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By K. Zinner

While the gas turbine by itself has been applied in particular cases for power generation and is in a state of promising development in this field, it has already met with considerable success in two cases when used as an exhaust turbine in connection with a centrifugal compressor, namely, in the supercharging of combustion engines and in the Velox process[1], which is of particular application for furnaces. In the present paper the most important possibilities of combining a combustion engine with a gas turbine are considered. These "combination engines" are compared with the simple gas turbine on whose state of development a brief review will first be given. The critical evaluation of the possibilities of development and fields of application of the various combustion-engine systems, wherever it is not clearly expressed in the publications referred to, represents the opinion of the author.

The state of development of the internal-combustion engine is in its main features generally known. It is used predominantly at the present time for the propulsion of aircraft and road vehicles and, except for certain restrictions due to war conditions, has been used to an increasing extent in ships and rail cars and in some fields applied as stationary power generators. In the Diesel engine a most economical heat engine with a useful efficiency of about 40 percent exists and in the Otto aircraft engine a heat engine of greatest power per unit weight of about 0.5 kilogram per horsepower.

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*"Die Verbindung von Verbrennungsmotor und Gasturbine." VDI Zeitschrift, May 13, 1944, p. 245.

[1]An article on the Velox supercharging process, which can be applied with advantage wherever the processes are accelerated by high pressure and where large quantities of heat are to be converted or exchanged, has recently appeared in this journal (reference 20).

The gas turbine - this term has come into use to denote a turbine working with combustion gas or hot air - is intended to combine the operating advantages of a machine without reciprocating motion of masses with the operational and economic advantages of the internal-combustion engine. The idea in itself is not new but has received new interest and a fresh outlook through the progress made in recent years in the field of heat-resistant materials and in the aerodynamic research applied to the compressors.

PRESENT STATE OF DEVELOPMENT OF THE GAS TURBINE

A brief review will first be given of the state of development of the gas turbine for the direct production of power as based on known test results and publications of recent years. (Among others see references 22 and 23.)

The dry gas turbine, which alone will be considered here, uses atmospheric air or the gases arising from the combustion of an air-fuel mixture and consists essentially of a compressor, a combustion chamber, and a turbine. The efficiency of the system depends strongly on the temperature of the working substance at the inlet to the turbine and on the efficient design of the turbine and the compressor. The maximum temperature in the gas-turbine cycle is, for reasons of strength, much lower than in the internal-combustion engine because, in the former case, the same structural parts are constantly subjected to high-temperature gas whereas in the latter the temperature peaks occur only for a short time and the structural parts therefore assume a mean temperature lying far below the maximum value.

Depending on whether isothermals or adiabatics are desired during the compression and the expansion, whether the heat is supplied at constant pressure or constant volume, whether and at what point of the cycle there is heat exchange, etc., numerous modifications of the gas-turbine process are possible (references 6, 23, 29, and 33). As in the case of the reciprocating engines, the gas turbines are divided into constant-volume turbines and constant-pressure turbines according to the nature of the combustion process (heat supply after compression either at constant volume or at constant pressure). (See reference 36.) This mode of classification has nothing to do with that of steam turbines for which the terms constant pressure or high pressure refer to the pressures ahead of and behind the impeller.

THE CONSTANT-VOLUME TURBINE

In the constant-volume turbine the explosion chambers are filled with air and fuel (combustion gas), a precompression of the charge being required not for carrying out the process but to attain useful efficiencies. In the combustion chambers the mixture is electrically ignited and burns at constant, or approximately constant, volume. Shortly before the end of the combustion the outlet valve opens and the high-pressure hot gases flow into the turbine to deliver the useful power.

The efficiency of the ideal work cycle of the constant-volume turbine is, as seen from the T-s diagram (fig. 1) greater than that of the constant-pressure turbine for equal precompression and equal maximum temperature. Because the ratio of the adiabatic heat drop in the turbine to that in the compressor is considerably greater in the case of the constant-volume turbine, the ratio is less dependent on the compressor efficiency, on the assumption of similar turbine efficiency, than in the case of the constant-pressure turbine. At a time at which there were as yet no centrifugal compressors with sufficiently good efficiency, successful results were therefore first obtained in tests on the constant-volume turbine, the development of which is closely associated with the name Holzwarth (references 6, 9, and 36). The periodically working explosion chambers offer a number of technically difficult problems. In the case of several particularly critical structural parts, water-cooling must be used to a large extent, the combustion chambers thereby constituting a source of considerable heat losses. Because the efficiency of an intermittently acting turbine is also likewise below that of a continuously acting turbine, the actual efficiencies lie far below those of the ideal process. The constant-volume turbine is economical only if the compressor is driven by a steam turbine whose boiler is heated with the exhaust heat of the gas turbine. Thereby the process becomes complicated and can be used with advantage only for very definite limited fields of application. With recent installations of this type, an efficiency of 20 percent was attained (reference 23), whereas an increase to 25 percent is expected (reference 6).

THE CONSTANT-PRESSURE COMBUSTION TURBINE

In its simplest form, the cycle of the constant-pressure turbine is formed of two adiabatics and two isobars (fig. 1(b)). A setup working on "open circuit" that takes air from the surroundings and discharges the combustion gas with still relatively high temperature into the air after expansion in the turbine has the advantage of very great simplicity (fig. 2) but the efficiencies attained at present with this system are still low in comparison with those of the constant-volume turbine (references 10 and 37).

The effective output of the gas turbine is the excess of the turbine output over the compressor output. Because in the case of the constant-pressure turbine the compressor power forms a considerable portion of the turbine power, the turbine is used to advantage only in connection with a high-efficiency compressor, which at present is available in the axial compressor. For a perfect gas and for definite assumptions as to the highest maximum and minimum cycle temperatures, the effect of the individual efficiencies of compressor and turbine on the over-all efficiency neglecting losses in the combustion chamber is shown in figure 3. From this figure we see that, for a gas inlet temperature in the turbine of 600° C and individual efficiencies of 0.85, the over-all efficiency of the setup is still below 20 percent. With present heat-resistant steels, temperatures of 600° C can today be used also for continuous operation. A gas-turbine unit working on this process built by Brown Boverie and Co., Baden, and installed in an electric-power generating station in Neuchatel, Switzerland, as an emergency power unit attained a brake efficiency of 18 percent at a power of 4000 kilowatts (reference 37). The inlet temperature of the combustion gas in the turbine was 552° C and the individual efficiencies of the compressor and the turbine were 84.6 and 88.4 percent, respectively. The compressor consumed 73.5 percent of the turbine power.

Improvement in Efficiency through Heat

Recovery from the Exhaust Gases

For the most economical compression ratio of the constant-pressure turbine according to the process of figure 1(b), the temperature T_4 of the combustion gas after expansion in the turbine remains higher than the temperature T_2 of the compressed air before entrance into the combustion chamber. The efficiency can therefore be improved by the heat transfer from the exhaust air to the compressed air before the entrance of the latter into the combustion chamber. The heat to be supplied to the air by combustion decreases for constant power by the quantity of heat exchanged (fig. 1(c)). As shown in figure 4, considerable increases in the efficiency are thereby made possible although the heat exchanger involves an increase in the size of the system.

The Gas-Turbine Process with Closed Circuit

A constant-pressure turbine work-process developed by Ackeret and Keller (references 1 and 11) and applied by the firm of Escher Wyss differs from the one described previously in two respects: During compression an isothermal change of state is approximated through intercooling and the work medium works with higher density in a closed circuit (fig. 5). As a result of the higher density, the dimensions of compressor, turbine, and heat exchanger for the heat recovery are smaller. As in the case of the boiler, however, the heat of combustion must be supplied to the work medium (air) through an "air heater" consisting of combustion chamber and heat exchanger. The gas-turbine unit, working according to this process, is termed by the inventors an "aerodynamic heat engine."

The process is based on the so-called double-isothermal cycle (fig. 6). The working gas, with heat Q_0 removed, is isothermally compressed from state 1 to state 2. Between states 2 and 3 the gas is heated at constant pressure from the lower temperature $T_2 = T_1$ to the upper temperature $T_3 = T_4$, the total quantity of heat of the gas flowing back from the turbine to the compressor being transferred so that the temperature is again lowered to T_1. Only during the isothermal expansion is the heat Q supplied externally to the turbine. Inasmuch as in the case of the ideal process the heat added and the heat removed are throughout in a state of equilibrium, that is, occurs with the temperature difference zero, all individual processes are reversible and hence the efficiency is equivalent to that of the Carnot cycle. The process can be only approximately realized and, as seen in figure 7, would require a unit with a great many parts because not only the air heater and the heat exchanger but also the intercooler and the intersuperheater are required between the individual compression and expansion stages. In order to simplify the unit, the intersuperheating is dispensed with in the process shown in figure 8(b).

The advantage of isothermal as compared with adiabatic compression in the case of the constant-pressure gas turbine for processes between equal temperatures and with the same pressures is readily seen from a comparison of the T-s diagram (fig. 8). The heat supplied Q is in both cases the same but the heat to be removed to the outside is in the case of the isothermal compression smaller and hence the heat converted into work greater. The lower temperature at the end of the isothermal compression requires no increased heat supply in the combustion chamber because a greater heat recovery from the gases leaving the turbine is possible. The isothermal compression naturally brings about greater efficiency only if a correspondingly larger heat exchanger is actually used to transfer the greater heat quantity.

The isothermal compression produced by intercooling can naturally also be used in the case of the open circuit but, due to the lower air density, leads to essentially larger dimensions of the heat exchanger and intercooler than in the case of the closed circuit. In order to approximate the isothermal expansion in the turbine in the case of the open circuit, combustion in stages has been proposed (reference 16) and this would require a subdivision of the combustion chamber and interconnection between the individual turbine-stage groups.

Comparison between the Open and Closed Circuits

Because in the case of the closed circuit the combustion gases come in contact only with the air heater but not with the compressor and the turbine, the danger of corrosion and wear of the blades is avoided in using fuels containing tar and ash. Corrosion of the blades produces a considerable lowering in the efficiency, whereas the wear produces in addition an impairment of the durability.

In the closed circuit, the power can be regulated for unchanged speed and unchanged position of the operating point in the characteristics of turbine and compressor, through the density (pressure head) of the working medium in the circuit alone, whereby, as compared with the open circuit, better fractional load efficiencies are possible. However, the frequent pumping of the unit for rapid load changes involves losses.

For the closed circuit, heat exchangers are required as is not the case for the open circuit: A precooler for cooling the circuit air before entry into the compressor, an air heater for transferring the heat from the combustion gas to the circuit air before entry into the turbine, and an air preheater for the combustion air to utilize the heat still remaining in the combustion gas after exit from the air heater. Because the combustion gas, in the most favorable case, can leave the air heater with the inlet temperature T_3 of the circuit air, which, however, on account of the heat already taken up in the heat exchanger is sufficiently high, there would be a considerable heat loss and hence decrease in efficiency without the air preheater.

The closed circuit is more restricted in the application of high gas temperature than the open circuit because the heat-exchanger tubes in the circuit-air heater assume a temperature above the maximum cycle temperature, whereas the turbine blades, which form the critical point in the case of the open circuit, remain below the maximum cycle temperature and in any given case may be held below it by cooling. The requirement of heat-resistant steels is limited in the case of open circuit to the turbine blades, whereas for the closed circuit also a part of the air-heater tubes must be of high-quality material.

