THE UNIVERSITY OF MICHIGAN
COLLEGE OF ENGINEERING
Department of Mechanical Engineering

Technical Report

THE MODERN COMPOUND ENGINE

E. T. Vincent

ORA Project 05847

under contract with:

U. S. ARMY TANK-AUTOMOTIVE CENTER
PROPULSION SYSTEMS LABORATORY
CONTRACT NO. DA-20-018-AMC-0729-T
WARREN, MICHIGAN

administered through:

OFFICE OF RESEARCH ADMINISTRATION ANN ARBOR

September 1965

This report was prepared as a paper to be presented at the SAE Annual Meeting, Detroit, Michigan, January 1966.

TABLE OF CONTENTS

Page

ABSTRACT iii

LIST OF FIGURES iv

LIST OF TABLES v

OBJECT vi

1. THE COMPOUND CYCLE 2

2. COMPOUNDED ENGINES 7
    - 2.1 Compound Cycle Achievements 9
    - 2.2 Napier Nomad Aircraft Diesel 10
    - 2.3 The Orion Tank Engine 10
    - 2.4 Curtis-Wright Aircraft Engine 11
    - 2.5 Discussion of Past Achievements 11
    - 2.6 Four-Cycle Compound Engine 13
        - 2.6.1 Simple Compound Engine 14
        - 2.6.2 Compound Engine with Turbine Developing the Power Output 17
        - 2.6.3 Compound Engine with Turbine as Power Generator with Bypass Air 18
        - 2.6.4 Compound Unit in Which the Turbine Can Operate Separately for Emergency Starting, Auxiliary Purposes, etc. 22
        - 2.6.5 Indirect Compounding 25
            - 2.6.5.1 Cylinder Cooling 26
            - 2.6.5.2 Transmission Cooling 27

3. DISCUSSION 29

4. CONCLUSIONS 31

5. ACKNOWLEDGMENTS 32

6. REFERENCES 33

ABSTRACT

This report discusses possible improvements in compression ignition engine performance characteristics by greater utilization of the excess energy of exhaust gases of a typical engine. It also considers complications involved in this increased utilization as well as resulting differences in cost.

Various direct and indirect methods of achieving results in these areas are suggested. The almost necessary use of variable geometry in the turbine unit if any real progress is to be made and the requirements involved in its use are also discussed.

LIST OF FIGURES

Figure

1. Simple compound engine.

2. Compressor map.

3. Compound engine turbine develops output.

4. Compound engine turbine as power generator with engine bypass.

5. Compound engine with turbine starter.

6. Indirect cooling of transmission.

7. Relative torque curves with variable geometry.

LIST OF TABLES

Table

I. Performance of Simple Compound Engine

II. Performance of Compound Engine, Turbine Developing all Output

III. Performance with a Balanced Engine and Compressor

IV. Performance of Compound Engine with an Engine Bypass Circuit

V. Emergency Output with Turbine Unit Operating as Power Plant

VI. Performance at Slow Speed with Declutched Charger

VII. Engine Cooling with Available Exhaust Gas Energy

VIII. Indirect Cooling of Transmission

IX. Indirect Cooling of Transmission

OBJECT

The object of this report is to present a preliminary analysis of work to improve the performance of typical high-duty compression ignition engines for military purposes.

The possible improvement of low-speed torque capabilities through more efficient use of the available energy of the engine exhaust gases has been investigated. Necessary changes in present systems are indicated in the report.

THE MODERN COMPOUND ENGINE

The compound internal combustion engine has been of considerable interest to engineers for many years. Its origin or the origin of the principle of its operation could be considered to coincide with Ackinson's invention of the cycle which bears his name. With this invention, he attempted to achieve complete expansion of the gases to atmospheric pressure and recover the maximum available energy, the major objective of a modern compound unit.

To conform with some modern ideas of what constitutes a compound engine, this concept requires some amplification. A definition of what is to be understood by a compound engine in this paper and according to Army requirements is given. It is believed that there should be at least two definitions:

1. A compound internal combustion engine is one in which complete expansion of the working fluid to atmospheric pressure is attempted in one way or another, with the object of producing the maximum available energy from the overall cycle of operations.
2. Compounded engine systems are those in which more than one type of internal combustion engine are joined together to produce a single power output, the object being to employ the two or more units singly or together to achieve a desired spectrum of power outputs, fuel consumption, torques, etc.

To differentiate between these two types of units, the first will be referred to as a thermodynamically compound unit or a compound cycle. This unit corresponds to the steam compound engine having HP and LP cylinders in series, or HP and LP turbines with the same objective, viz., maximum power and minimum fuel consumption. It is true that in the internal combustion version there may not be HP and LP cylinders as such, but the gases pass through two mechanisms in series achieving the same principle.

The second type of unit can be referred to as a compound engine since more than one separate engine is involved. Each engine of the units can operate separately as a power plant irrespective of the other unit, i.e., the cycles are independent of each other but are compounded into a single output when necessary. The purpose of the compounding is to provide emergency power when required, to obtain high output in a small space, to fulfill certain torque requirements, etc., and is not associated with the achievement of maximum availability of any given fuel added to a particular cycle.

As examples of these two principles, some of the early work in this field can be quoted illustrating the principles involved and dating some of the early experiments in this field.

1. THE COMPOUND CYCLE

Although there have been one or two modern attempts to apply Atkinson's cycle, in slightly different forms, to problems such as charge cooling, fuel economy at part load with a spark ignition engine, etc., we will not discuss these at present. Instead, let us look at definite compound cycle engines.

The earliest compound cycle engine with which the writer had any connection was a 1914-1918 spark ignition engine by Ricardo which had both HP and LP cylinders with a transfer valve between them. The gases worked at high pressure in the HP cylinder, then continued their expansion in the LP cylinder reaching the lowest possible pressure. This engine was also supercharged to what would be considered a high value today. This, then, is the concept of the definite compound cycle: increasing pressure throughout the first cylinder to make recovering energy from the second worthwhile. Both factors tend to improve the availability of the cycle.

The disadvantage of this approach is that, with a reciprocating mechanism, frictional losses amount to a high percentage of the mean pressure of a naturally aspirated engine. These losses probably amount to about 25 to 35 psi mean pressure of the 130 to 165 psi in a four-cycle engine of this type. The degree of loss would also depend upon the type of cycle—spark ignition or compression ignition—and upon the means by which gases from one cylinder are to be expanded in another perhaps three or four times larger one. Despite the fact that the gas pressures in it are much lower, it cannot be expected that the frictional losses of this LP cylinder will be any less than those of the small HP cylinders. Thus, if the low pressure cylinder has an indicated mep of 25 psi, there is the possibility of a net loss of engine power by the addition of the LP portion of the cycle, as well as the introduction of an additional complication. Thus, if compounding is to be attempted, a high degree of supercharge will be required. This will produce an increase in the pressure level in the LP cylinder. Thus, frictional losses of the larger cylinder can be offset and additional power recovered.

For these reasons, most cycle-compounded engines have used some form of gas turbine in conjunction with the reciprocating unit for the low-pressure portion of the operation. With the use of a gas turbine, the frictional losses of the rotating element seldom exceed 2% of the power generated; thus energy is recoverable in greater bulk and under conditions that would be impossible with a reciprocating unit by itself.

This raises the question of "what is a turbocharged engine?" Is it a cycle-compounded unit or not? In a turbocharged engine, gases are adding work to the cycle which would otherwise have to be provided by the output shaft of the engine. This work does not appear as shaft horse power, but nevertheless extra energy is recovered from the working fluid. One can properly define a turbocharged engine as a cycle-compound engine, therefore, even though its objectives are somewhat limited.

Considerable work on cycle compounding was carried out about 1945-55, mainly in the aircraft engine field where the attractions of this work were greater due to the effects of altitude. Some work on ground-based equipment was also carried out, although it never reached production stages, probably because compound installations would have been quite expensive due to the need for high-speed gear trains and other complications.

It must also be remembered that the above period was one when major developments in compressor and turbine machinery were taking place, efficiencies were being improved, pressure ratios were being increased, high speeds of rotation were becoming possible, and high-temperature materials were becoming available at reduced costs. These improvements also benefited the turbocharger, which was capable of adding 50 to 100% in power with ease but without the improved fuel economy of other engines. The increase in the use of turbochargers brought costs down and made turbocharging the most economical method for power increase. The additional complication of the compound cycle—the gear train—was not justified by any possible additional economy. At the same time the engine developments necessary for increased ratings had to be effected to withstand the higher mean and maximum pressures, heat transfer, etc.

In the period under review, the pressure ratio from small compressors was also somewhat limited by the lack of design knowledge and by limited speeds of rotation.

The combination of factors discussed above limited the improved power and economy of cycle compounding to about a 8 to 10% improvement relative to turbocharging, an improvement which was offset by the increased costs for high-speed gears.

The situation today is that many turbocharged engines employ small high-speed chargers of 30 - 100,000 rpm giving good reliable performance when producing pressure ratios in the inlet manifold of 3.0:1 to 4.0:1 and higher. Examination of these engine cycles reveals that cylinder gas pressures of 300 psi or more exist at the point where the exhaust valve opens with gas temperatures of 2300°F approximately, ideal conditions for additional energy recovery. These engines operate best with exhaust manifold pressures of 2.5

to 3.5:1 ratio or higher. The reason for this drop in pressure is of course the need for scavenge cooling the cylinder and the fact that these engines charge at a high volumetric efficiency. Secondly the exhaust-driven turbo must not develop more power than the compressor can absorb. These necessary limitations have perhaps tended to restrict the outlook of the producers of these units, since high-efficiency units were not necessary, cheap construction could be employed, etc. Today, however, there is emerging an apparent need for exhaust-gas-operated turbines with high efficiency and a wide range of gas flow at high gas temperatures, in order to produce more powerful engines of smaller bulk and at the same time recover as much available energy as possible under a varying speed and load schedule. The present type of turbocharger is not well suited to a variable speed vehicular operation since almost all charger effects are lost at lower speeds. If the charger can maintain its pressure ratio over a wide speed range in one way or another, the equivalent of a larger engine is secured with considerable savings in engine space, weight, and fuel consumption. It has been shown that it is possible to produce about three to four times the torque now achieved at half engine speed from a given engine displacement by the effective use of the exhaust gas energy, given the required developments. Since this could eliminate the need for torque converters in the transmission with their poor efficiencies, reductions in both engine displacement and weight seem possible. It is hoped that designs capable of such achievements become available soon.

