 - T_6), \text{ lb/sec} \tag{22}$$

If an accurate value of the fuel consumption is desired, an effective heating value h, for use in equations(21) and (22), can be calculated from an analysis of the cycle involving variable values of c_p, γ, and R. An empirical equation applicable only to this analysis was found and, in the range under consideration for the illustrative case, h was found to be $19,450 \times (1 - F/A)$ Btu per pound for dodecane. The total fuel consumption will then be

$$C = C_a + C_b, \text{ lb/sec} \tag{23}$$

With the mass M from equation (8), the compressor and turbine powers can be found from equations (1) and (4), respectively.

The engine brake horsepower for the foregoing conditions is given by the following formula, which is a summation of the powers for each of the processes during the cycle:

$$\frac{P_e}{\eta_e} = \frac{R'M_e}{550}\left(\frac{R}{R'}BT_5 - T_4\right) + \frac{M_T R' T_6}{550(n' - 1)}\left[\left(\frac{p_4}{p_6}\right)^{\frac{n'-1}{n'}} - 1\right]$$

$$+ \frac{(p_2 - p_7)V_d N/2}{550 \times 60 \times 12} - \frac{M_e R T_3}{550(n - 1)}\left[\left(\frac{p_4}{p_2}\right)^{\frac{n-1}{n}} - 1\right], \text{ bhp} \tag{24}$$

The net power of the system will be

$$P_N = P_e + \eta_g (P_t - P_c), \text{ bhp} \tag{25}$$

The specific fuel consumption for the system becomes

$$c = \frac{C}{P_N} \times 3600, \text{ lb/bhp-hr} \tag{26}$$

and the fuel-air ratio is given by

$$F/A = \frac{C}{32.2M} \tag{27}$$

The compression ratio of the engine can be obtained from equations (5) and (6) and is

$$r = \frac{V_3}{V_4} = 1 + \frac{V_d}{V_4} \tag{28}$$

The specific weight of the system can be found from the weights of the component parts, which are given by the following formulas:

$$W_t = w_t \times P_t \text{ , lb}$$

$$W_c = w_c \times P_c \text{ , lb}$$

$$W_g = w_g \times P_g \text{ , lb}$$

$$W_e = w_e' \times P_e' \text{ , lb}$$

where P_e' and w_e' are the horsepower and the specific weight, respectively, of the original engine around which the system is built. If air injection is used, the weight of the reciprocating compressor and the parts necessary for the injection of fuel W_{rc} may be assumed proportional to the engine weight. The total weight of the system will be

$$W_T = W_e + W_t + W_c + W_g + W_{rc} \text{ , lb}$$

and the total specific weight

$$w_T = W_T/P_N \text{ , lb/bhp}$$

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1. Hammar, H. G., and Johansson, E.: Thermodynamics of a New Type of Marine Machinery: Combustion Engines with Pneumatic Power Transmission. Trans. Inst. Marine Eng. (London), vol. LI, no. 4, May 1939, pp. 139-157.

2. Gerrish, Harold C., and Voss, Fred: Influence of Several Factors on Ignition Lag in a Compression-Ignition Engine. NACA TN No. 434, 1932.

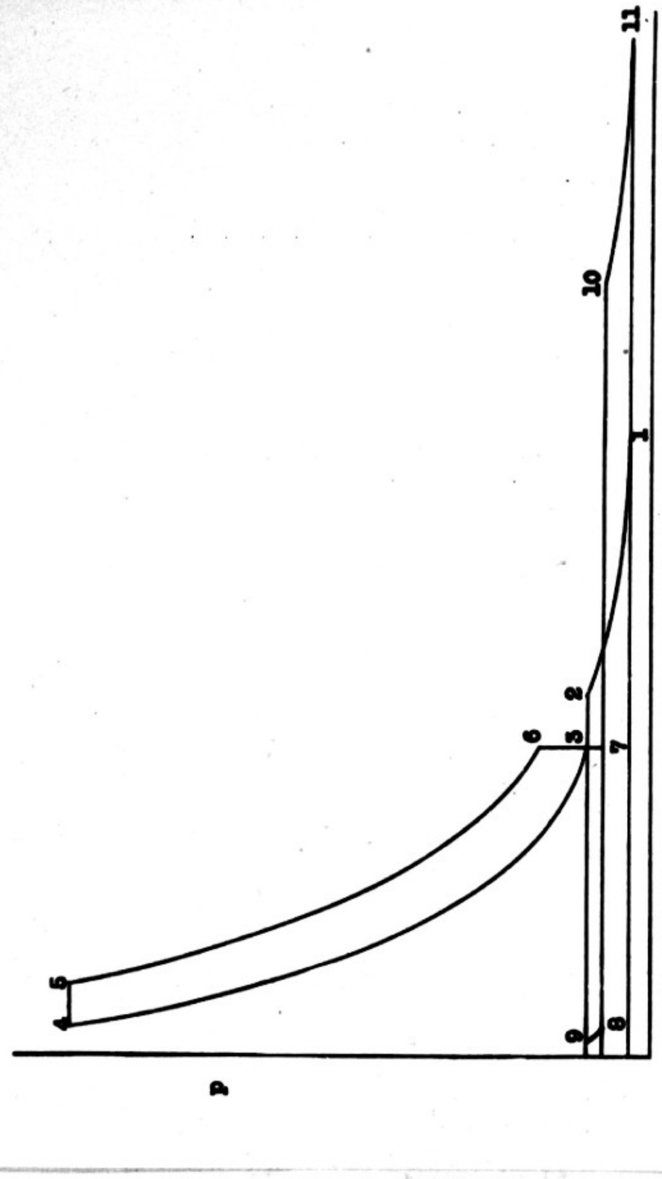

Figure 1.- Cycle of Diesel engine-turbine-compressor system.