With the need for application of the circuit-air heater and the combustion-air preheater, the unit with closed circuit requires many parts and in spite of reduced dimensions of the turbine and the compressor becomes more bulky than in the case of the open circuit. The circuit-air heater and combustion-air preheater are in their dimensions entirely comparable with the boiler units of steam-power installations because the heat transfer occurs on one (air heater) or on both sides (air preheater) only at atmospheric pressure. The Velox process may be applied to this heat transfer (references 11 and 20) but in view of the tube temperatures of the air heater, the heat transfer on the side of the combustion gas may not be arbibrarily raised.

If heat recovery is dispensed with at the expense of lowered efficiency, a very simple gas-turbine unit from the structural and operational viewpoints is possible with the open circuit, which also meets with high requirements as reqards bulk and weight inasmuch as with the open circuit the combustion chamber can be held to small dimensions because of the greater air density under which combustion occurs.

The advantage of the closed circuit lies in the greater applicability of solid fuels and in the better possibility of utilizing the intercooling and heat recovery, the space and weight requirements of which are less as compared with those of the air heater. This process is therefore in particular suitable for stationary units where it enters less in competition with the internal-combustion engine than with the steam turbine.

COMBINATION ENGINES

A combination engine is to be understood as a unit consisting of an internal-combustion engine, a turbine, and a compressor. Depending on where the power is taken off, the type of coupling of these three machines, and the degree of the supercharging, three cases may be distinguished:

the exhaust-gas turbosupercharging,
the propellant-gas process, and
the high supercharging.

The exhaust-gas turbosupercharging has for its object the utilization of the energy still remaining in the working gas exhaust. The adiabatic work recoverable by continuing the expansion from this state to the charging pressure amounts for the Diesel engine from 25 to 30 percent and for the Otto engine from 35 to 40 percent (on account of the smaller expansion in the cylinder) of the indicated work of the engine (reference 31). Inasmuch as only a small fraction of this can be regained, the continuation of the expansion in a second stage by applying an exhaust-gas turbine alone would not be of advantage unless a compressor is simultaneously used, that is, the compression is also made two-stage and an increased charge thus supplied to the engine.

For the exhaust-turbosupercharging system, the internal-combustion engine constitutes the principal machine from whose shaft the useful power is taken off. Between turbine and centrifugal compressor (supercharger), there must be equality of output and this determines the ratio of the supercharger pressure to the pressure ahead of the turbine. The turbine impeller and the compressor wheel can, as a rule, be so designed that they run with the same speed and may therefore rotate on a common shaft. The combining of the exhaust-gas turbine and the compressor to form the exhaust-gas turbosupercharger or the supercharger unit possesses the following principal advantages as compared with other types of supercharging:

1. The power for driving the supercharger is obtained from the energy of the exhaust gases.

2. No gearing is necessary and there is a certain freedom in the arrangement of the supercharger unit.

In the case of the propellant-gas process the useful power is taken from the turbine shaft so that the turbine becomes the principal machine. The internal-combustion engine drives a (piston) compressor, which supplies air for charging the former and these two machines must therefore have equal output. The exhaust gases of the internal-combustion engine (Diesel) are raised to a definite pressure and together with the scavenging part of the air through the cylinder form the propellant gas for the turbine. The supercharged Diesel engine thus, in addition to its function as drive machine for the compressor, takes over the part of the "combustion chamber" for the gas turbine. This combination of compressor and highly supercharged internal-combustion engine is denoted a propellant-gas producer (references 33 and 39).

In the process denoted high charging, both the compressor and the turbine are coupled to the shaft of the internal-combustion engine as a result of which there is somewhat greater freedom in the dimensioning and individual powers of these three machines.

Proposals for designing various combination engines after the rapid development of the combustion and steam engines at the turn of the century, appear relatively early in patents and in the technical literature (references 2, 13, and 40).

THE EXHAUST-GAS TURBOSUPERCHARGING OF FOUR-STROKE DIESEL ENGINES

The exhaust-gas turbosupercharging of four-stroke Diesel engines is at the present time carried out exclusively as "Büchi Supercharging", the chief characteristic of which is the exhaust-gas driven turbo-supercharger with subdivided exhaust piping (reference 3). This type of charging is the last stage of a long development in which the following steps, according to the patents granted, may be traced: DRP. No. 204,630, v. 16, Nov. 1905 describes a unit consisting of a centrifugal compressor, four-stroke internal-combustion engine, and turbine, all three machines working on a common shaft. The air or the fuel-air mixture is compressed in the centrifugal compressor as far as possible isothermally to several atmospheres and after being cooled is supplied to the internal-combustion engine. In contrast to the designs prevalent until that time, not only a part but the entire exhaust gases entered the turbine with greatly increased pressure. Originally Büchi had in mind raising the pressure ahead of the turbine up to the pressure at the end of expansion in the cylinder, for in one example (reference 2), an isothermal precompression in the centrifugal compressor up to 3 to 4 atmospheres and a pressure ahead of the turbine of 16 atmospheres was assumed. The turbine was in this case to take over the main load and the internal-combustion engine was essentially to drive only the compressor.

There are two factors operating against the realization of a process with so high an increase of the exhaust-gas pressure above the charge pressure. The residual gas with high pressure remaining in the compression space of the cylinder at the exhaust stroke will expand at the intake stroke and hinder complete charging of the cylinder with fresh charge. Even if the residual gas is expanded through a special valve to the surrounding pressure and a full charge thereby made possible, the ratio of the work done during the exhaust stroke by the piston against the high pressure in driving the gas from the cylinder (area 4-1-7-6 in fig. 9) to its expansion work in the turbine (area 4-5-6-7) is so unfavorable that, with account taken of the unavoidable throttling, friction, and heat losses, no appreciable gain would result.

The realization of the effect of the residual gas led Büchi to a charging process (DRP. No. 454,107, v. 27.3, 1921, Priority, v. 2., Nov. 1915) in which the pressure in the cylinder during the exhaust stroke is made approximately equal to the pressure of the precompressed charge in a certain load range and even somewhat lower. The common coupling of engine, turbine, and compressor is dispensed with, the turbine driving the compressor. Moreover, the simultaneous opening of inlet and outlet valves (valve overlap) at the end of the exhaust stroke is provided for in order to be able to scavenge the residual gas out of the cylinder through the charging pressure, which lies above the exhaust pressure. In this way, without increasing the weight of the charge, there is also obtained an increase in the volume of the charge, which is greater the larger the "dead space," that is, the lower the compression ratio of the engine. Because the scavenging furthermore lowers the temperature of the charge and of the combustion-chamber walls, it becomes of greatest importance to make possible high supercharging.

The Impact Turbine

Figure 10 shows the P-v diagram of the ideal work process of a combustion engine with exhaust-gas turbosupercharger for which the charging pressure is held above the pressure assumed constant ahead of the turbine. An exhaust-gas turbine with an approximately constant inlet pressure above that of the surroundings, which would be established in an exhaust piping acting as reservoir for a multicylinder engine, is denoted as an impact turbine (references 8 and 31). A utilization of the exhaust energy represented by the area 1'-4-5 is here dispensed with insofar as it does not contribute, through turbulence of the flow energy, to heating of the gas and hence to a volume increase from point 5 to point 8. The adiabatic work of expansion corresponding to the area 6-7-8-9 must be greater by the amount of the losses in the turbine and the supercharger than the adiabatic work of compression corresponding to the area 0-1-10-6.

The adiabatic heat drop in the turbine amounts to $\Delta i_T = i_8 - i_9$, where the symbols have the meaning indicated in figure 10, and the adiabatic pressure head expressed in heat units in the supercharger is $\Delta i_L = i_1 - i_0$. If G_L denotes the amount of charge to be compressed and G_A that of the exhaust gas, η_L the supercharger efficiency, and η_T the turbine efficiency, then the condition for the equality of the exhaust-gas turbine and compressor output is

$$(i_1 - i_0)\, G_L/\eta_L = (i_8 - i_9)\, G_A\, \eta_T \qquad (1)$$

Substituting the pressure ratios there is obtained

$$\left[\left(\frac{p_1}{p_0}\right)^{\frac{\kappa_L-1}{\kappa_L}} - 1\right] = \left[1 - \left(\frac{p_8}{p_9}\right)^{\frac{\kappa_A-1}{\kappa_A}}\right] \cdot \frac{C_{p_A}}{C_{p_L}} \cdot \frac{G_A}{G_L} \cdot \frac{T_A}{T_L} \eta_L \eta_T \qquad (2)$$

where C_{p_L} and C_{p_A} are the specific heats at constant pressure, κ_L and κ_A the adiabatic exponents, $T_L = T_1$ the air inlet temperature in the compressor, and $T_A = T_8$ the gas inlet temperature in the turbine. For otherwise equal ratios, therefore the pressure ratio producible in the supercharger is directly proportional to the absolute gas temperature T_A ahead of the turbine and to the turbine and the supercharger efficiency η_T and η_L and inversely proportional to the air temperature T_L ahead of the supercharger.

The production of an effective scavenging pressure drop, that is, a sufficient pressure difference between the intake piping and the exhaust piping for scavenging the residual gases, offers difficulties on account of the relatively low exhaust temperature of the Diesel engine with the simple impact turbine. A pressure ratio of 1.5 in the compressor, for example, for an exhaust temperature of 550° C, an air inlet temperature into the compressor of 25° C, and a pressure ratio of 1.3 in the turbine, requires an efficiency of the charging unit of $\eta_L \cdot \eta_T = 60$ percent which, on account of the relatively small dimensions of the machines with the simple impact turbine, is not readily attainable and was not attainable at the start of the supercharging development.

The Exhaust Turbine

In a further stage of development, therefore, Büchi divided the exhaust piping in such a manner that only the cylinders with at least 240° crank-angle phase difference of ignition exhausted to the same pipe (DRP No. 568,855, v.19, Nov. 1926, Priority, v. 30, Nov. 1925). (See fig. 11.) The individual pipe lines are led to separate nozzle chambers of the exhaust-gas turbine. The exhaust-gas turbocharging of Diesel engines through this process, which is today known as "Büchi turbocharging" first became practical. The exhaust impulses produce in the exhaust piping a strongly fluctuating pressure variation of which the section of the low pressures was used for scavenging the combustion space under simultaneous opening of the inlet and outlet valves (references 3, 28, 30, and 34).

Figure 12 shows as an example the oscillographically measured pressure variation in the exhaust piping of a Diesel engine with Büchi-charging. Because the scavenging is cut off before the pressure in the piping has again begun to rise due to the exhaust impulse from the next cylinder, this process makes possible the use of pipes of small volume in which the pressure peaks are more marked. If the turbine-nozzle cross sections are made large enough so that the exhaust impulses may flow off immediately without appreciable building-up of pressure, the turbine is denoted as an exhaust turbine (references 8 and 31). The conversion of the energy takes place both over the pressure waves traveling through the piping, which are first converted into velocity in the turbine nozzles, and over the velocity already directly produced in the outlet valve.

It is seen from figure 12 that the scavenging pressure drop between the charging pressure, which for multicylinder engines with common intake piping is subject to only slight fluctuations, and the pressure valley during the scavenging period is considerably greater than would be the case for a constant mean pressure in the exhaust piping. Because the high pressures occur only in the neighborhood of the piston top dead center, that is, over a period in which the piston motion is small, the exhaust stroke work to be performed by the piston also remains small for this process.