All of these developments raise the question, "What advantage now exists in producing a compound engine?" In the following pages estimates will be made of what can now be expected for the additional costs and complications involved.

2. COMPOUNDED ENGINES

The compound engine consisting of two or more different power plants can be built up in a variety of ways depending upon the purpose of such an engine. There are perhaps two reasons for such engines, viz.:

a. Two or more similar engines are connected together into one output shaft for a particular power plant because a single engine of the required characteristics does not exist. Examples of this were seen during World War II.

b. There may be a need for special performance characteristics to fulfill some particular purpose. The need may exist, for example, for the maximum possible economy over some wide range of speeds and loads for the major portion of the time, possibly as high as 80-90%. There may also be a need for extreme power for short periods without the accompanying weight and space handicap accompanying the use of a single compression ignition engine. Such applications in recent times have involved the use of a small, lightweight gas turbine geared into the transmission to assist the main engine when required.

The history of such applications is not new. One which comes to mind was the use on the German pocket battleships of the compression ignition engine for crusing, with the addition of steam turbines to the system for emergency power. Recently, designs of turbine-reciprocating engine combinations for tank operation have been made and tested. For such a use, a highly turbocharged engine of say 500 hp could be fitted with a small lightweight simple-cycle gas turbine of 500 hp for a total emergency power of 1000 hp. The reciprocating engine would be capable of supplying all the power for normal operation over pavement or off-road conditions, except when difficult terrain was involved or obstacle crossing was required. This combination of units would give extremely good fuel economy so long as the small compression ignition engine only was used, resulting from its operation at a higher load factor than that of the full 1000 hp engine. Specific fuel economy would suffer when the turbine was employed, but since the turbine would be employed less than 10% of the operating time, this increase could be tolerated. One must remember, however, that under combat conditions, a tank crew would probably not be concerned about conserving fuel by operating the oil engine alone but would wish to have all available manueverability and acceleration. Thus the turbine would probably be idled to keep it available for instant use. The result would be a large increase in fuel requirements since the simple gas turbine does not idle very economically.

There is one other problem involved in the employment of a torque converter transmission. Imagine that high resistance, such as a large obstacle, is encountered; the oil engine will go to full load and speed, and the turbine must be brought in to assist in the operation. Assume that the vehicle would be stalled in such extreme circumstances and the converter would have 100% slip, exerting some torque on the drive shaft. The turbine now picks up in power and applies additional torque to the input. Since additional torque to the output is only possible with additional relative slip, the reciprocating engine must increase in speed also. Thus the oil engine governor tries to control the speed by shutting off the fuel, transferring some of its load to the turbine. Assuming that this continued, it would be possible for the turbine to take all of the load and actually to motor the reciprocating engine at high speed without moving the vehicle. Even when the vehicle was not stalled, a transfer of load could occur to some extent. It appears that the application of the combined units to the typical transmission of today could occur only under a set of rather special circumstances where both engine torques and speeds provide a balanced system.

Apparently, compound engines for the purpose of heavy vehicular operation have some rather definite limitations, unless a rather complicated control and limit system is devised. Each installation would need to be uniquely designed for effective service.

The discussion of this type of compounding will be left at this point and compound cycles will be concentrated upon, since in this case the objectives are clear and a comparison between the various cycles, principles, etc. is possible.

2.1 COMPOUND CYCLE ACHIEVEMENTS - There are a number of major articles on this subject in the SAE Transactions. Those of immediate interest can be summarized as follows:

2.2 NAPIER NOMAD AIRCRAFT DIESEL (1)* - A two-cycle engine was provided with a twelve-stage axial compressor giving a pressure ratio of 6.28:1 with 8.25:1 at top speed at an efficiency of 85-87.5%. The compressor was driven by a three-stage turbine having an efficiency of 84-86% and an expansion ratio of 4.6:1 at sea level. These two units were geared to the crankshaft through an infinitely-variable-ratio gear in order to permit control of the degree of supercharge provided over the speed, altitude, and load range.

The air flow amounted to 13 lb/sec producing a net hp of 3135 for a specific fuel consumption of 0.345 lb/hp/hr including a slight thrust. An important feature was the small variation of SFC with load.

The above hp was made up of 2660 by the diesel, 2250 by the turbine with 1840 received by the compressor. Thus the net recovery from the rotating components amounted to about 410 hp or 15.4% of the diesel output.

2.3 THE ORION TANK ENGINE (2) - This experimental engine almost amounted to a total energy concept, since the heat lost in cylinder cooling was made available later in the cycle. The unit consisted of a two-cycle opposed piston diesel engine whose function was that of driving the compressors only. One compressor supplied air for engine cooling while a second supplied the combustion air. The combustion gases leaving the diesel engine were joined by the hot cooling air of the cylinders. The mixture, at a temperature of 500°F, was then passed through a turbine which produced all of the net output of the unit. The reciprocating engine was a gas generator for the power turbine.

---

*Numbers in parentheses designate References at end of paper.

The specific fuel consumption varied from 0.57 lb/hp/hr at full speed of 2350 rpm to 0.95 at 1500 rpm.

Turbine inlet pressures of 38 in. to 56 in. Hg abs. were employed with a gas flow through the turbine of 15.0 lb/sec at 2350 rpm, made up of 3.3 lb/sec of combustion air and 11.7 of cooling air. The net output of the unit was 600 bhp.

2.4 CURTIS-WRIGHT AIRCRAFT ENGINE - A third example, which will not be considered here, was the Curtis-Wright Aircraft engine, a spark ignition engine of the four-cycle type.

2.5 DISCUSSION OF PAST ACHIEVEMENTS - Of the above items, two are aircraft engines and one is a tank engine. The two former can be considered successful while the latter could only be considered of value where some consideration such as space, weight, turbine torque converter, etc. are more important than fuel consumption.

Aircraft applications came at a time when the jet engine was entering the field and thus decreasing the need for other power plants. The Nomad was, in fact, a combination of diesel engine and turbine, the latter unit being far more than a simple turbine unit and actually involving both multistage turbines and compressors. This complication, as well as the cost factor, was the main handicap to its useful application at the time.

The four-stroke cycle application to the gasoline aircraft engine has persisted and many examples are still in use today. Has this point any particular significance?

In the two, two-cycle engines, air flow for the units was 13 and 15 lb/sec, while the overall F/A ratios were 0.023 and 0.00622 respectively. Obviously, the turbine inlet temperature was quite low, particularly in the Orion engine. This could be a mixed blessing. Although this condition would allow the use of the cheaper materials, it would be very poor for the available energy content, particularly since low inlet temperature is accompanied by a relatively low inlet pressure, namely 56 in. Hg abs. This low temperature feature in both engines is associated with the fact that two-cycle engines can use a large excess air coefficient; again, conditions which are good for the engine but poor for the energy level in the turbine. The Nomad excess air coefficient was about 2.15 while the Orion was 10.5 overall with approximately 1.42 on combustion air only.

Thus, the turbine receives a large mass flow relative to the horsepower needs of the reciprocating engine, but receives it at a correspondingly low temperature so that the available energy per lb is reduced. The isentropic work of one pound of gas at 1000°R for an expansion ratio of 2:1 amounts to 43.3 Btu/lb. If the temperature is 2000°R, its value is 87.6 Btu for the same ratio. This represents more than a 2:1 increase. It follows that, for any given back pressure on the engine, the higher the temperature, the greater the horsepower per lb of flow. There are, of course, limits to the temperature that can be imposed upon the material employed in the construction of the engine.

This raises the question of the usefulness of the four-cycle engine for compounding purposes, since, in this particular case, the excess air is at a minimum, being only that amount necessary for satisfactory clean burning of the fuel. By the same method of calculation used for the two-cycle engine, the excess would amount to only 0.25 to 0.50.

It should be pointed out that, for the same air supplies, 46800 lb/hr of the Nomad and 54000 lb/hr of the Orion, a straight turbocharged four-cycle engine would develop about 4500 hp and 5500 hp in place of 3135 hp and 600 hp of the Nomad and Orion respectively. The Orion's relatively poor performance per lb of air is apparent.

In this connection, it should be remembered that the Orion uses high pressure air for cooling which is then mixed with the engine exhaust gas and fed to the turbine. It is true that the energy used to compress this air is more than recovered in the turbine; however, it is also true that it is still expensive air as far as the hp output is concerned. A greater return would be obtained from the power plant if normal air-cooling pressures were employed; gases would then reach the turbine at greatly increased temperatures.

2.6 FOUR-CYCLE COMPOUND ENGINE - The above remarks have indicated some of the problems associated with two-cycle engine compounding of the past. We will now examine the conditions under which a four-cycle compression ignition engine could be compounded today and determine the expected performance of such a unit. This problem will be approached in several ways: as a simple compound engine; as a compound unit in which only the turbine delivers power; as a compound unit in which both engine and turbine provide power, with the latter supplying additional power in emergency operations; as indirect compounding in which accessories, etc. are driven by recovered excess energy.

2.6.1 Simple Compound Engine - It is intended to keep the overall power plant as simple as possible, yet, at the same time, to take advantage of modern developments which, though not immediately available, should be practicable within the very near future. It will take two or three years to develop an engine capable of taking advantage of charger and turbine developments.