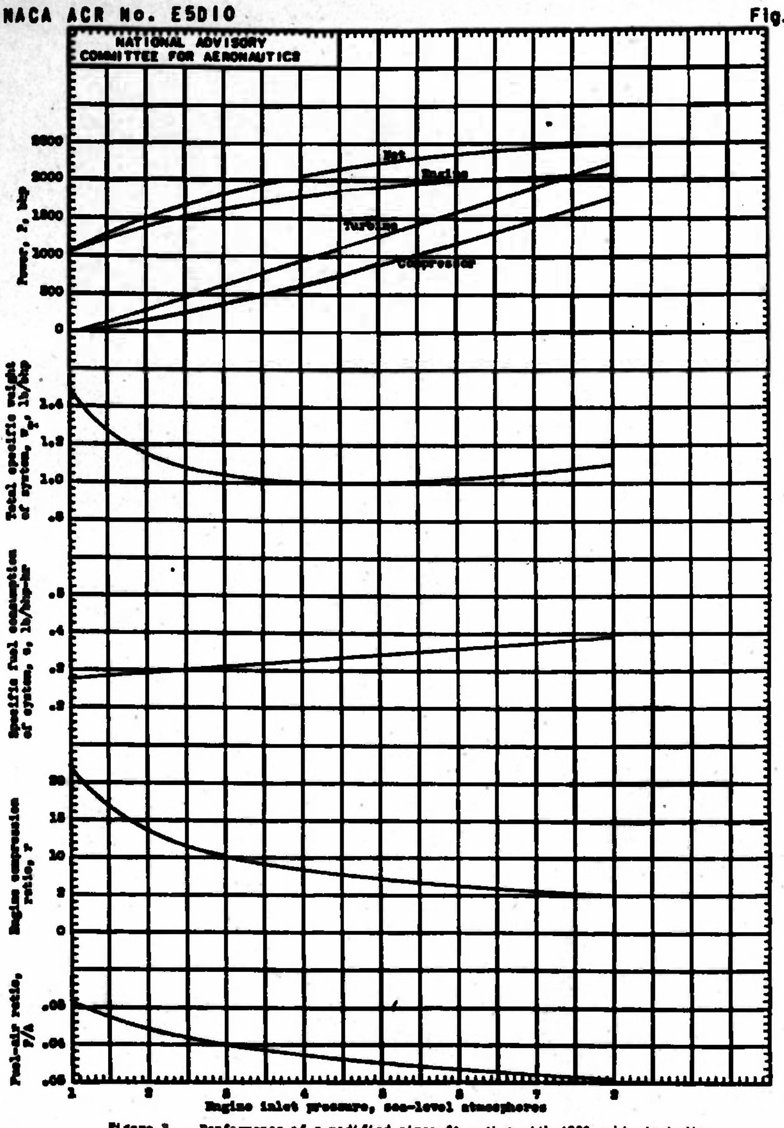

Figure 2. - Performance of a modified aircraft engine with 1820 cubic-inch displacement equipped with geared turbine and compressor and fuel-injection equipment for sea-level operation with engine exhaust pressure equal to engine inlet pressure.