The analysis of the energy conversion in the exhaust turbine is difficult because not only the pressure of the exhaust gas but also its temperature and its velocity are subject to strong fluctuations. If the mean temperature and mean pressure of the gas ahead of the turbine are used to compute the adiabatic drop, apparent turbine efficiencies of over 100 percent, and hence apparent over-all efficiencies of the supercharger unit of over 60 to 80 percent may be found (reference 28). With this mode of computation, there is not, on the one hand, formed the correct mean value of the static drop because for the pressure peaks above the mean pressure, greater exhaust-gas quantities flow through the turbine and, on the other hand, the kinetic energy produced in the outlet valves is not taken into account. The first effect is generally considerably greater because the velocity losses in the numerous changes in cross section and direction of the exhaust piping are considerable. The apparent turbine efficiency becomes higher the greater the proportion of the pressure and velocity impulses in the total energy.

The charging pressures attainable with the simple exhaust turbine, for which the pressure in the exhaust piping must drop during the exhaust process to the surrounding pressure, are of the order of magnitude of several tenths atmospheres gage pressure. This process, also at low exhaust temperature, makes possible a scavenging of the combustion space and charging.

If the residual gases are completely scavenged, any further increase in the charge is possible only through an increase in the charge weight. In order to attain the required higher charging pressure, an increase in the drop in the turbine is required and this leads to the application of the impact turbine with utilization of the exhaust impacts. The higher the charge pressure, the greater the decrease in the proportion of the pressure and the velocity impacts in the total energy. Because for a greater charge not only must the residual gas be scavenged to reduce the thermal stressing of the engine but fresh air must be passed through the combustion space, a greater scavenging pressure drop is desirable. The scavenging of the air results in an increase of the air consumption with increasing charge. The scavenging drop is higher the greater the individual efficiencies of blower and supercharger. Figure 13 shows the variation of charging pressure, mean pressure, and mean temperature of the exhaust gas ahead of the turbine plotted against the engine power for the case of a recent exhaust-gas turboblower for four-stroke Diesel engines.

Application of Büchi-Charging

The Büchi system of supercharging is variously applied for increasing the power of stationary, marine, and automotive engines. The charging unit is mounted either on the front side of the engine over the coupling or, particularly in the case of V-engines, over the engine itself. Because the space above the coupling is in most cases free anyway, the first arrangement as a rule requires no additional space and leads to somewhat longer but straight exhaust piping. The second arrangement is applicable only if space is available above the engine and leads to shorter but more sharply bent piping. Figure 14 shows a section through a Diesel engine charging unit consisting of exhaust-gas turbine and supercharger.

The increase in power attained through Büchi-charging of Diesel engines amounts to 60 percent and more for an increase in weight of 5 percent and less. The fuel consumption referred to the output is lowered by 3 to 5 percent through the exhaust-gas turbocharging, whereas in using a supercharger directly driven by the engine (mechanical charging) the fuel consumption would be increased by several hundredths.

EXHAUST-GAS TURBOSUPERCHARGING OF OTTO ENGINES

The power obtainable from unit weight of the charge is, in the case of the Otto engine, larger on account of the considerably smaller excess of the combustion air so that the blower power required for precompressing the charge is therefore smaller in relation to the engine power than in the case of the Diesel engine. For otherwise equal conditions, that is, equal surrounding and charging pressures, the saving attainable in fuel consumption through the use of an exhaust-gas turbosupercharger as compared with the mechanical supercharger is therefore smaller than in the case of the Diesel engine. The exhaust-gas turbosupercharging shows up to advantage only if high compression ratios are used.

Aircraft Engines

Exhaust-gas turbosupercharging is of decided importance for the aircraft engine for attaining a large high-altitude output. In this case the exhaust-gas turbosupercharging is not so much for the purpose of raising the charging pressure on the ground (boosting) as to maintain the pressure up to as high an altitude as possible, as the outside pressure drops with altitude. For this purpose, the exhaust-gas turbosupercharger is particularly well suited because it can operate well on the greatly expanded gas with decreasing air pressure and because the drop in the turbine increases as the supercharger power required increases with increasing altitude (reference 8), the supercharger unit thus being self-regulating to some extent.

Because, in the case of the Otto engine, the exhaust temperatures due to the smaller expansion in the cylinder, are considerably higher than in the case of the Diesel engine, there is available for equal compression ratio a greater heat drop and therefore a greater power in the turbine. In the case of the Otto engine, it is therefore easier to keep the charging pressure above the pressure ahead of the turbine and thereby produce the required pressure drop for scavenging the residual gases. The exhaust-gas turbosupercharger of the aircraft engine draws further advantage from the fact that the air temperature decreases with increasing altitude so that the supercharger power becomes smaller, equation (2). The pressures in aircraft-engine superchargers and therefore the supercharger powers become so great at high altitudes that they cannot be attained with an exhaust turbine but only with a high drop produced by a building-up of the inlet pressure. The energy of the exhaust impulses depends essentially only on the engine power and therefore remains approximately unchanged with altitude. Its ratio to the total turbine power decreases with increasing heat drop in the turbine so that for aircraft engines a subdividing of the exhaust piping may generally be dispensed with.

The difficulties of the exhaust-gas turbosupercharging of aircraft engines are chiefly of a structural nature. Because the aircraft engine represents the extreme case of light construction, very high peripheral speeds of turbine and blower wheel must be chosen in order to realize smaller dimensions. The high stresses that then arise can be controlled, however, only at not too high temperatures of the structural parts. The exhaust gases of aircraft engines whose temperature is of the order of magnitude of 1000° C are therefore either cooled before entry into the turbine (gas cooling) or a direct cooling of the critical parts of the turbine (structural part cooling) is provided (reference 32).

The development of exhaust-gas turbosupercharging of aircraft engines was undertaken in France by Rateau as far back as the first world war (references 4, 21, and 22). It is worth noting that the first aircraft engine with an exhaust-gas turbosupercharger already possessed an intake-air cooler for cooling the charge heated by the compression in the blower (figs. 15 and 16).

Automotive Engines

Recent attempts have been made to apply the exhaust-gas turbosupercharger also for raising the power of automobile Otto engines using generator gas. In reconverting for generator gas, there occurs what is known as a considerable power drop to be ascribed to the low heat value of the generator gas-air mixture and to the high suction in the intake piping and hence low volumetric efficiency of the cylinder due to the resistances in the intake-gas generator and in the gas cleaning apparatus. Through the exhaust-gas turbosupercharger, which can be mounted without any changes in the engine, the resistances are overcome and therefore the volumetric efficiency and power of the engine increased. Of particular promise is a supercharger process applied by Brown Boveri and Co. in which the blower compresses only air that divides into a stream leading to the gas generator and into a stream leading directly to the mixing apparatus (references 12 and 38). In the mixing nozzle, compressed air and gas are mixed and led to the engine under a pressure slightly above atmospheric.

CHARGING OF TWO-STROKE DIESEL ENGINES

The exhaust-gas turbosupercharging of two-stroke engines (reference 41) differs from that of the four-stroke engine mainly in the following two respects:

1. The two-stroke engine requires a greater quantity of air per unit power for scavenging the combustion gases from the cylinder than the four-stroke engine. The temperature of the exhaust gases is therefore lower and the energy available in the turbine smaller than in the case of the four-stroke engine.

2. In the case of the two-stroke engine, in order that charging of the cylinder should occur at all, there must exist for each load condition a positive scavenging pressure drop between the intake and exhaust piping. This positive drop cannot be supplied by the exhaust-gas turbosupercharger in starting and at low engine loads.

Whereas in the case of the supercharged four-stroke engine the greatest part of the combustion gases is removed by the piston and only the residual gas in the dead space is removed by the scavenging gas, in the case of the two-stroke engine the scavenging air must take over the entire work of removal of the combustion gases. The four-stroke engine draws in fresh charge also when, with decreasing load, the charging pressure becomes smaller than the pressure ahead of the turbine; whereas such operating condition cannot be maintained in the case of the two-stroke engine.

The magnitude of the scavenging pressure drop in the two-stroke engine depends on the type of the scavenging system (through flow, cross-flow, and reverse flow scavenging, etc.), the dimensions of the ports, the resistances in the scavenging and outlet passages, the ratio of the air discharged to the displacement volume, and the piston speed. In most engines the scavenging pressure drop for the required amount of air to be passed through is so high that particularly on account of the relatively low temperature of the exhaust gases it cannot be produced even at full load by the exhaust-gas turbosupercharger alone.

The difficulties of the exhaust-gas turbosupercharging of two-stroke engines are countered chiefly by two devices:

1. The engine retains the mechanically driven scavenging blower to which in addition the exhaust-gas turbosupercharger is connected (Curtis Patent DRP. No. 545,907, Priority v. 2, 12, and 24).

2. The compressor (or a piston blower) of dimensions corresponding to the required charging is driven by the engine shaft and the exhaust-gas turbine also delivers its power to the engine shaft. Through the fixed speed ratio between the engine and blower, the latter always produces a sufficient charging pressure. This combination leads to the type of charging denoted above as high charging.

The power increase attainable through exhaust-gas turbosupercharging is smaller in the case of the two-stroke engine on account of its lower exhaust-gas temperatures than for the four-stroke engine, whereas the dimensions of the charging unit are larger on account of the greater air consumption per unit power of the two-stroke engine. Exhaust-gas turbosupercharging has therefore not yet been commonly adopted for the two-stroke engine, the application of the turbosupercharger remaining limited to particular cases. It has been used with great practical success only in the case of the Junkers aircraft Diesel engine (references 21 and 22). This engine is a through-flow scavenged two-stroke Diesel engine with opposed pistons with divided-shaft construction, one piston controlling the inlet ports, the other the outlet ports. The exhaust-gas turbosupercharger is connected to a blower driven by the engine.

Recent tests have been conducted by Sulzer, Winterthur, on opposed-piston Diesel engines with through-flow scavenging (reference 24). A piston compressor that delivers the scavenging and charging air was directly connected to the engine crankshaft. The exhaust gases are slowed behind the engine and drive an exhaust turbine geared to the crankshaft (fig. 17). The charge can thereby be raised to very high mean effective pressures. The following table shows the relation between the mean effective pressure and charging pressure where the latter denotes the pressure in the exhaust piping between the engine and the exhaust-gas turbine:

Charging pressure, atmospheres absolute	1	2	3	6
Mean effective pressure, kilograms per square centimeters piston area	6	12	15	18

The power required by the piston compressor over a wide charging range is not greater than the power delivered by the exhaust-gas turbine. By this process, the highest known mean effective pressures of two-stroke engines have been attained.

The particular suitability of the opposed-piston engine for supercharging together with the high degree of scavenging brought about by the through-flow type of scavenging is chiefly a consequence of the small scavenging pressure drop made possible by the large inlet and outlet ports.

GAS TURBINES WITH PISTON-TYPE PROPELLANT-GAS GENERATOR

The Work Process

The fundamental idea of the propellant-gas process, that is, the charging of a combustion engine to such a degree that the required compressor power must be covered by the engine cylinders and the useful power is taken off the turbine, is not new and the testing and development work connected with various modifications already can look back on a relatively long history (reference 40). The principal advantages of the propellant-gas process are to be found in the possibility of a spatial separation of the propellant-gas generator (the "boiler" or "combustion chamber") from the turbine and in the possibility of attaining efficiencies resulting from the combination of the piston engine, which is suited for high pressures and temperatures, with the turbine, which is suited for low pressures and hence large volumes. The P-v diagram of the process has fundamentally the same appearance as that for the combustion engine with the exhaust-gas turbosupercharger (fig. 10). Because the compressor is driven by the internal-combustion engine, the work area of the engine 1-2-3'-3-4 must be greater by the losses in the engine and compressor than the work area of the compressor 0-1-10-6. The degree of attainable charging pressure is determined by this condition. Also with the propellant-gas process here described, there is obtained through the expansion of the exhaust gases a loss in work corresponding to the area 1'-4-5 (fig. 10). In the case of the nonsupercharged two-stroke engine, it is possible to utilize the exhaust energy for scavenging without using a scavenging blower (reference 5) but the utilization of this energy becomes relatively small as soon as the free outflow is obstructed. The indicated efficiency of the propellant-gas process therefore differs little from that of the piston engine. For driving the compressor, a highly charged four-stroke engine or two-stroke engine may be used but the two-stroke process is preferable because of its greater output per unit displacement volume and because of the better possibility of scavenging large quantities of air through the engine as is necessary in order to lower the exhaust-gas temperature and the thermal loading of the engine.