The simplest engine, at the present time is the turbocharged engine with its limited use of complete expansion of exhaust gases. Engines to be used for military purposes are already developing mean effective pressures of 300-350 psi with charger pressure ratios of 3.0 to 3.5:1 when equipped with after-coolers. Even these high values have been exceeded in some recent developments. The BSFC of such engines has not, in general, deteriorated to any marked extent, but, of course, high compression and firing pressures have been encountered. These latter can be controlled by some form of variable compression ratio device, such as that of Continental Aviation and Engineering Co. as reported in Reference 3. It follows that the significant power increases possible need not involve increases in engine weight or space to any extent.

From this standpoint, the simplest advance to complete cycle compounding is the gearing of the turbocharger to the engine output shaft in order to permit the maximum energy to be recovered in the turbine from the exhaust gases of the reciprocating unit. This is the same principle as that employed by the Nomad engine, except for the use of a four-cycle engine and the absence of the variable-speed charger drive, used in that system to permit maximum use of the power plant under widely varying altitude conditions.

With this combination of engine, turbine, and charger (shown in Fig. 1), there is still a rapid drop in manifold pressure as speed is reduced, common to all charged engines, which, of course, is responsible for a corresponding reduction in engine torque. However, military use of vehicles demands, if at all possible, high torque at low speed. In present installations, this is obtained by the use of a torque converter in the transmission. If torque requirements could be fulfilled without this device, an improved power output and fuel economy would result over an appreciable portion of the operating range.

For this simple compound cycle, the following estimate has been made for an engine fitted with chargers capable of producing a turbine expansion ratio of 4:1 and 5:1, which means a compressor ratio of about 4.6:1 or 5.8:1 to provide for adequate scavenging of the engine cylinder. Figure 2 gives a compressor map which almost fulfills the above requirement, at least as far as the 4.6:1 ratio is concerned. Further advancement must be made in some direction, perhaps in the direction of two-staging, in order to meet the other ratio. The machine of Fig. 2 was not designed for military use. It is assumed that given the necessary developmental work, some advances in efficiency are still possible. It is possible to have an engine full load and speed performance as shown below in Table I, when the unit is fitted with an aftercooler producing a manifold temperature of 200°F.

It can be seen that there is a possible 10% increase in power and torque with a corresponding reduction in fuel flow. However, is such a gain worth the additional complication? To complete the evaluation, the situation at low speed must also be examined. Assume that it is necessary to operate at about half engine speed. In this case, the compressor is also at half speed when the pressure ratio produced is about 1.7:1. Now, if the compressor is turbo-driven in the normal manner without the gear set connecting it to the crankshaft, a pressure ratio of about 2.0:1 would be expected at full throttle and half speed. It follows that the compound engine will have a reduced low-speed performance relative to the straight turbocharged engine with the same pressure ratio at full speed. This, of course, is not a desirable characteristic for an Army vehicle. Thus, it would be necessary to limit the speed range over which the engine operated by the use of the transmission and torque converter. This is not impossible and, in fact, the engine speed range of most tank engines is limited in this way. However, such means of producing necessary speed and load ranges are not conducive to fuel economy.

The problem remains: what means, if any, can be used to improve this low-speed condition produced by the power and torque situation, and, at the same time, obtain the improved economy of compounding. The charger is geared to the engine; hence its speed-pressure relationship is fixed. Thus, increase in pressure is impossible unless a variable-speed ratio is added, further complicating the system. Alternatively, the use of a variable-geometry nozzle in the turbine could increase the turbine inlet pressure and increase the power recovered in this unit. This recovery would be rather small, however, compared with the engine power reduction for a 1.7:1 to 2.0:1 change in ratio. Also a back pressure on the engine in excess of the charger pressure to achieve greater recovery would affect the engine output slightly.

It is concluded that a single fixed-drive compounded turbocharger would produce gains of about 10% in power and economy at full speed tapering away to zero at some point before half engine speed is reached. If a two-step gear ratio is employed, economies and power increases of the above magnitude could be gained over a fair proportion of the speed range. An infinitely variable gear ratio would be necessary for maximum gains at all operating conditions. The only other alternative is the limitation of engine speed over a rather narrow band by the use of gears and a torque converter. In this case, improved engine operating conditions would be secured at all times, relative to a turbocharged power plant operating over the same speed range.

2.6.2 Compound Engine with Turbine Developing the Power Output - This type of unit is shown diagrammatically in Fig. 3. It consists of an engine driving a compressor and acting as a gas generator for the turbine unit. This is the same scheme used in the Orion engine except that a four-cycle engine without heat recovery from engine cooling has been specified as the power plant. Calculating this system for the same net 1400 hp as in the previous case, the data given in Table II was obtained for turbines having expansion ratios of 4, 5 and 6 and the exhaust gas temperatures indicated.

This table is based upon a turbine output of 1400 hp for given pressure ratios and turbine inlet temperatures. It brings out one very important difference between two- and four-cycle engines, namely, the excess air flow of a two-cycle engine can be increased as necessary, by modification of port design, until the air flow requirement for the turbine is achieved. In this case of the four-cycle engine, the cylinder displacement acts as a meter on the air flow. The result is that, if an F/A ratio of sufficient magnitude is employed to produce the required gas temperature, engine displacement necessary for the turbine is roughly three times that needed to drive the air compressor.

The conditions discussed above will now be re-evaluated on the basis of balanced engine and compressor requirements and what can then be expected from the turbine under these conditions will be determined. These data are shown in Table III for the same engine displacements as shown in Table II in order to meter the same air flow.

Table III shows that a balanced design for the engine and compressor horsepowers reduces the F/A ratio to quite low values and results in exhaust temperatures of 600 to 700°F in place of 1500° to 1800°F. The effect is that the turbine output is approximately one-half that desired, while the SFC, based on turbine output, is improved due to the reduced fuel per lb of air required.

Thus, the original system assumed is an impossibility, both theoretically and practically. To act as the air meter, the size of diesel required would alone develop more than double the power of the turbine at half the SFC. The two-cycle engine, due to its air bypass capabilities, can be employed in the system analyzed but a four-cycle engine could not.

2.6.3 Compound Engine with Turbine as Power Generator with Bypass Air - It has been shown that the scheme discussed in Section 2.6.2 fails because of the incompatibility of the engine and turbine on an air flow basis. It is now proposed to add a bypass line from the compressor directly to the turbine in order to permit equality of displacement and horsepower of the reciprocating engine with the compressor and turbine requirements. This would result in some fraction of the air passing through the engine and picking up heat in the process. The balance of the air required for the turbine would not pass through the aftercooler, but would go directly to a combustion chamber where additional fuel could be added as necessary. It would then go to the turbine, mixing with the exhaust gases from the engine in the process. A schematic diagram is given in Fig. 4.

In this system, the engine can meet its own air and fuel requirements, while the additional flow requirements of the turbine can be met by the bypassed air. However, the fuel added to this turbine air will not be used as efficiently as it would be if added to the engine cycle.

This arrangement provides for a high degree of flexibility since the engine can now operate at any desired speed and supply the right amount of gas to meet turbine requirements. However, for high power, high engine speed is necessary to obtain high air pressure. Part load would best be effected by high air pressure also. This could be achieved to some extent by the use of variable geometry nozzles in the turbine. To secure both speed and full load, both sources of gas—engine exhaust and bypass air—must be used. Emergency loading for a combat vehicle would involve a still greater addition of fuel in the turbine combustion chamber. Reduced load could be achieved by reducing the engine fuel supply in a normal manner.

While conditions are such that the engine could handle all of the gas flow necessary, the bypass would be closed and the turbine nozzles set for the highest possible pressure. Should the load demand conditions under which the compressor would surge, an indicator would respond, opening the bypass until surge is suppressed. Thus, a much wider range of engine operating conditions could be possible than would otherwise be the case. However, under all conditions, the engine should be operated at the highest possible mean pressure at the slowest speed in order to maintain the unit at peak efficiency, providing as much gas for the turbine as possible. Variable geometry is definitely indicated.

Using a compressor pressure ratio of 4.5:1 with an efficiency of 78%, the power requirements are 126.0 hp/lb of air delivered at a temperature of 448°F when the inlet air is at 80°F.

The compression ignition engine, with a manifold pressure of 130 in. Hg abs at 200-F and F/A = 0.045 and with an exhaust gas temperature of 1450°F, will develop a BMEP of 400 psi including cylinder cooling.

The turbine is assumed to have a pressure ratio of 4.0:1 with an inlet temperature at 1450°F and an efficiency of 86%, in which case it will develop 188.0 hp/lb of gas/sec.

It is now possible to determine the various combinations of air flows, etc., using the following equations:

Total Air Flow = Engine air + Bypass air

$$W_a = W_e + W_b \text{ lb/sec}$$

$$\text{Total Gas Flow} = 1.045\, W_e + (1+x) W_b$$

x = F/A Ratio of Turbine Combustion Chamber

HP of Compressor = bhp of Engine

$$= \frac{\text{mep x Disp. x rpm}}{792000}$$

Engine Displacement $= \frac{\pi d^2}{4} \times \ell \times n$

d = dia. of cylinder in.

ℓ = length of stroke in.

n = number of cylinders

η_v = volumetric efficiency

$$\text{Vol. Flow} = \frac{W_e RT}{\eta_v P} = \frac{\pi d^2 \ell n}{4 \times 1728} \times \frac{rpm}{2 \times 60} \text{ cu ft/sec}$$

The value of x, the fuel per lb of air in combustor, is given by

$$h_{T_a} + x(0.5T-287+\eta_c h_c) = (1+x)h_{T_g}$$

h_{T_a} = enthalpy of air at compressor exit

h_{T_g} = enthalpy of gas at desired turbine inlet temperature

h_c = calorific value of fuels

η_c = combustion efficiency

Applying these relationships, the following performance estimates were made (see Table IV).

When 1400 hp is required for an emergency condition an SFC of 0.446 is obtained, by cutting out the fuel to the turbine combustion chamber the power is reduced to 860 hp at 0.419 lb/THP/hr. If the same size of diesel engine was operated as a straight turbocharged unit, 928 hp would be delivered for 0.389 lb/bhp/hr.