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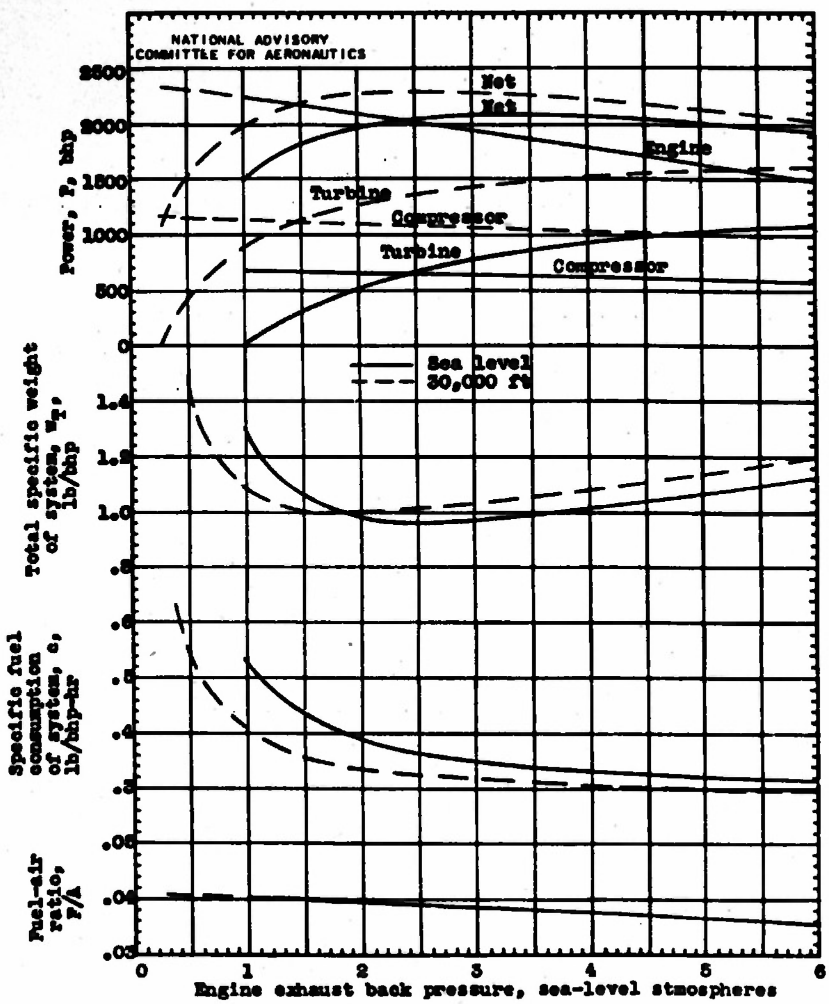

Figure 3. - Performance of a modified aircraft engine with 1820 cubic-inch displacement equipped with geared turbine and compressor and fuel-injection equipment with engine inlet pressure equal to 4 sea-level atmospheres.

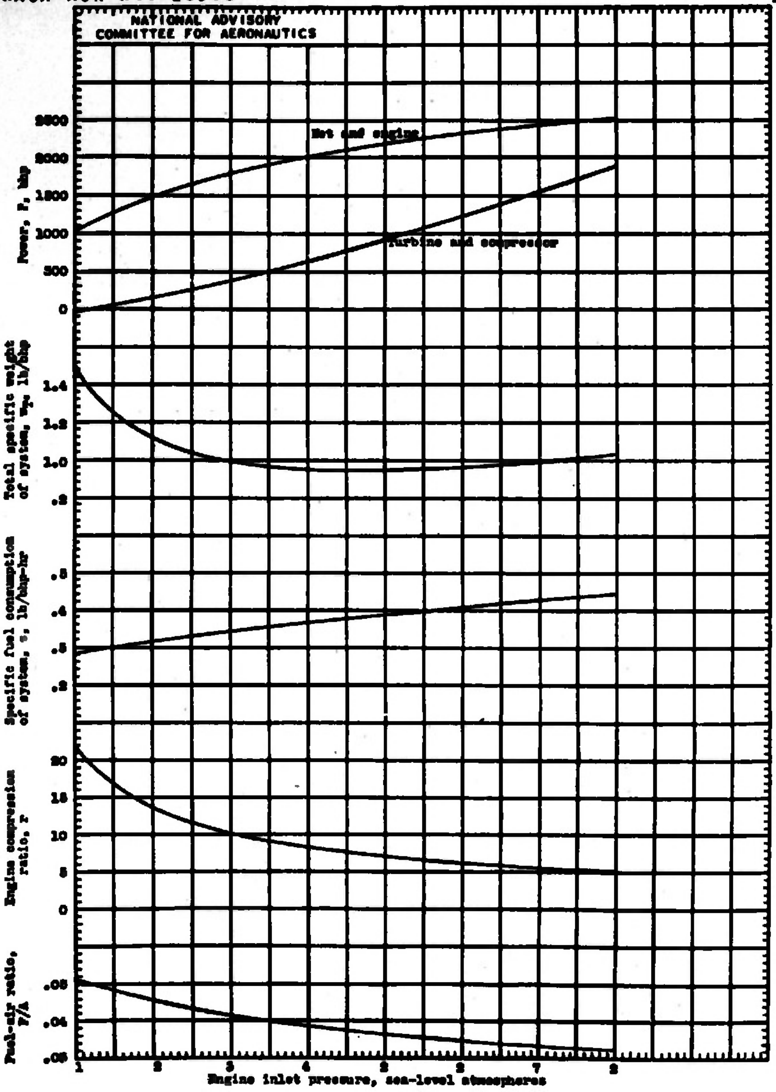

Figure 4. - Performance of a modified aircraft engine with 1820 cubic-inch displacement equipped with turbosupercharger and fuel-injection equipment for sea-level operation with turbine power equal to compressor power.

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AUTHOR(S): Hall, Eldon W.

ORIGINATING AGENCY: National Advisory Committee for Aeronautics, Washington, D. C.

PUBLISHED BY: (Same)

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ABSTRACT:

The analysis shows possible performance of an aircraft engine modified to operate on Diesel cycle combined with a compressor and turbine, while still adhering to present limitations on maximum cylinder pressure, engine speed, and exhaust gas temperature. With inlet pressure of 4 atm, power output of engine would be 2000 hp with low specific fuel consumption and fuel/air ratio of 0.037. Because of large excess of air in mixture, cooling should be greatly simplified.

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