The Crank-Type Propellant-Gas Generator

If the compressor piston is driven over a crankshaft by the Diesel piston, a "crank-type propellant-gas generator" (fig. 18) is being dealt with. An installation of the Göta Works, Göteburg (reference 7) operating on this process uses a slowly running, simply acting, two-stroke engine with through scavenging, the inlet ports being controlled by the piston and an outlet valve in the cover through cams and rods. In order to assure ignition in starting and at low loads without having the combustion pressure in the cylinder too high at high load, the compression is regulated by the closing-time point of the exhaust valve as a function of the propellant-gas pressure behind the engine, that is, by the load. The actual compression amounts, at starting and at low loads, (propellant-gas pressure up to 0.5 atmosphere gage), to 85 percent of the piston stroke and at high loads with 3 atmospheres gage to only 40 percent of the piston stroke. As the lowest fuel consumption for this installation, the value of 185 grams per horsepower hour is given, which for an average lower heating value of the gas oil of 10,100 kilogram calories per kilogram corresponds to a brake efficiency of 33.8 percent. The supercharged two-stroke engine has a mean effective pressure of 7.5 to 8 atmospheres; the weight of the entire installation is 20 kilograms per horsepower. The process is exclusively intended for marine propulsion, as would be expected from the choice of the large slowly running engine and has already been put in practical operation in many installations.

The Floating-Piston-Type Propellant-Gas Generator

The raising in pressure of the exhaust gases and the high-pressure scavenging with the entire compression-air lead in the case of the floating-piston compressor (references 19 and 25) to the "floating-piston propellant-gas generator" in which, as in the case of the floating-piston compressor, the Diesel engine and compressor pistons are directly coupled to a crankshaft, whereby a particularly simple type of construction of the propellant-gas generator is obtained. Figure 19 shows the type of construction according to the patents of Pescara and Junkers. Through combustion of the charge in the Diesel engine, the opposed pistons are thrust outward and compress air in the compressor cylinder or cylinders, which take up the energy of motion of the pistons. Only a part of the air is displaced out of the compressors into the receiver formed by the engine housing, the remainder of the air, through its expansion, again throwing the pistons toward the bottom dead center and thereby compressing the new charge. In the Junkers design the energy of the back motion of the pistons is stored up only in the suitably dimensioned dead spaces of the compressor, whereas in the design of Pescara two air cushions, which are additionally built in for regulating purposes, take over a part of the energy. As soon as the pistons in their top-dead-center positions expose the ports in the cylinder walls, the combustion gases are scavenged into the propellant-gas piping and the Diesel cylinder is filled with the precompressed air.

Test installations with floating-piston propellant-gas generators have been developed by Pescara (reference 26) and Sulzer (reference 24). The efficiency of the propellant-gas process for the Sulzer installations is given as 35 to 40 percent and the propellant-gas pressure between engine and turbine as 5 to 6 atmospheres. For an increase of the mean effective pressure in the Diesel cylinder to three times that of the nonsupercharged engine, the built-up pressure behind the engine (propellant-gas pressure) must be raised sixfold. In order to attain still higher charging pressures and hence still smaller dimensions, in particular of the piston compressor, two-stage precompression of the charge air may be used, the first stage being formed by a centrifugal blower driven by an auxiliary turbine.

In order to evaluate the performance of the propellant-gas process, a knowledge of the weight-power ratio and the behavior in operation is necessary in addition to the efficiency and fuel consumption. As follows from test data thus far published (references 24 and 26) and from theoretical investigations (reference 39), the useful efficiency of the Diesel engine may thereby, in the most favorable case, be attained but not exceeded. Fundamentally, through the exhaust-gas turbosupercharging as compared with the nonsupercharged engine, an improvement in the efficiency is obtained that again decreases, however, with the present-day separate efficiencies of compressor and turbine in the range of very high charging pressures such as are required for the propellant-gas process.

As also regards weight, the crank-type propellant-gas generator can hardly compete with the Diesel engine particularly if the supercharged Diesel engine is used for comparison. The floating-piston-type propellant-gas generator, because the crankshaft is missing and the force transmission between the Diesel engine piston and the compressor piston is simpler, is as regards bulk and weight much superior to the crank-type propellant-gas generator. This advantage, however, does not show up to the same degree for multicylinder installations because the distance between cylinders of independent floating-piston-type propellant-gas generators cannot be made as small as in the case of the crankshaft engines.

In the case of the floating-piston propellant-gas generator, regulation constitutes a particularly difficult problem. As there are no uniformly moving masses, the piston energy stored by the compressed air must be based on the working stroke of the piston, that is, on the load. In addition to this adjustment of the piston energy to be stored in the dead space of the compressor or in the air cushion to the variable propellant-gas pressure and hence the variable volume of the combustion space, a number of other regulating functions must be fulfilled (as the adjustment of the propellant-gas quantity and pressure to the load of the turbine and the determination of the piston stroke for the most favorable operating values), which require a sufficiently complicated regulating apparatus on the reliable functioning of which the operation of the entire installation depends.

Because the propellant-gas generator, both the crankshaft- and floating-piston types, does not have any special scavenging pump, the scavenging being carried out with the high-pressure air of the compressor, the engine cannot idle. In order that the work cycle be maintained, air must constantly be compressed into the cylinder and propellant-gas constantly generated. With decreasing load of the turbine therefore, insofar as several gas generators work on one turbine, the individual machines must be successively disconnected, that is, for bridging over the individual partial load stages and for idling of the individual units, bypass valves around the turbine must be opened through which the propellant gas may escape without performing work. This regulation results in an impairment of the partial load efficiency.

FIELDS OF APPLICATION OF THE VARIOUS PROCESSES

In concluding, a brief discussion will be given of the fuel consumption, weight-power ratio, and several other determining characteristics of the different installations considered.

With the propellant-gas process, it is possible to operate a turbine with very good efficiency but the process does not have the operational simplicity of the simple gas turbine. Inasmuch as the propellant-gas process is to be considered as a particular case of the exhaust-gas turbosupercharged Diesel engine and uses the same fuels, it is to be compared chiefly with the Diesel engine. The most important characteristic of the propellant-gas process is the force transmission by means of compressed air and the consequent possibility of transforming the speed and torque and, by connecting together several units, of combining a large power in one turbine and therefore on one shaft. Because the propellant-gas process shows no advantages as compared with the Diesel engine with regard to fuel consumption, mode of operation, or space requirements, it will find application only in cases where the advantage of the speed and torque conversion is an important factor.

For direct power generation, the gas turbine finds the following fields of application:

1. Propulsion of large vehicles, in particular, forms of aircraft.

For this purpose, an operationally simple design (open circuit) using liquid fuel with only small waste heat recovery comes into consideration. According to the present stage of development, it is possible with such installations to attain, starting from a given output, the weight and space requirements of the internal-combustion engine and even to better them for very large outputs only, however, at considerably higher fuel consumption. The gas-turbine installation of the BBC Gas-Turbine Locomotive (references 18 and 35) possessed for an output of 2200 horsepower, a weight-power ratio of 12 kilograms per horsepower, and a brake efficiency of 17.5 percent. Diesel engines for locomotives are now designed with weight-power ratios of 8 to 12 kilograms per horsepower for brake efficiencies of 36 to 40 percent. The advantages of the gas turbine here lie in the possibility of using cheaper fuel, in the smaller oil consumption, and in the essentially simpler maintenance. For marine propulsion, more efficient installations would be necessary, which are possible with extensive waste heat recovery though at the cost of greater complication.

2. Peak and emergency loads.

The emergency current installation of Brown Boverie and Co. in Neuchatel has for an output of 4000 kilowatts a weight of 64 tons including generator and starter, which gives a weight-power ratio of 8 to 9 kilograms per horsepower for the gas-turbine installation alone without heat exchanger. This value lies considerably below the design weight of a stationary Diesel engine of equal output and intended for the same purpose, the corresponding value of which would be about 30 to 40 kilograms per horsepower. The efficiency of the gas turbine is 18 percent as compared with about 40 percent of the Diesel engine but the fuel consumption is only a secondary factor for application as an emergency installation.

3. Stationary Power Generation.

For this application only an installation using solid fuels or gaseous fuels obtained from the former with as extensive heat recovery as possible has any prospects of competing with other stationary power generators. With such installations, which would have, however, considerable space requirement, it is possible to exceed the efficiency of the steam turbine. For an increase in the efficiency of from 18 to 26 percent, there is required for the open circuit a heat-exchange area of about 1 meter squared per horsepower, increasing the weight of the installation by about 10 kilograms per horsepower. Each further increase in the efficiency makes necessary an increasingly larger heat exchanger and therefore a heavier installation (reference 27). For the application of solid fuels and for the attainment of high efficiency, the closed-circuit system is more suitable. The latter possesses, it is true, an air heater corresponding in its dimensions to the steam boiler but the apparatus for raising the efficiency, as intercooler and heat exchanger, are smaller and may be designed with smaller temperature and pressure losses than for the open-circuit system.

The advantages of the gas turbine as compared with the internal-combustion engine increase with the power. In the first place, the efficiencies of the gas turbine improve with increasing size; secondly, the weight-power ratio does not increase with the power, whereas in the case of the internal-combustion engine large outputs are attainable only through greater weight-power ratio (larger cylinder dimensions) or through a large number of cylinders.

From the preceding discussion, it may be inferred in what fields the internal-combustion engine will continue to maintain its position: In the first place, in the field of small outputs up to about 1000 horsepower where the gas turbine, on account of the lower efficiency of flow machines at small sizes, becomes too uneconomical and where the high-speed combustion engines can still use few cylinders. In the second place, also in the field of medium and large outputs wherever small space and weight and small fuel consumptions are simultaneously required. In particular the exhaust-gas turbo-supercharging, which likewise will continue to benefit from the further development of flow engines, makes possible not only a considerable lowering in the weight-power ratio but also a certain improvement in the efficiency, for example, through better utilization of the exhaust energy.

As to how far the simple gas turbine will succeed, mainly by way of an increase in the temperature, in entering the efficiency range of the internal-combustion engine also for small space and weight conditions, no final answer can at present be given.

Translation by S. Reiss,
National Advisory Committee
for Aeronautics.