These results indicate that a choice must be made on other than an economy basis. Undoubtedly, the straight turbocharged compression ignition engine, if made to fulfill the total requirements of 1400 hp, would be larger and heavier than the compound unit producing 1400 hp. The latter would also have the advantage of a built-in torque converter, if a free turbine unit was employed, in addition to the fact that the turbine would always be in operation heated up and ready for emergency power when required. The question to be answered thus becomes "Is the complication of two different power units in one vehicle, plus the maintainance problem, plus the increased fuel storage capacity necessary, worth any gains made in space and weight?" This question can only be answered by those intimately connected with the overall design and operation of the vehicle, plus a complete analysis of the typical schedule of day-to-day operation.

2.6.4 Compound Unit in Which the Turbine Can Operate Separately for Emergency Starting, Auxiliary Purposes, etc. - This unit will be similar to the simple compound engine discussed in Section 2.6.1. In this application, however, the compressor-turbine combination is fitted with a combustion chamber and a clutch mechanism so that it can operate as a separate unit when necessary. This arrangement is diagrammed in Fig. 5, where the two high-speed units are coupled together through a clutch to the engine, and where the bypass circuit includes a combustion chamber. Normal operation will be with the bypass closed and the clutch engaged. In this condition we have the duplicate of the unit discussed in Section 2.6.1, a simple compound cycle, with the additional advantage of the bypass available for the control of surge characteristics (as in the unit discussed in Section 2.6.3) at other than design conditions. At the same time, a simple gas turbine unit is available by supplying fuel to the combustion chamber which can then be used as starter for the compression ignition engine or other auxillary duties when the main engine is shut down. Secondly, where compressor capacity permits, air can be bypassed and fuel added to make available an emergency power condition for short periods. In addition, the excess air in the engine exhaust could also be consumed to some degree for high power output. The conditions differ from those described in Section 2.6.3, inasmuch as the turbine does not develop all of the available power.

A calculation of this cycle yield the following data when a 4.0:1 expansion ratio for the turbine is assumed. The complete unit, without the use of the bypass, will duplicate Table I. When the turbine-compressor unit is employed as a starter, the turbine hp will be reduced slightly for the same gas temperature in proportion to the smaller fuel flow and will be 175 hp in place of 180.0/lb of air/sec.

Compressor hp = 136.3 hp/lb/sec
Turbine hp = 175.0 hp/lb/sec
Net Output = 38.7 hp/lb/sec
Air Flow = 3.0 lb/sec
Total Output of Turbine Unit = 116.1 hp

This power would be ample for starting and warming of the engine under cold conditions. Since the compressor can be at full speed at the start, the air to the engine cylinder will be at about 200°F when the atmospheric temperature is -65°F.

In the case where some proportion of the excess air in the engine exhaust is consumed to achieve maximum power, the performance becomes as shown below if fuel is burnt to produce a turbine inlet temperature of 1800°F: for a 22.0% increase in fuel flow, there is a 12% increase in output (see Table V).

The results shown in the table are achieved by the addition of a bypass, combustion chamber, and controls. In exchange there is a powerful starter, heater, and a 12% increase in power output. These items would need to be balanced against each other to reach a final evaluation for any power plant application.

In order to take advantage of the additional possibility (if compressor capacity permitted) of bypassing some air to the combustor directly from the compressor and adding additional fuel for a further power increase, variable geometry would be required in the turbine. Variable geometry is thought to be a requirement in any case for any major advance in turbocharging capabilities, and is required for the simple compound engine at low speeds.

There is still one other important feature associated with the arrangement under consideration and that is the use of the turbo combination to apply high boost pressure to the engine at low speed. To do this, the clutch is disengaged from the engine allowing the turbine to drive the compressor at any speed desired, depending upon the available energy in the exhaust gases. In the case examined, the oil engine is at full throttle but at half engine speed; it is assumed that the aftercooler will still maintain a manifold temperature of 200°F. Under these conditions, the engine will draw 1.5 lb of air/sec while the compressor, driven at its full speed, would again deliver 3.0 lb/sec against a 4.6:1 ratio, the additional 1.5 lb of air must then be bypassed to the combustion chamber. Since the manifold pressure is constant, the IMEP will be almost constant thus the ihp is 0.5 that at full speed; however, since frictional losses will be reduced, a bhp of 655 should be available in place of the approximately 320 produced by a normally turbocharged engine. The output torque will be at least 90% of that at full load and speed in place of 48% for the conventional turbocharger, a two-to-one torque increase at slow speed, a very important factor in vehicular propulsion. Of course, to effect this increase in torque, additional fuel has to be added to the combustion chamber to produce the turbine power required. The values are as shown in Table VI.

An important point in this approach is the fact that the compressor air flow is the same for the two speeds, while the turbine flows are 3.13 lb/sec when compounded and 3.16 lb when boosting at low speed. Since the gas temperature under the boost condition will be lower than the compound state, a variable-geometry nozzle is still required. It is apparent that conditions do not vary so wildly that great changes in nozzle areas etc. are required.

This scheme also presents the possibility of an actual increase of engine torque above that at full load conditions by the addition of an extra fuel supply to the combustor which would in turn overspeed the turbine and increase manifold pressures. How far this could be carried out depends upon the design features.

2.6.5. Indirect Compounding - This application is that of recovering energy from the cycle otherwise wasted in one way or another in such a manner that necessary functions are performed by means of this recovered energy. Examples of this are the following:

- a. Driving cooling fans for
    - 1. Cylinder cooling
    - 2. Lubricating oil cooling
    - 3. Transmission cooling
- b. Electrical generation
- c. Air conditioning

It must be remembered that some of the items such as (b) and (c) must be at full power while the main engine is idling or even shut down. Engine cooling etc., however, is not required unless the power plant is in operation. It is proposed, therefore, to examine how the available excess energy could be used effectively. It is not proposed to discuss the details of the practical application of the various ways of solving the problems involved but merely to establish if a particular function can, or cannot, be performed by the excess energy available.

2.6.5.1 Cylinder cooling - In the case of either water- or air-cooled engines, the cooling fans must provide sufficient circulation of air for all the cooling requirements. However in the case of both water- and air-cooled engines, it would be relatively easy to separate the engine and oil cooling functions from the rest of the cooling system and provide some form of turbine drive to operate fans, which could be geared to the turbocharger if the turbine was designed to recover all available energy.

The capability of fulfilling this requirement will be investigated. It will be assumed that a 4.0:1 expansion ratio across the turbine is available, and a somewhat lower exhaust temperature 1250°F will be assumed, to be on the safe side. The turbo efficiency is assumed to be 87% while 85% will be employed when a separate fan-drive turbine is used with a pressure drop of 3% between the two turbines. If 80°F inlet air is used, then the requirements for the turbocharger become as shown in Table VII.

It is seen that there is ample power for cylinder cooling when the engine is operating at full speed.

If we repeat the calculations for half engine speed, and use variable nozzle geometry to keep the turbine pressure ratio up, we can establish that there is 106 hp available, whereas the fan would need only about 70 hp at this speed. At idling speed, however, when the fan would need only 15 hp for operation, the turbine expansion ratio would be so low that it would be impossible to generate even this small hp. The scheme fails, therefore, due to the requirements for cooling at the idle condition.

2.6.5.2 Transmission cooling - The transmission cooling requirements, under engine idling conditions, are negligible. Ample cooling would be available, therefore, if the cooling can be accomplished over the range from one-half to full speed.

Transmission cooling is at present provided by the cooling fans at the engine. Based upon present data, it would be necessary to dispose of about 1100000 Btu/hr in the case of a 1400 bhp output. This would occur at slow vehicle speed with full throttle. The present fan power required for transmission cooling is estimated at 40 hp approx. It is seen that this amount could be provided via the excess gas energy without difficulty, effecting a 40 hp saving in power output, a rather minor quantity.

The alternative scheme outlined in Fig. 6 was investigated. In this case, all of the available energy of the exhaust is recovered in the gas turbine which drives a compressor capable of absorbing all of this power by pumping additional air. The excess air is bypassed off to an expansion turbine unit after passing through the aftercooler; it thus leaves this unit at low temperature and flows to an expansion nozzle forming an injector where atmospheric air is drawn in and passed through the cooler. This scheme permits placing the cooler in almost any convenient place since no mechanical drive for a fan is required.

The following data, in Table VIII, represent this process.

The air flow must be such as to balance out both the flow requirements and the horsepower generated. It is seen that the air expander adds considerable work to that of the turbine and results in an increased air flow. For each lb of air flowing through the turbine, 0.722 lb of air is available through the expander, giving a total air flow of 1.722 lb/sec in the compressor per lb of air in the turbine. The injector action can now be investigated (see Table IX).

A ratio of 3:1 has been assumed between driven and driving air in the injector with an outlet temperature of 200°F.

This method of cooling seems adequate and has some advantages, mainly in the placement of the units since the air for the injector can be piped readily to almost any place. The complication to the power plant itself is minor since no gear train is involved, and at the same time the bypass circuit can be used as a surge suppressor if need be. Some compartment cooling could also be effected in this manner.

3. DISCUSSION

The results presented show some of the main characteristics of current four-cycle engines together with some methods of recovering available energy from these engines. Examination of the cycles shows that the percentage gains possible are not greatly different from those of some 10-15 years ago, despite the increased level of turbocharger pressure. This arises from the fact that the cycle mep has also increased with manifold pressures. The vehicle power plant with its earth-bound environment does not show to so great an advantage as in the case of aircraft engines, where increasing altitude produces increasing turbine expansion ratio and greater availability of the energy present.