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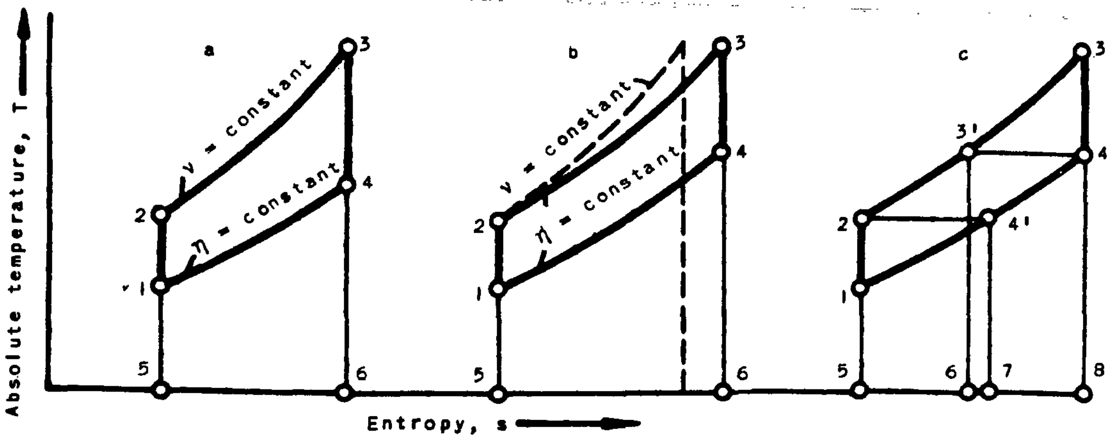


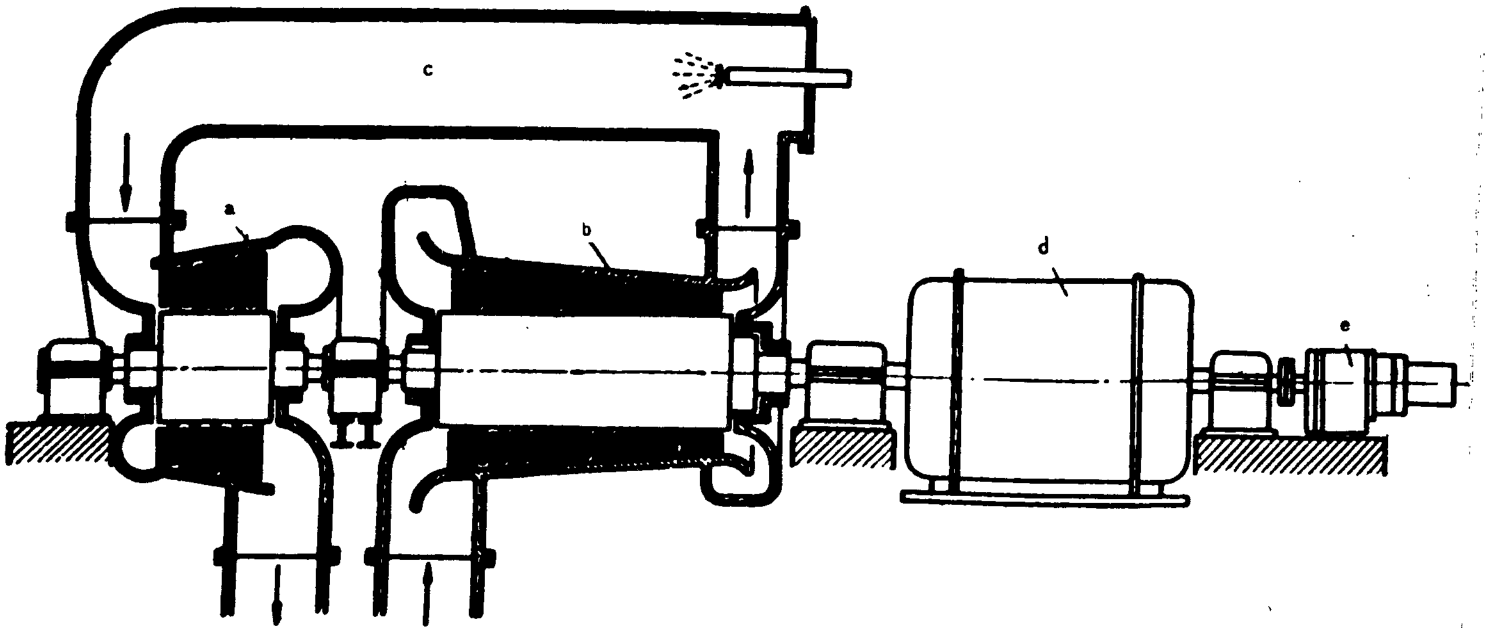
a: Exhaust turbine
Area 2-3-6-5 = heat supplied
Area 1-2-3-4 = work available

b: Constant-pressure turbine without heat recovery
Area 2-3-6-5 = heat supplied
Area 1-2-3-4 = work available

c: Constant-pressure turbine with exhaust heat recovery
Area 3'-3-8-6 = heat supplied
Area 1-2-3-4 = work available

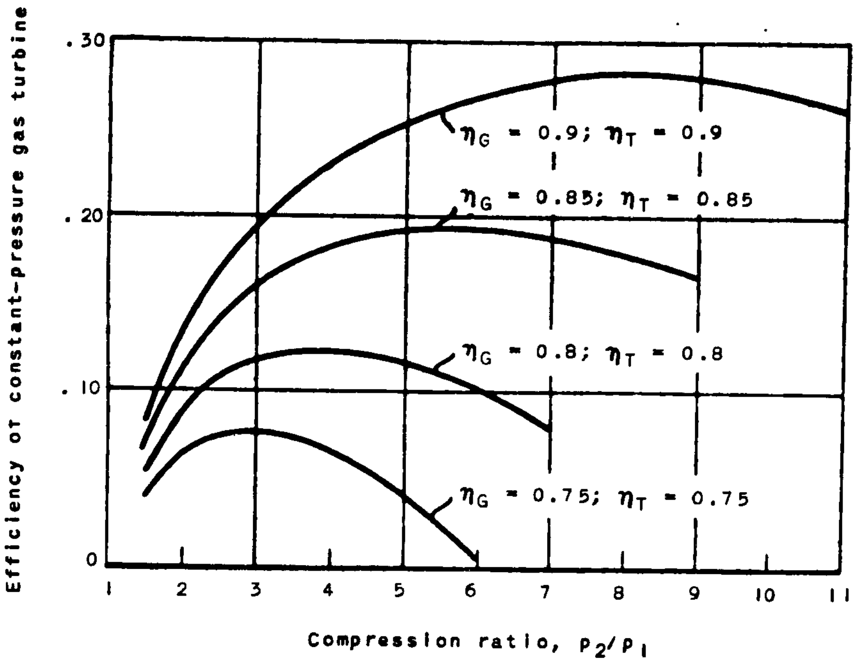
Area 2-3'-6-5 = 4'-4-8-7 = exchanged heat

Figure 1. - Ideal work cycle of the gas turbine, T-s diagram.



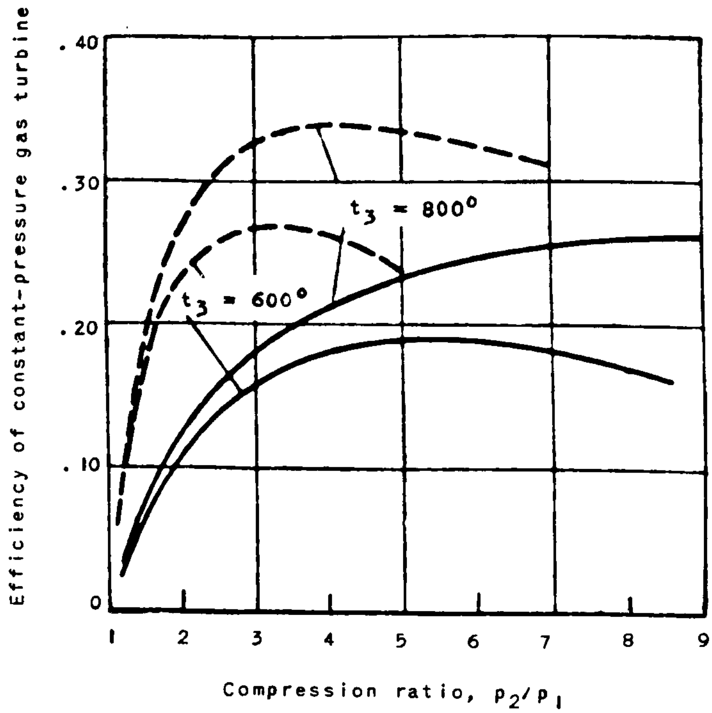
a. Turbine
b. Air compressor
c. Combustion chamber
d. Current generator
e. Starting motor

Figure 2. - Constant-pressure gas turbine with open circuit without exhaust heat recovery.



Ideal gas, lowest cycle temperature t_1 = 20° C; highest cycle temperature t_3 = 600° C. no loss in combustion chamber.

Figure 3. - Effect of compressor efficiency η_G and of turbine efficiency η_T on efficiency of constant-pressure gas turbine without exhaust heat recovery.

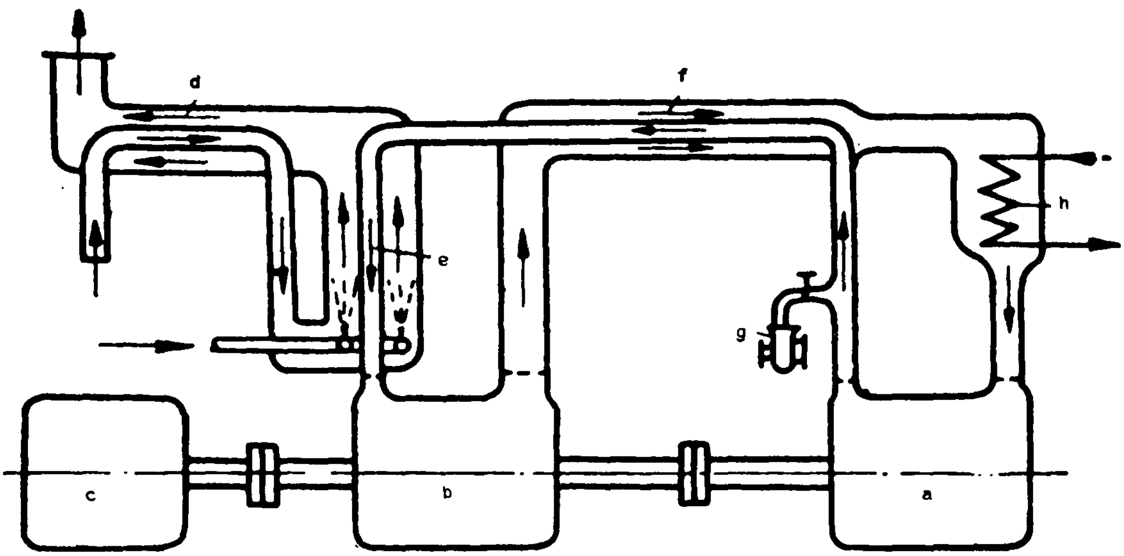


Ideal gas, no heat and pressure loss; compressor efficiency η_G = 0.85; turbine efficiency η_G = 0.85.

———— Efficiency without exhaust heat recovery

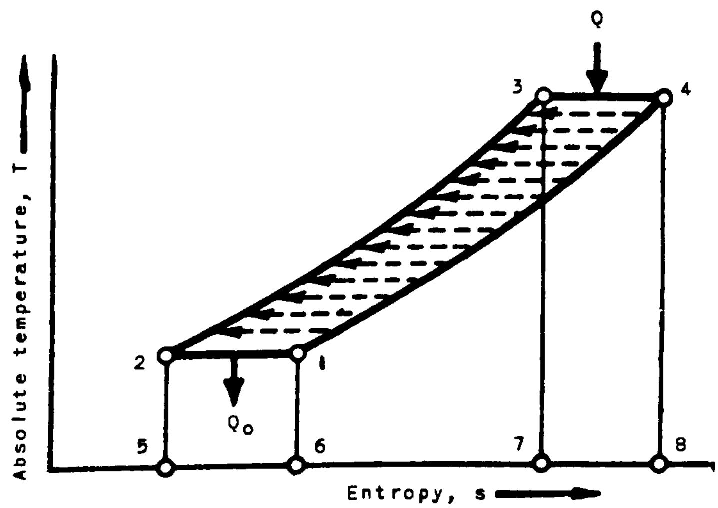
-------- Efficiency for transfer of 70 percent of heat from exhaust gases to compressed air before entrance into combustion chamber

Figure 4. - Efficiency of constant-pressure gas turbine as a function of compression ratio and gas inlet temperature t_3 in the turbine.



a. Compressor
b. Turbine
c. Current generator
d. Combustion-air preheater
e. Circuit air heater
f. Heat exchanger
g. Pump for regulating air in circuit
h. Precooler

Figure 5. - Work process of constant-pressure gas turbine with closed circuit according to Ackeret and Keller.

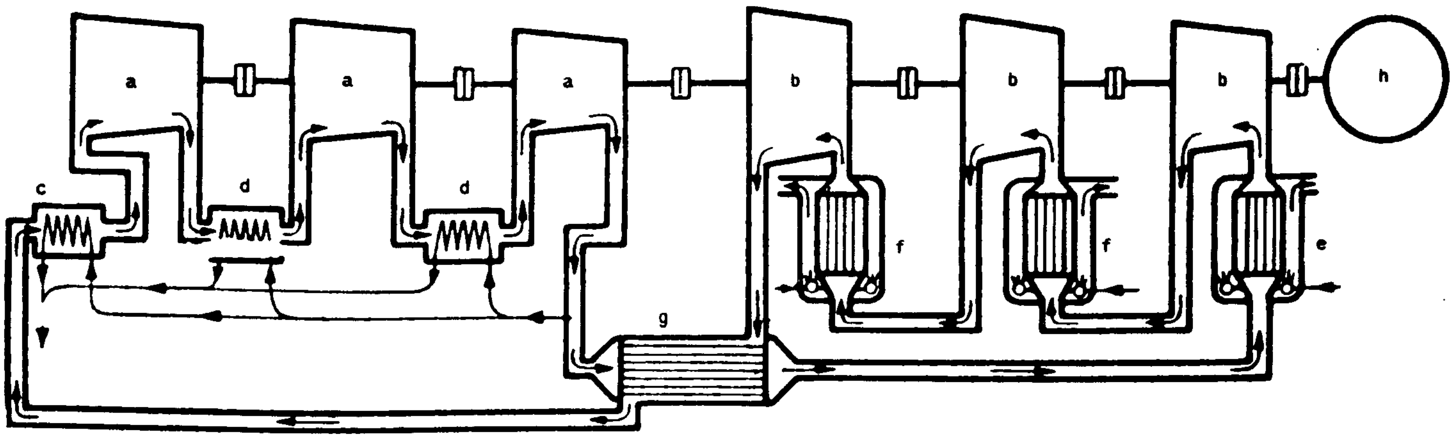


Dashed arrows denote heat exchange between compressed and expanded working gas.

Area 3-4-8-7 = heat supplied Q

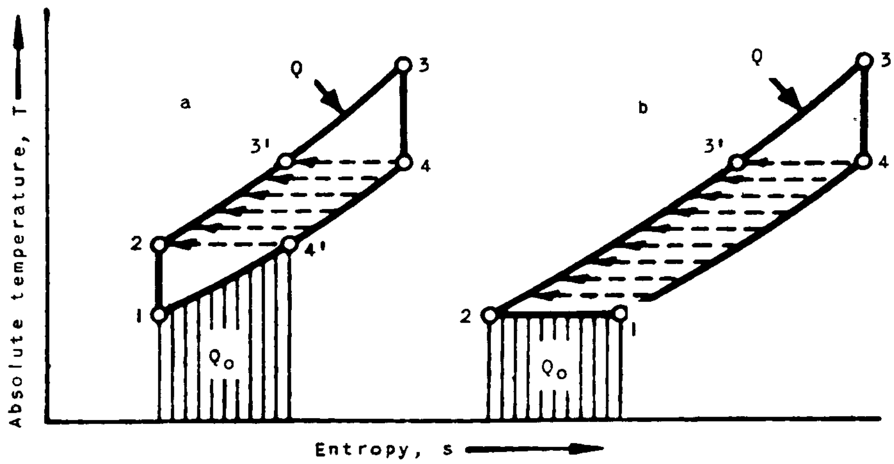
Area 1-2-5-6 = heat removed Q_0

Figure 6. - Double isothermal cycle represented in T-s diagram.



a. Compressor-stage groups
b. Turbine-stage groups
c. Precooler
d. Intercooler
e. Air heater
f. Intersuperheater
g. Heat exchanger
h. Current generator

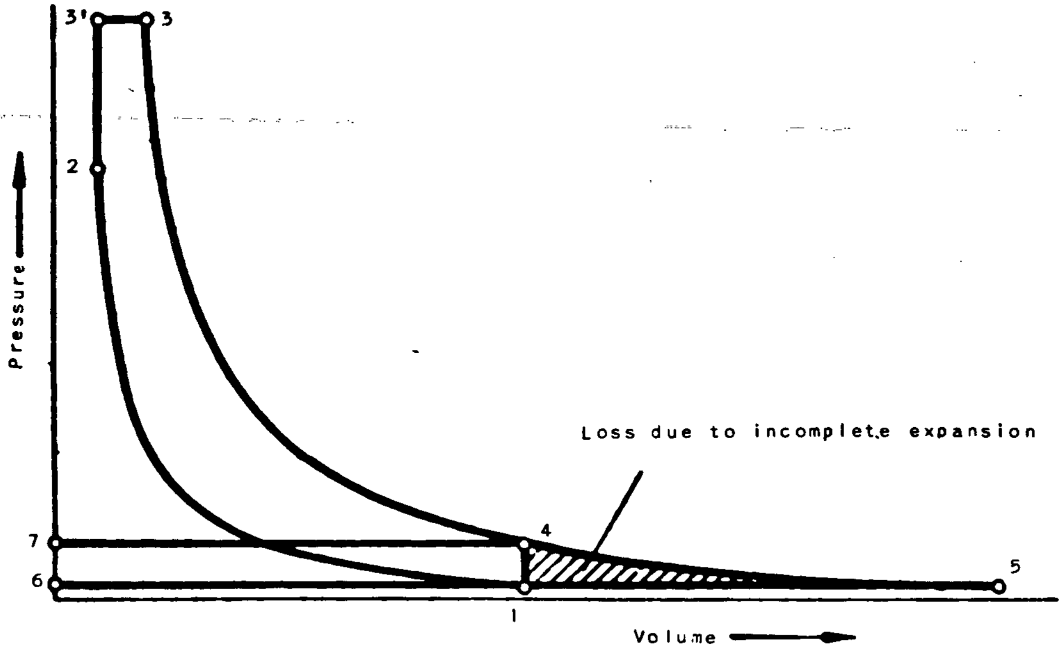
Figure 7. - Installation for approximate double isothermal cycle according to Ackeret and Keller.



Q = heat supplied, equal in both cases for equal pressures and compression ratios
Q_0 = heat removed

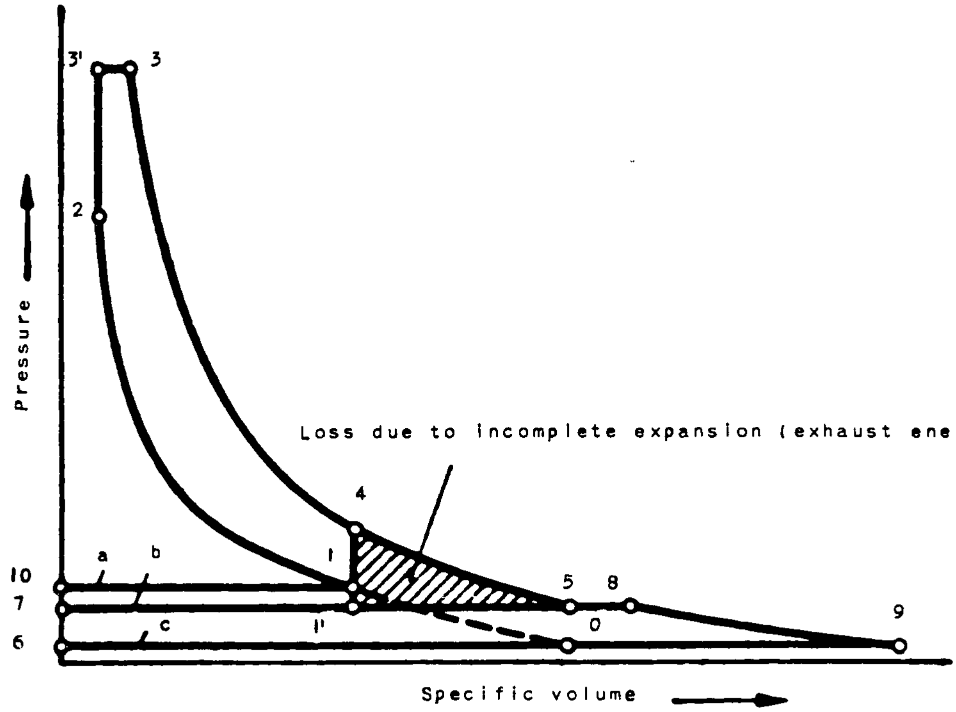
Area 1-2-3-4 corresponds to heat transformed into work. Heat recovery in exchanger is represented by dashed arrows.

Figure 8. - Ideal work process of constant-pressure gas turbine with exhaust heat recovery for adiabatic and isothermal compression.



Area 1-2-3'-3-4 = work obtainable in engine cylinder
Area 1-4-5 = loss due to incomplete expansion in cylinder
Area 1-4-7-6 = work during exhaust stroke when pressure at end of expansion is P_4 in cylinder
Area 4-5-6-7 = work obtained in turbine by using total expansion

Figure 9. - Ideal work cycle represented by P-v diagram of internal-combustion engine.