This examination has brought out the need for the use of a turbine with variable-geometry nozzles if any real progress in turbocharging or compounding is to be made. Engine full-power condition has already been improved in recent engines, but while idling and accelerating these engines are plagued with so much smoke that they are not satisfactory for military engines. The easy solution is to change the supercharge characteristic. One might keep the boost constant by means of a positive displacement charger, for example, or use combinations of positive and turbo chargers, etc. Neither of these possibilities seem entirely satisfactory since they would reduce the maximum power of a given engine and its fuel economy considerably. The alternative, the use of variable geometry, seems to be a good solution to the problem. Calculations of energy requirements have shown that over a speed range of 2:1, an increasing boost pressure is a possibility as speed reduces. This increase would help the smoke condition and at the same time would give an increasing torque as speed falls, a very desirable characteristic for a tank engine. It has been shown (see Fig. 7) that a torque increase of almost four times that of current engines is possible with variable geometry. This method of approach would eliminate the need for a torque converter in the transmission, with its accompanying inefficiencies, and at the same time would reduce the vehicle's maximum hp needs, its transmission cooling requirements, etc., and would therefore allow a smaller and lighter power plant. Of course such a major change in the method of engine operation would require some parallel work on the engine as well as on the supercharger.

The method of manifold pressure increase by the use of variable geometry can also be employed at constant speed to produce increasing torque with load. In this case, a smaller power plant than those now used for a given maximum horsepower is possible with gains in fuel economy due to higher load factors.

A careful survey of what might be done to turbochargers in the way of variable geometry, air flow ranges, and accompanying pressure ratios should be carried out by experts in this field with an open mind.

It must be admitted that the data given in this paper are preliminary only, covering a limited range, mainly at full load. To provide data for all conditions, a complete examination over the whole performance requirements should be made. In order to do a detailed analysis it would be necessary to have available fairly accurate compressor and turbine performance predictions. At the same time any increase in unit efficiencies achieved would pay off in increased pressure ratios. It is believed that the engine can be made to handle a greatly increased manifold pressure. Though the firing and compression pressures will be increased, the variable compression ratio engine that has just been developed may prove capable of handling this part of the problem successfully.

The cost of the type of charging unit required will undoubtedly increase, but if a smaller and lighter engine with improved economy results, the increased cost may well be a bargain for such purposes as are being considered here.

4. CONCLUSIONS

The employment of a compound compression ignition engine could increase maximum power and economy by about 10% when applied to a four-cycle type of engine.

The type of compound engine in which the low-pressure turbine unit develops all of the power output is not a possibility when employing a four-cycle engine and present-day pressure ratios unless the major air requirement for the turbine bypasses the reciprocating unit. In this case, the fuel economy of the unit suffers.

The compound unit in which the compressor and turbine can operate as a starter and emergency power unit is an attractive possibility particularly for cold starting. It can achieve an increase of 12% in maximum power for a 22% increase in fuel supply. This type of unit can also be operated in a manner which will permit obtaining about 90% full load and speed torque at 50% engine speed for a minor increase in fuel when such torque is required.

The excess available energy of the exhaust gases of a typical turbocharged high duty engine can also be employed for such secondary purposes as cooling transmission, etc.

A need has been established for a variable-geometry turbine, even for a simple turbocharged unit. With such equipment, the torque characteristic of a compression ignition engine can be adjusted in such a manner that the transmission system could be simplified considerably.

Further major progress in oil engine utilization for heavy military equipment depends upon a satisfactory variable-geometry turbine being available.

5. ACKNOWLEDGMENTS

The basic work for this presentation was carried out for Continental Aviation and Engineering Corp.; in addition, some investigations were made for the U. S. Army Tank Automotive Center. The author wishes to thank those institutions for permission to present the above data.

6. REFERENCES

1. H. Sammons and E. Chatterton, "Napier Nomad Aircraft Diesel Engine," SAE Trans., Vol. 63, 1955, pp. 107-125.

2. R. G. Hooker, '"Orion': A Gas-Generator Turbocompound Engine," SAE Trans., Vol. 65, 1957, pp. 293-330.

3. W. A. Wallace and F. B. Lux, "A Variable Compression Ratio Engine Development," SAE Trans., Vol. 72, 1964, pp. 680-689.

TABLE I

Performance of Simple Compound Engine

Turbine Expansion Ratio	4.0	5.0
Compressor		
Compressor ratio	4.6	5.8
Air inlet temperature, °F	100	100
Compressor efficiency	0.76	0.76
hp/lb of air/sec	136.3	153.6
Diesel Engine		
Full load F/A ratio	0.043	0.043
Manifold pressure (including losses aftercooler), psi	66.0	84.6
BMEP, psi (including cylinder cooling fans)	410.0	530.0
Manifold temperature, °F	200	200
Volume of air, cu ft/lb	3.71	2.91
Engine displacement, cu in/lb of air/sec at 2800 rpm	292	229
Diesel Performance Turbocharged		
bhp/lb of air/sec	422	429
Fuel, lb/hr/lb of air/sec	154.8	154.8
BSFC, lb/bhp/hr	0.367	0.361
Exhaust gas temperature, °F (predicted)	1300°	1400°
Turbine		
Expansion ratio	4.0:1	5.0:1
Inlet gas temperature, °R	1760	1860
Turbine efficiency	0.86	0.86
Turbine output, hp/lb of air/sec	180.0	201.5
Overall Performance		
Net output (including engine cooling fans), hp/lb of air/sec	465.7	476.9
Fuel flow, lb/hr/lb of air/sec	154.8	154.8
BSFC, lb/net hp/hr	0.332	0.325
Air flow for 1400 hp net	3.00	2.94
Engine displacement for 1400 hp, cu in.	878	674

TABLE II

Performance of Compound Engine, Turbine Developing All Output

Turbine expansion ratio	4	5	6
Inlet gas temperature, °F	1500	1650	1800
Turbine efficiency, %	86.0	86.0	86.0
Gas flow required, lb/sec for 1400 hp	7.3	5.96	5.10
Engine displacement, cu in. for 1400 bhp at 2800 rpm	2300	1500	1070
Air flow from compressor, lb/sec for F/A = 0.04	7.0	5.73	4.90
Fuel flow, lb/hr	1009.0	826.0	706.0
SFC (net output), lb/thp/hr	0.72	0.59	0.504
Compressor, hp	942	914	887
Engine hp at 2800 rpm and 0.04 F/A	2830	2830	2080
SFC, lb/bhp/hr diesel only	0.357	0.347	0.340

TABLE III

Performance with a Balanced Engine and Compressor

Turbine expansion ratio	4	5	6
Compressor, hp	942	914	887
Engine displacement, cu in. to give required air flow	2300	1500	1070
BMEP required for compressor, hp	118.0	176.0	240.0
F/A ratio necessary for BMEP	0.0175	0.02	0.0215
Exhaust gas temperature, °F expected	575	630	680
Turbine, hp	740	725	706
SFC, lb/thp/hr	0.595	0.569	0.537
Engine displacement, cu in. for compressor hp at F/A = 0.04	680	530	430

TABLE IV

Performance of Compound Engine with an Engine Bypass Circuit

	Output		
	Emergency	Normal	Diesel Engine Turbocharged
Turbine, hp	1400	860	928
Gas flow, lb/sec	7.44	7.37	---
Mixed gas temperature at inlet, °R	1910	1221	1910
Engine air flow, lb/sec	2.23	2.23	2.23
Comp. air flow, lb/sec	5.04	5.04	---
Engine fuel flow, lb/sec	0.1004	0.1003	0.1003
Comb. chamber fuel flow, lb/sec	0.073	---	---
Total fuel flow, lb/hr	624	361	361
Overall sfc, lb/thp/hr	0.446	0.419	0.389

TABLE V

Emergency Output with Turbine Unit Operating as Power Plant

Engine gas temperature = 1300°F
Turbine inlet temperature = 1800°F
Air flow = 3 lb/sec
Fuel to combustion chamber = 101 lb/hr
Fuel to engine = 464 lb/hr
Total fuel flow = 565 lb/hr
net hp produced = 1574
SFC = 0.36 lb/bhp/hr

TABLE VI

Performance at Slow Speed with Declutched Charger

Compressor pressure ratio = 4.6:1
Total air flow = 3.0 lb/sec
Air flow to engine = 1.5 lb/sec
Air flow to combustion chamber = 1.5 lb/sec
Air inlet temperature to engine = 200°F
Air inlet to combustion chamber = 448°F
Engine F/A = 0.043
Gas flow from engine = 1.65 lb/sec
Gas temperature from engine = 1300°F
Fuel flow to combustion chamber = 0.00772 lb/sec
Fuel flow to engine = 232 lb/hr
Fuel flow to chamber = 27.8 lb/hr
Total fuel flow = 259.8
Output, hp = 655 bhp
SFC = 0.396 lb/bhp/hr
Torque = 0.9 x full power torque
Engine, rpm = 1400
Gas inlet temperature to turbine after mixing and combustion = 915°F approx

TABLE VII

Engine Cooling with Available Exhaust Gas Energy

Turbocharger

Compression pressure ratio = 4.5:1
Inlet temperature, °F = 80
Compressor efficiency = 0.78
Delivery temperature, °F = 448
Work/lb air/sec, °F = 126.0
Turbine inlet temperature, °F = 1250
Gas flow/lb or air, lb = 1.043
Output of turbine, hp = 126.0/lb of air
Expansion ratio required = 2.61:1

Fan Turbine

Turbocharger outlet temperature, °F = 938
Fan turbine pressure ratio = 1.46
Fan turbine efficiency = 0.85
hp available = 176
Cooling fan requirements for 1400 hp = 135

TABLE VIII

Indirect Cooling of Transmission

Compressor

Inlet temperature = 100°F
Pressure ratio = 4.6:1
Compressor efficiency = 0.78
Work of compression = 133 hp/lb/sec

Expander

Inlet temperature = 200°F
Inlet pressure = 66.3 psia
Outlet pressure = 22.0 psia
Work of expander = 48.5 hp/lb air/sec
Outlet temperature = 57°F

Reciprocating Engine

Net output (including cooling) = 1400 hp
BMEP = 390 psi
rpm = 2800
Displacement = 1020 cu in.
Air flow (vol. effy. = 0.94) = 3.47 lb/sec

Turbine Engine

Exhaust temperature = 1550°F
Inlet pressure = 58.8 psi
Exhaust pressure = 14.7 psi
Turbine efficiency = 0.85
Turbine work = 194.0 hp/lb of air/sec

TABLE IX

Indirect Cooling of Transmission

Inlet temperature = 57°F
Inlet pressure = 22.0 psi
Induced air flow = 3 lb/sec/lb of air
Induced air temperature = 100°F
Temperature of mixture = 77°F
Total air flow in cooler = 2.888 lb/sec
Air flow in engine = 3.47 lb/sec
Heat absorption with 200°F outlet temp = 1,070,000 Btu/hr

FIGURE CAPTIONS

Fig. 1. Simple compound engine.