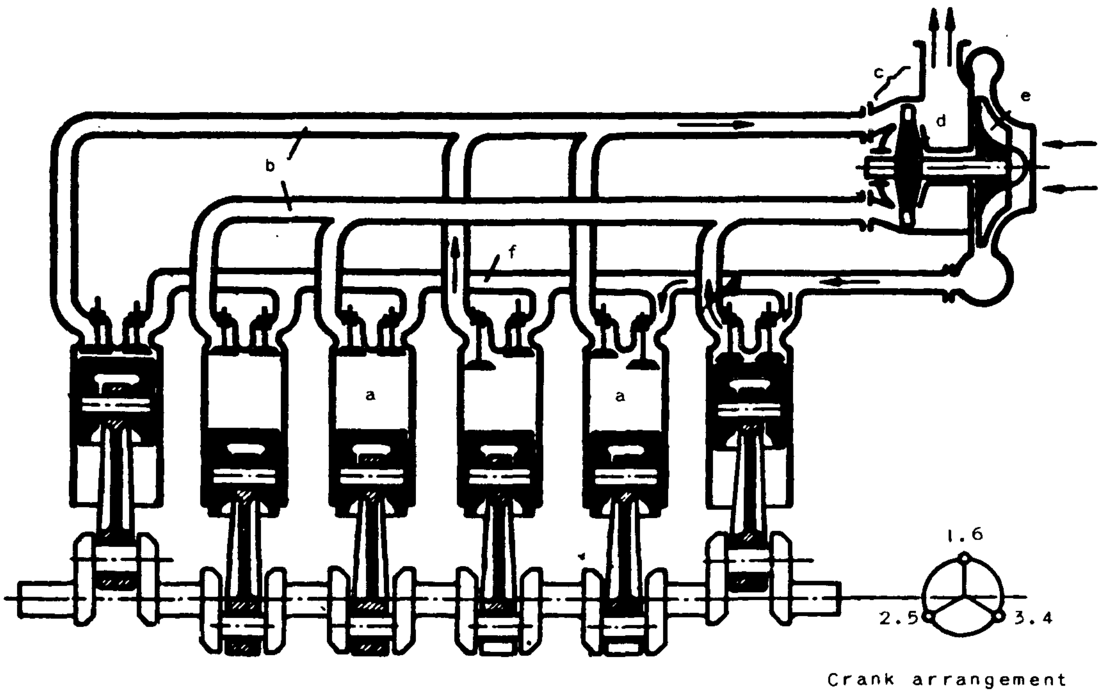


a. Charging pressure
b. Exhaust pressure ahead of turbine
c. Surrounding pressure

Area 1-2-3'-3-4 = work obtainable in engine cylinder
Area 6-7-8-9 = work obtainable in turbine
Area 0-1-10-6 = work to be expended in compressor
Area 1'-4-5 = loss due to incomplete expansion in cylinder

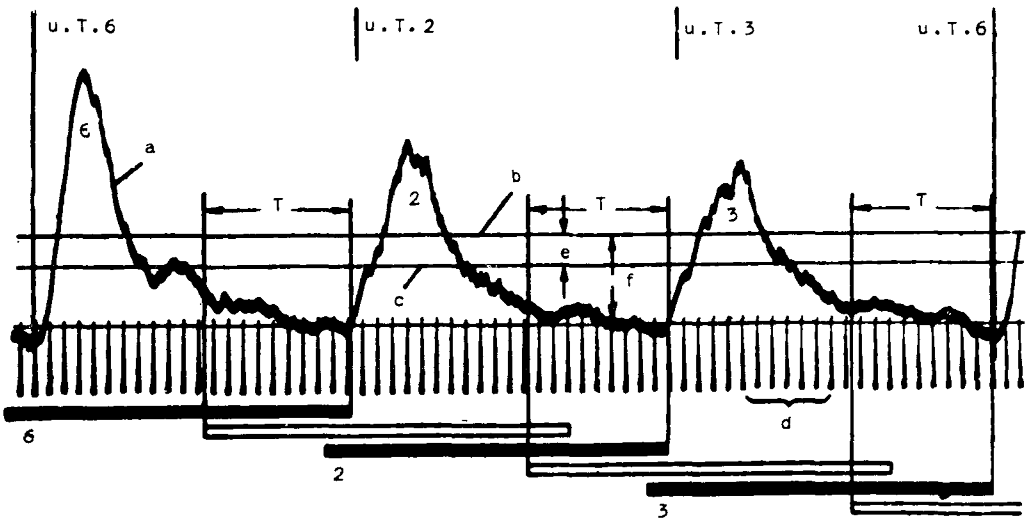
Diagram not drawn to scale

Figure 10. - Work cycle of combination engine with charging pressure above expansion exhaust pressure.



a. Engine cylinders
b. Exhaust piping
c. Charging unit with turbine impeller d and blower disk e
f. Charge-air piping

Figure 11. - Six-cylinder four-stroke Diesel engine with Buchi-charging.



b. Mean charging pressure
c. Mean pressure in exhaust piping
d. Time marks
e. Pressure difference between mean charging pressure and mean exhaust pressure
f. Scavenging time

Black bars = opening time of outlet valve
White bars = opening time of inlet valve
T, scavenging time

Figure 12. - Pressure variation (a) in exhaust piping of a Diesel engine with Buchi-charging.

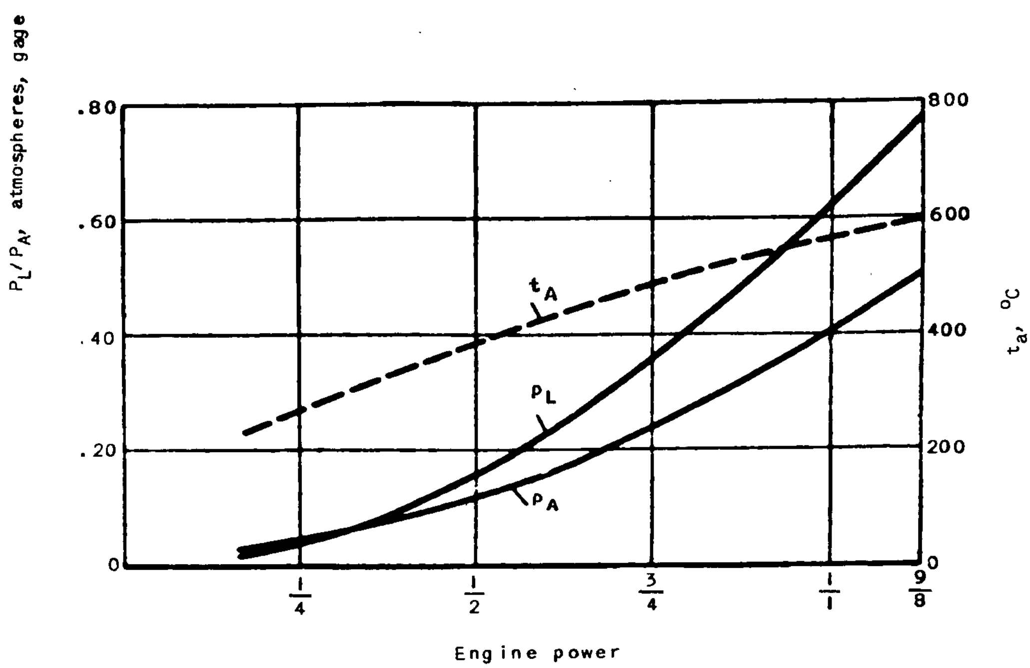
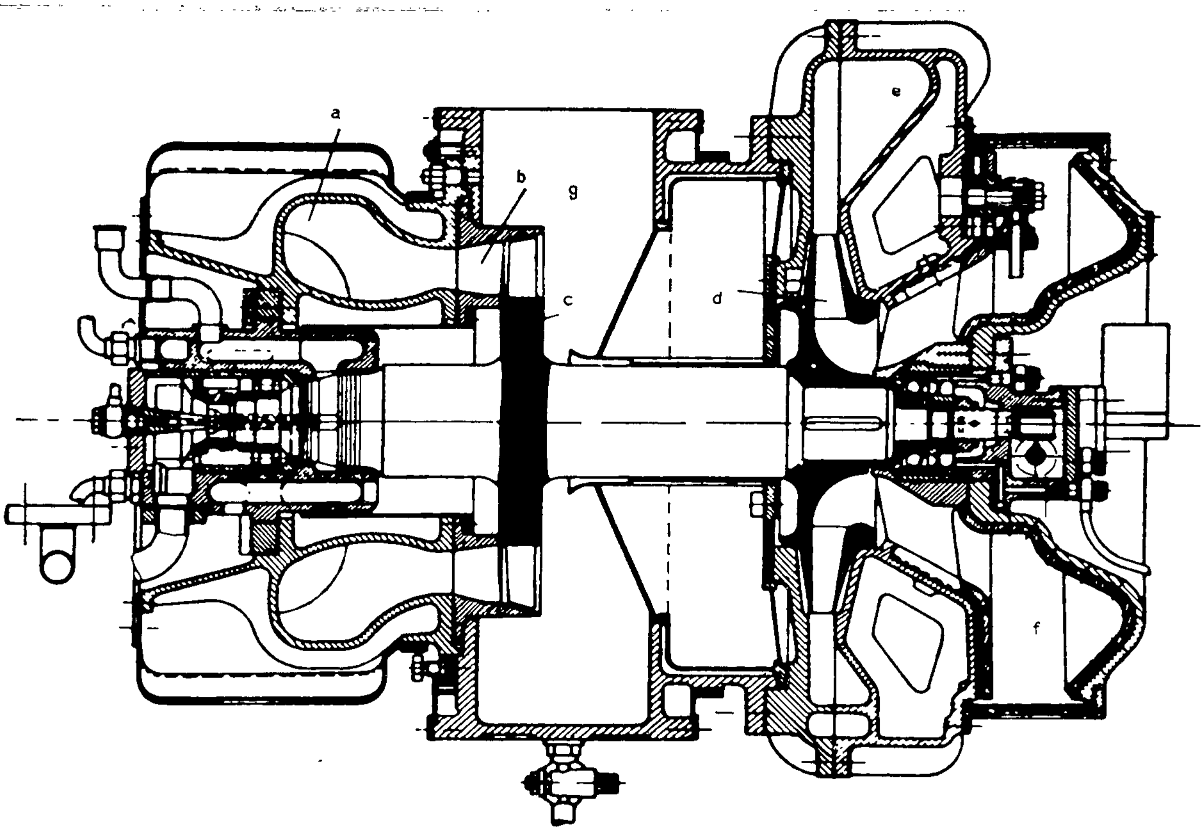
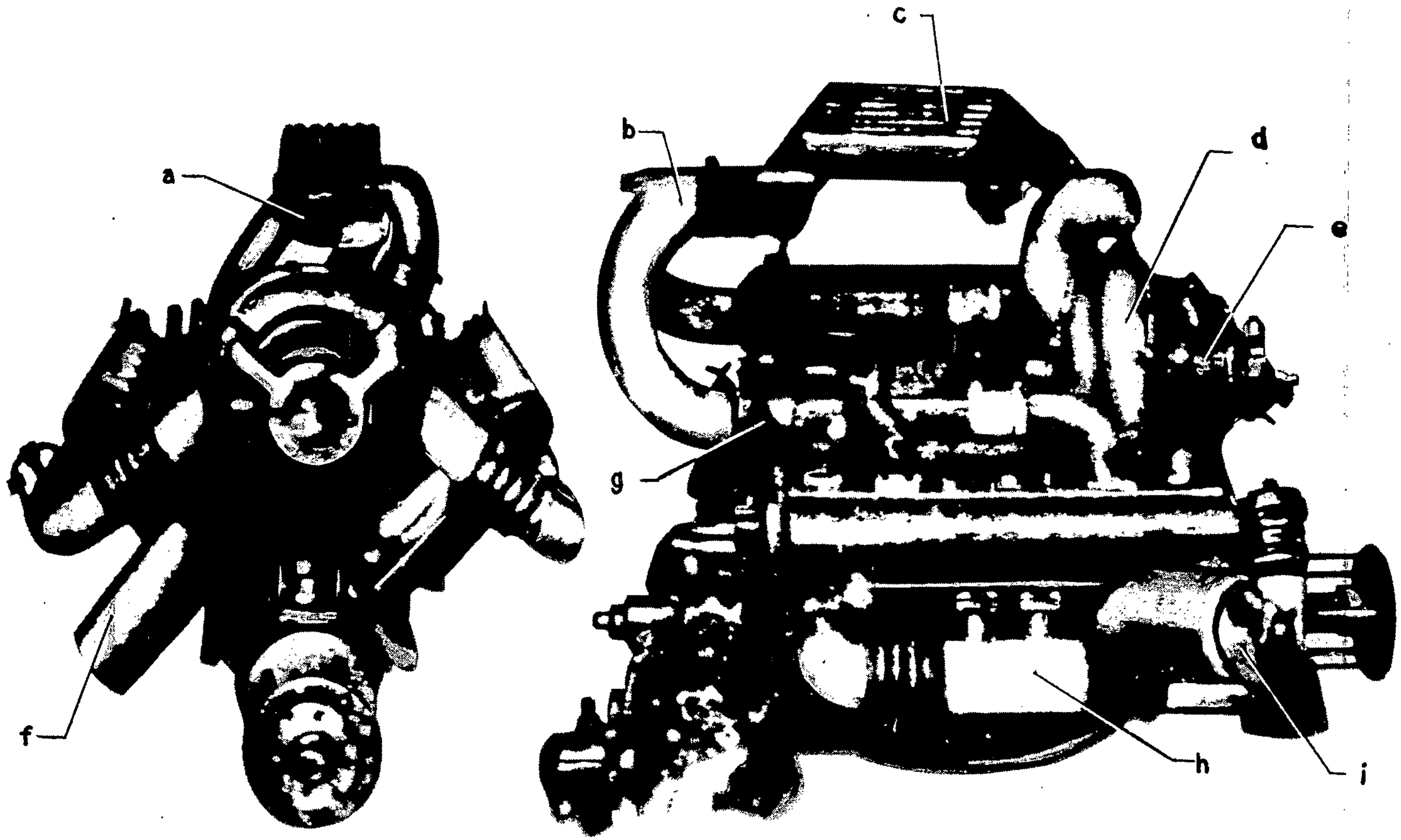