Fig. 2. Compressor map.

Fig. 3. Compound engine turbine develops output.

Fig. 4. Compound engine turbine as power generator with engine bypass.

Fig. 5. Compound engine with turbine starter.

Fig. 6. Indirect cooling of transmission.

Fig. 7. Relative torque curves with variable geometry.

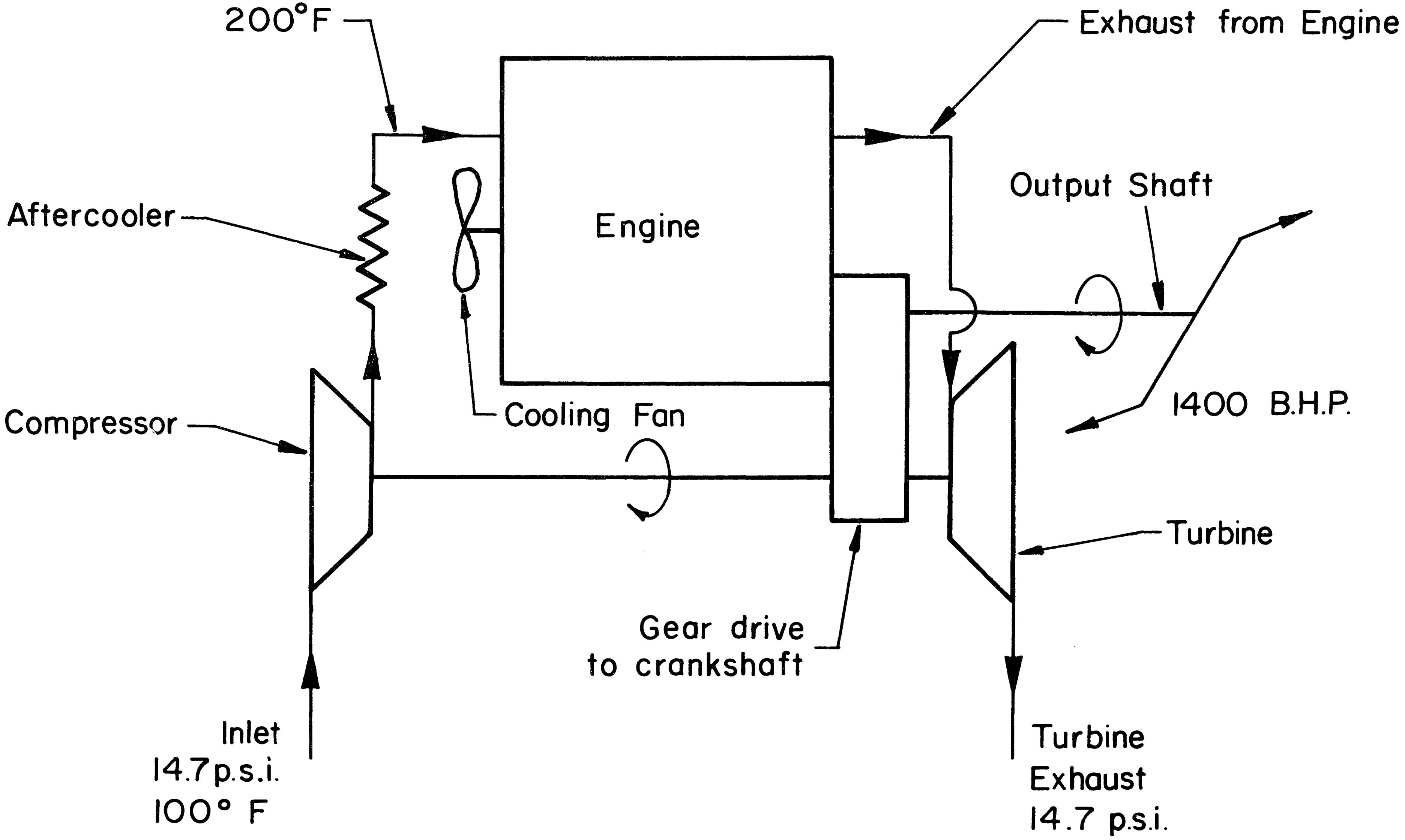
200°F
Exhaust from Engine
Aftercooler
Engine
Output Shaft
1400 B.H.P.
Compressor
Cooling Fan
Turbine
Gear drive
to crankshaft
Inlet
14.7p.s.i.
100° F
Turbine
Exhaust
14.7 p.s.i.

Fig. 1

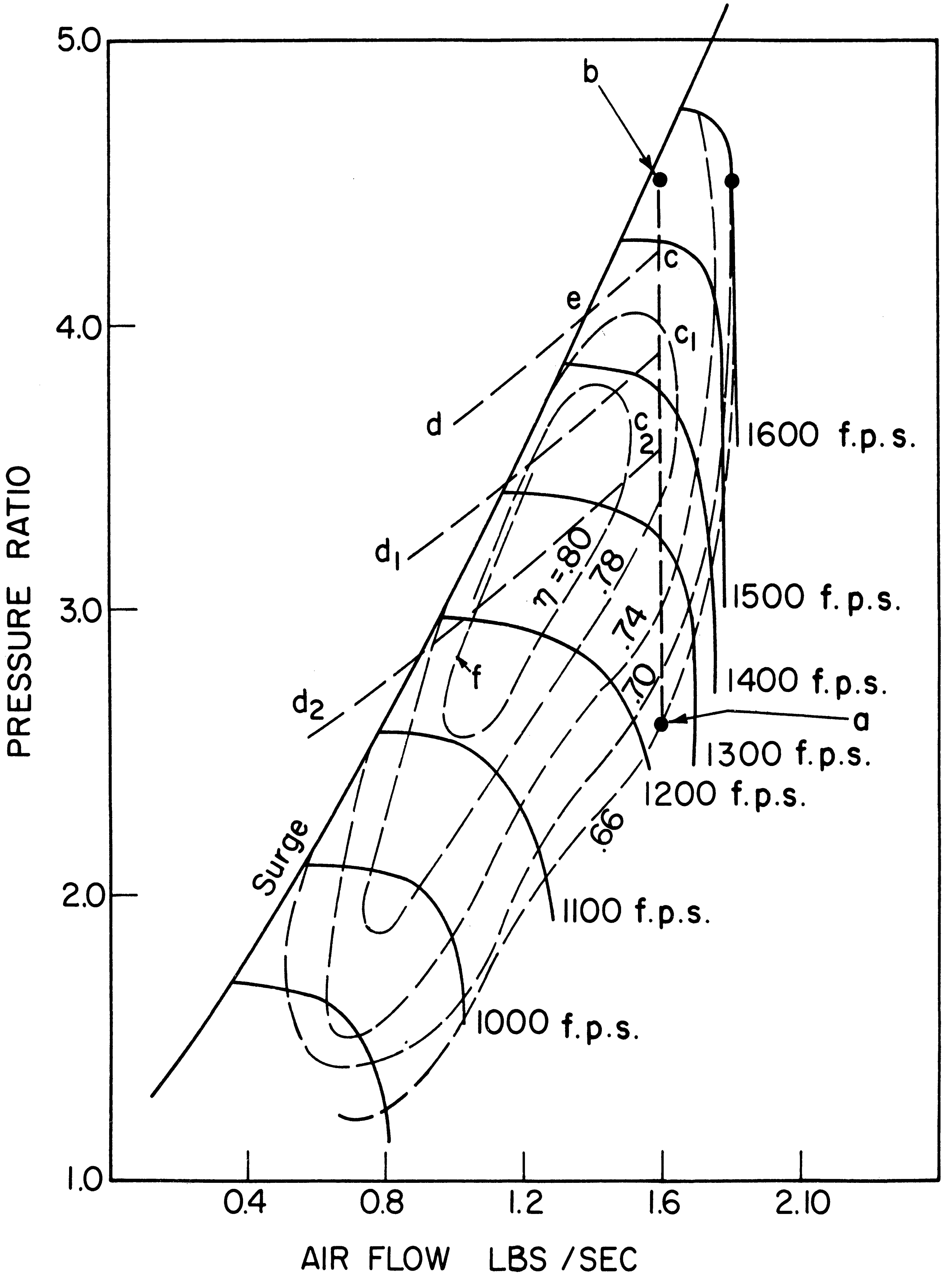
5.0
4.0
3.0
2.0
1.0
PRESSURE RATIO
0.4
0.8
1.2
1.6
2.10
AIR FLOW LBS /SEC
b
c
e
c1
d
c2
d1
η =.80
.78
.74
.70
.66
f
d2
a
Surge
1600 f.p.s.
1500 f.p.s.
1400 f.p.s.
1300 f.p.s.
1200 f.p.s.
1100 f.p.s.
1000 f.p.s.

Fig. 2

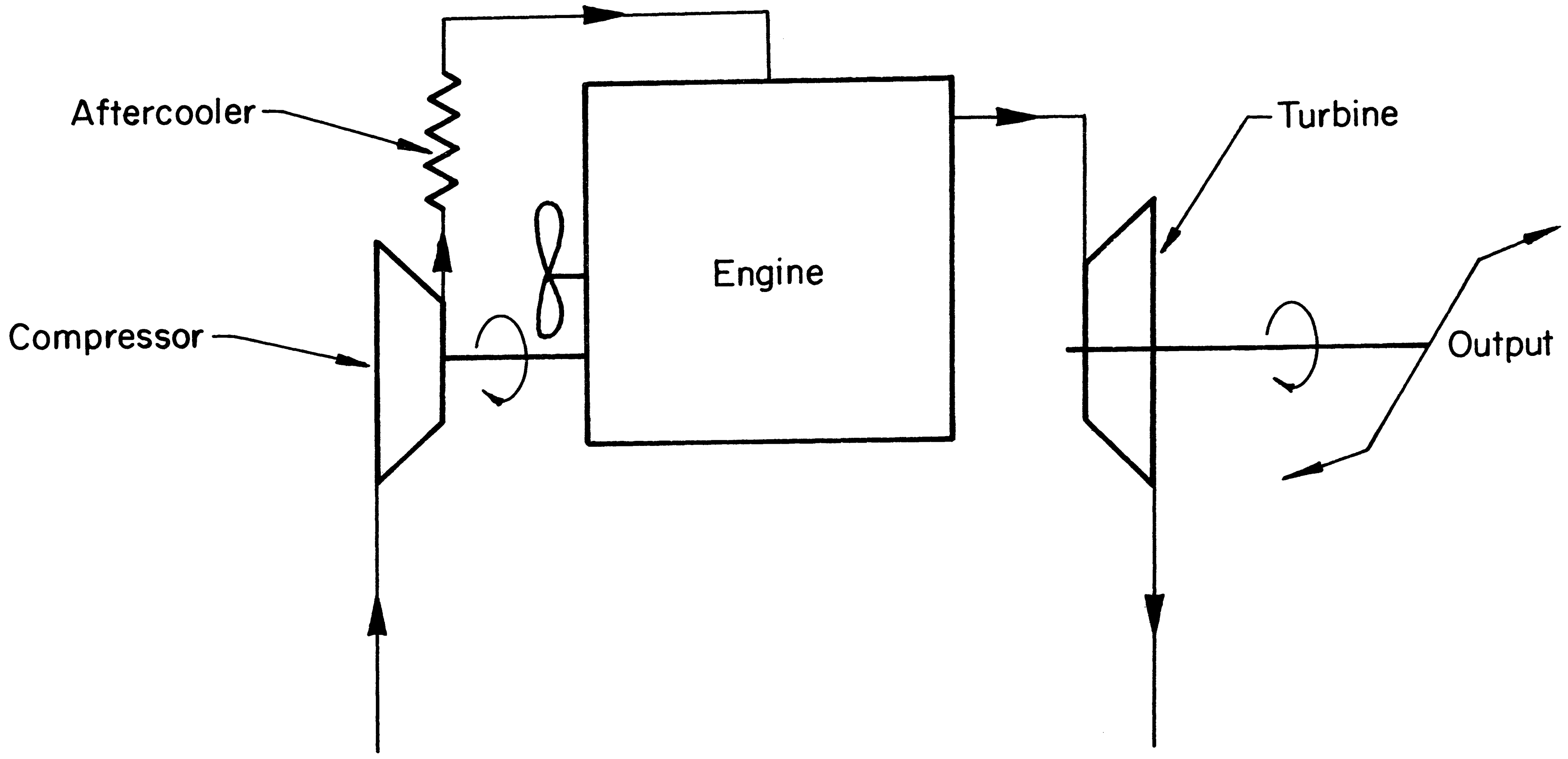

Fig. 3.

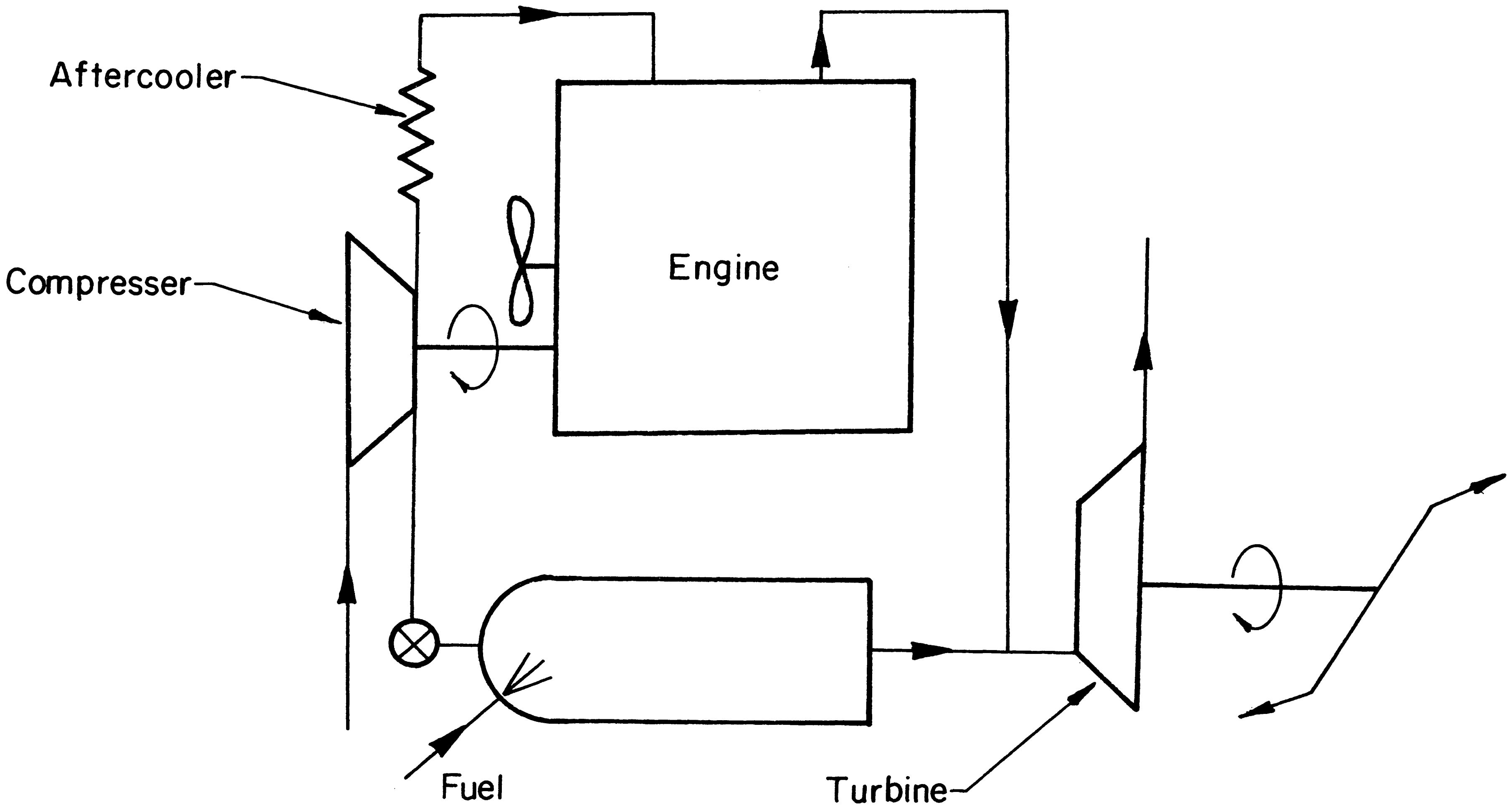

Fig. 4

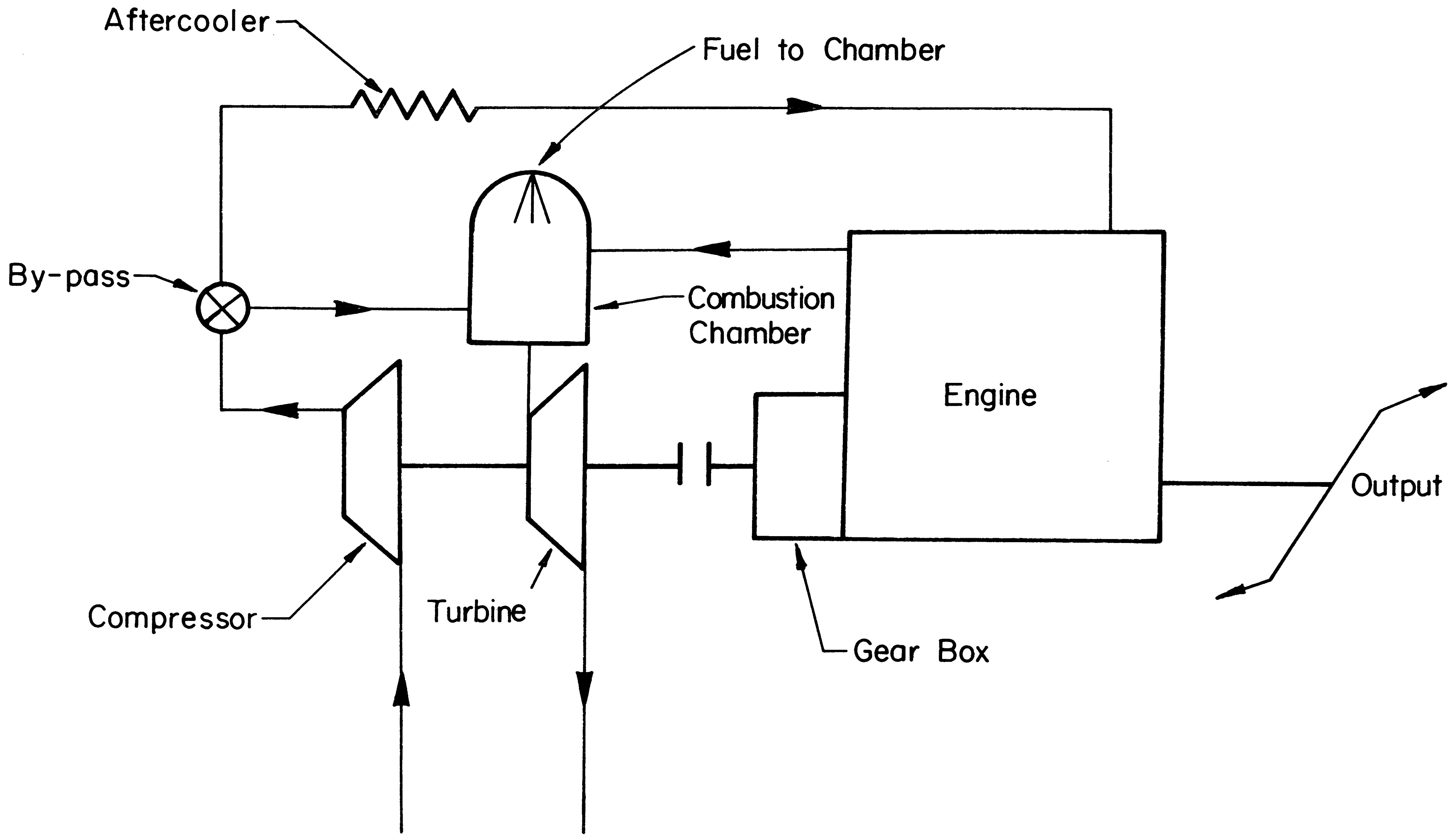
Aftercooler
Fuel to Chamber
By-pass
Combustion Chamber
Engine
Output
Compressor
Turbine
Gear Box

Fig. 5

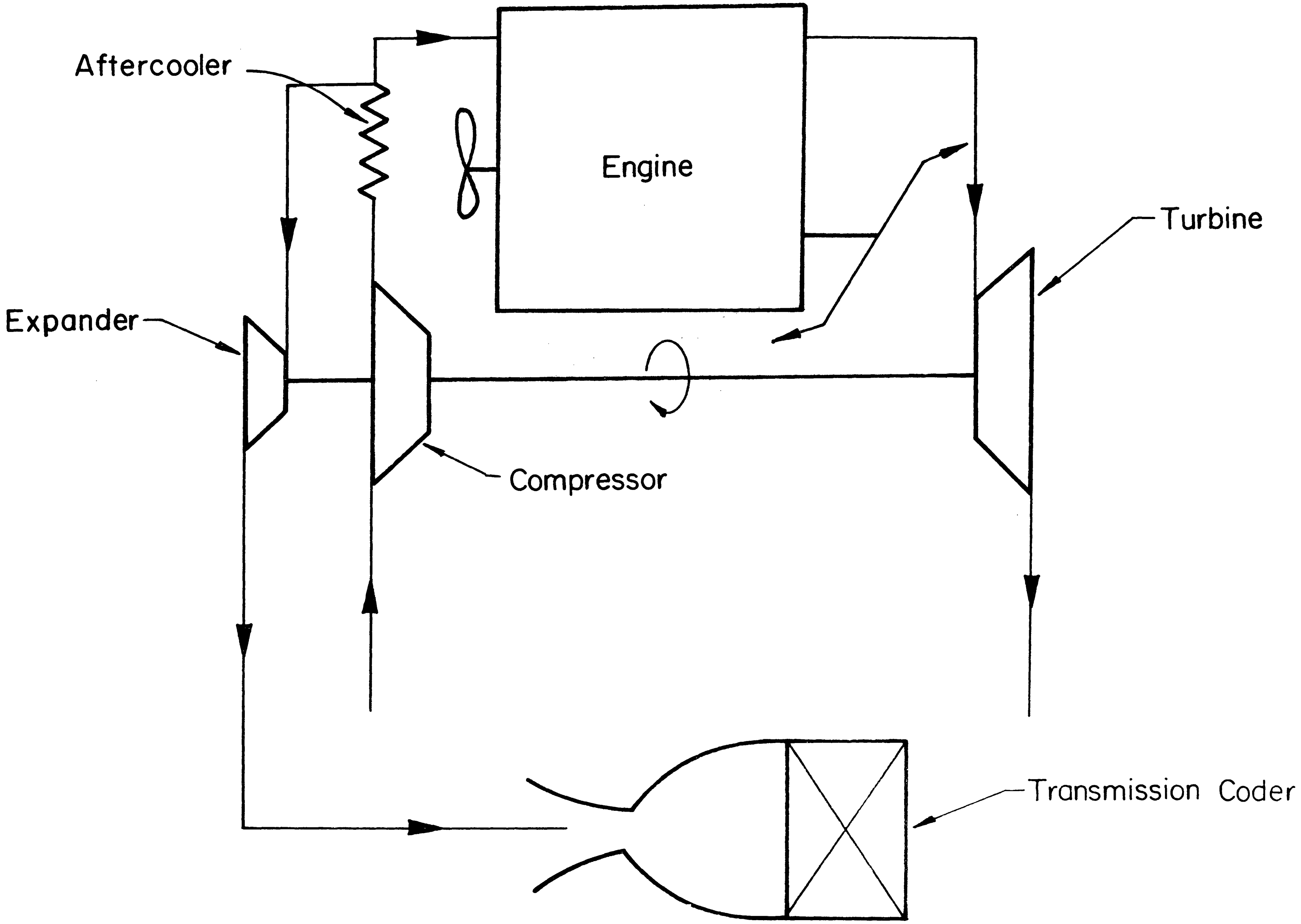
Aftercooler
Engine
Turbine
Expander
Compressor
Transmission Coder

Fig. 6

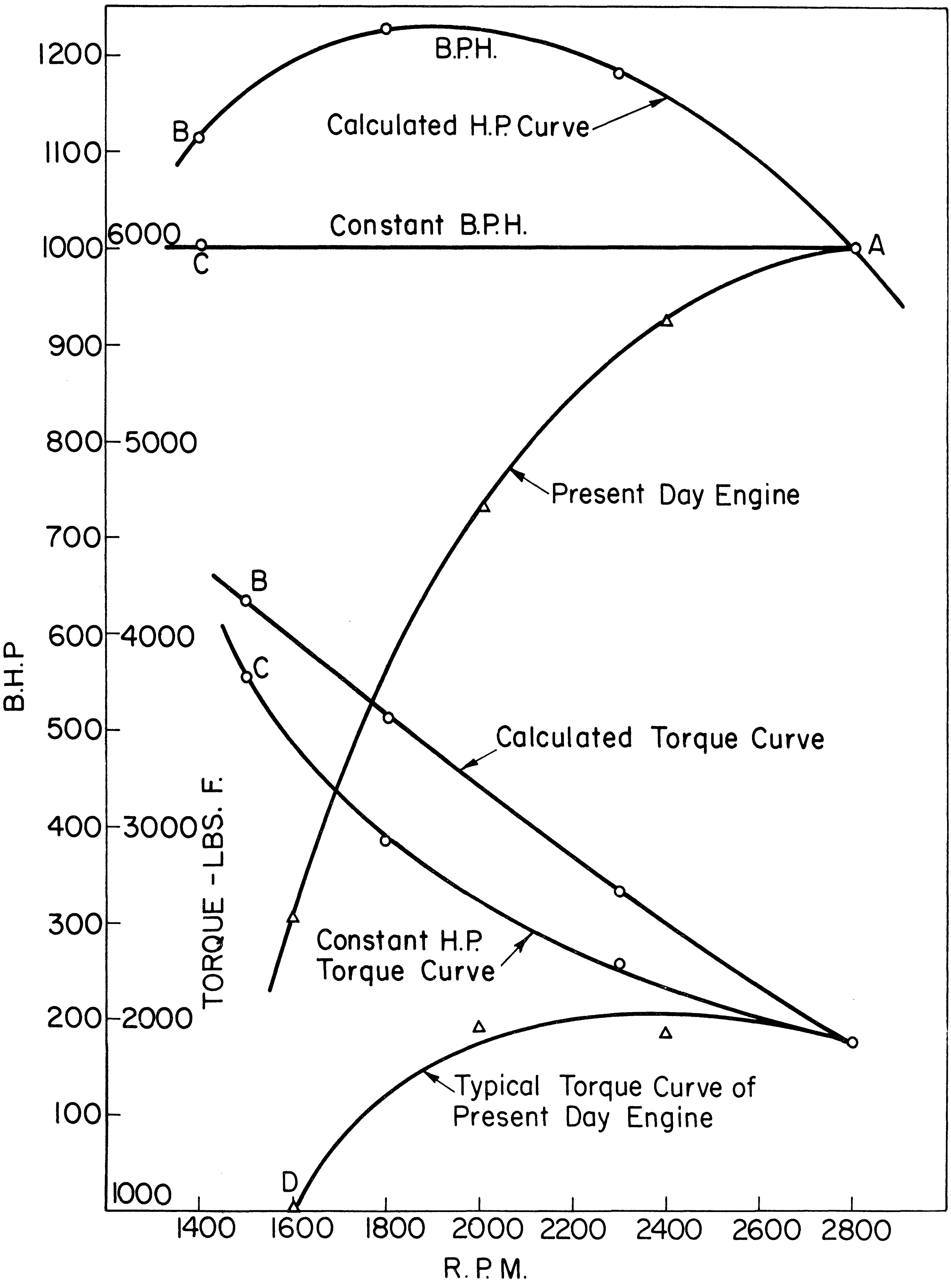
B.P.H.
Calculated H.P. Curve
B
Constant B.P.H.
C
A
Present Day Engine
B
C
Calculated Torque Curve
Constant H.P. Torque Curve
Typical Torque Curve of Present Day Engine
D
B.H.P.
TORQUE - LBS. F.
1200
1100
1000
900
800
700
600
500
400
300
200
100
6000
5000
4000
3000
2000
1000
1400
1600
1800
2000
2200
2400
2600
2800
R. P. M.

Fig. 7