Figure 13. - Mean exhaust temperature t_a, charging pressure p_L, and pressure p_A in exhaust piping, measured with an inert manometer, of a four-stroke Diesel engine with Buchi-charging. Surrounding pressure about 1 atmosphere.



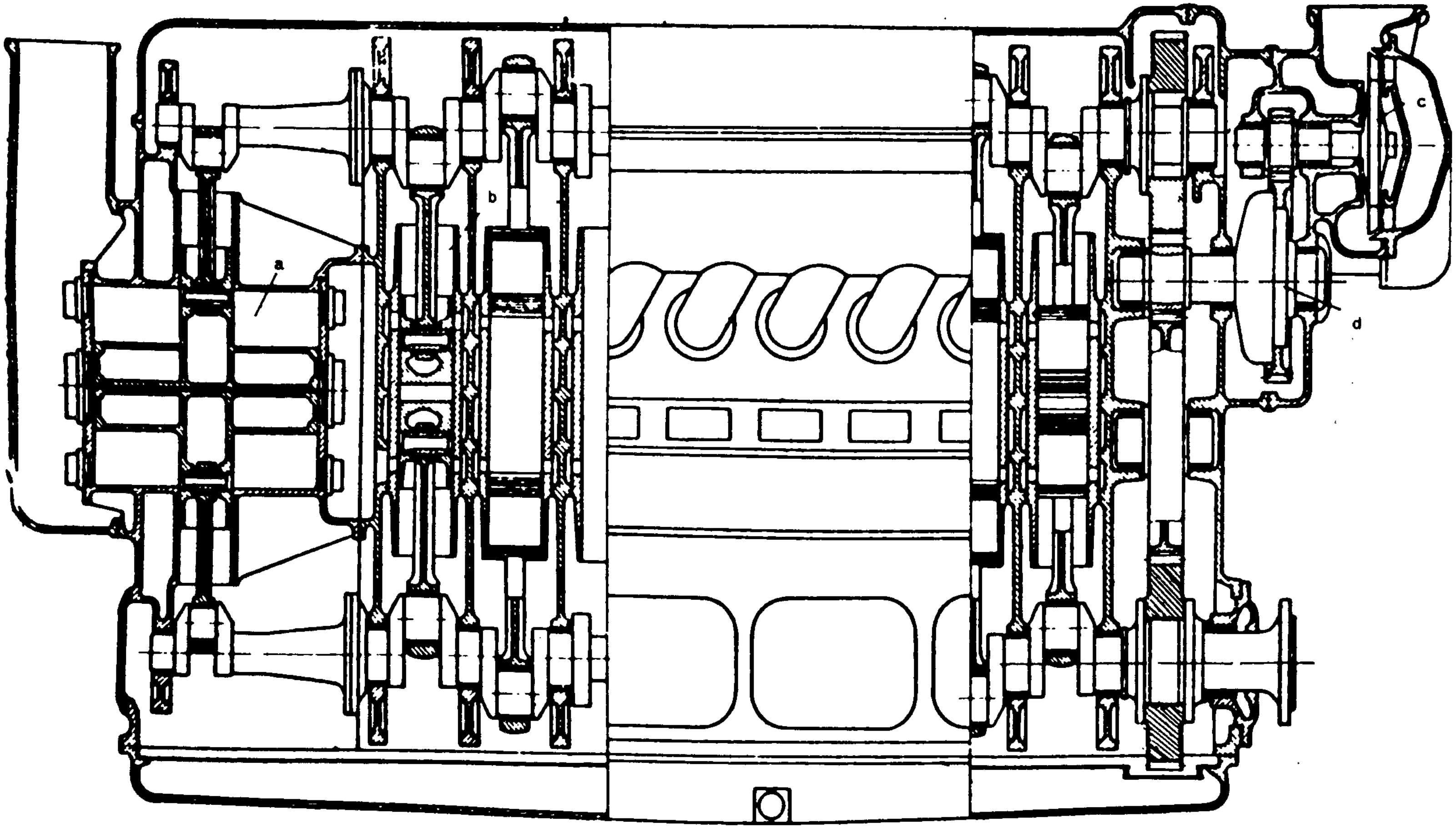
a. Gas-inlet housing
b. Guide apparatus
c. Turbine impeller
d. Blower disk
e. Pressure spiral
f. Air-inlet channel lined with sound-muffling material
g. Gas-outlet housing

Figure 14. - Section through supercharger unit.



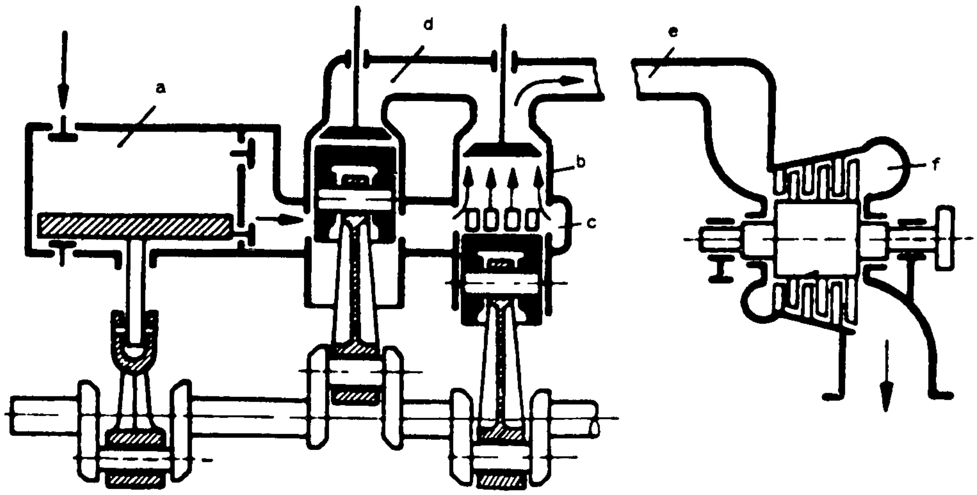
a. Scoop for intake air
b. Intake-air piping
c. Dry air cooler
d. Compressor
e. Exhaust-gas turbine
f. Exhaust-gas outlet to turbine
g. Pressure carburetor
h. Exhaust piping
i. Regulating valve

Figures 15 and 16. - Oldest aircraft engine with exhaust-gas turbosupercharger of Rateau type (reference 22).



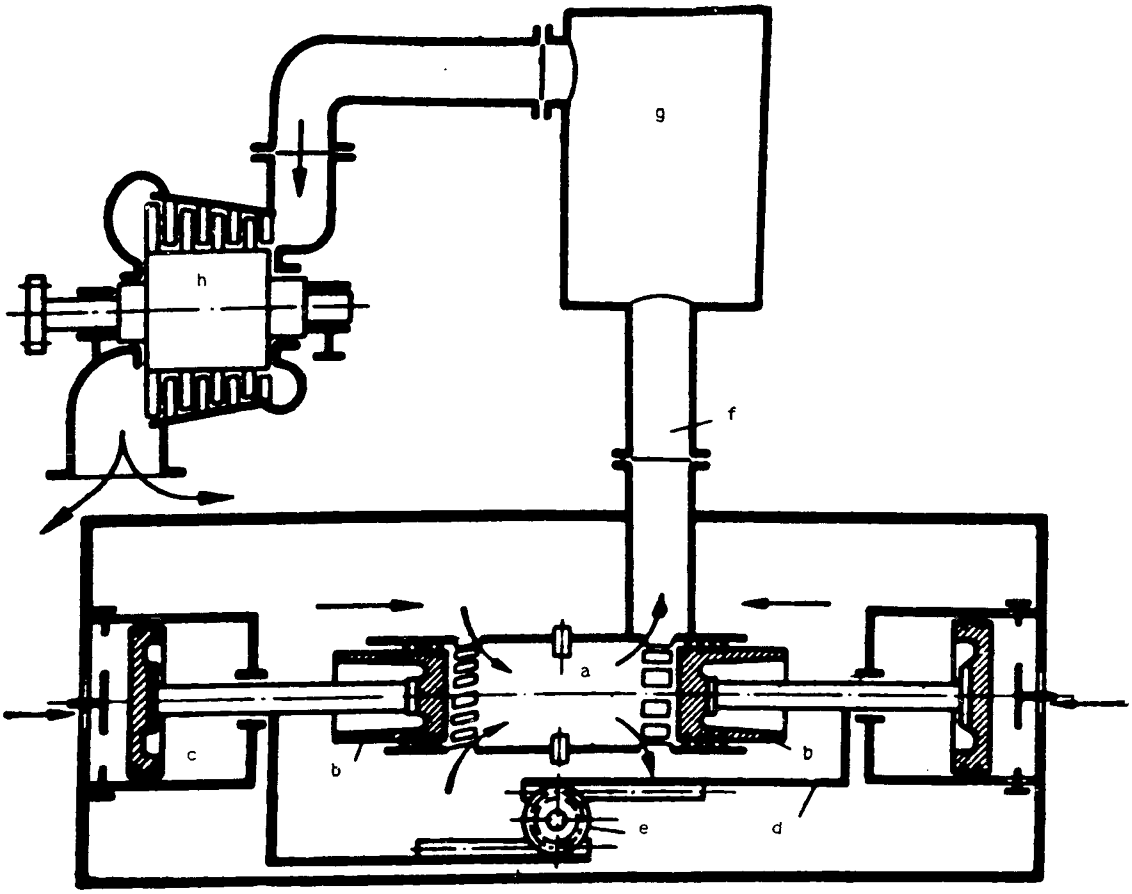
a. Compressor
b. Diesel engine
c. Exhaust-gas turbine
d. Gear

Figure 17. - Sulzer-type, eight-cylinder opposed-piston engine supercharged to 2.5 atmospheres absolute (reference 24).



a. Compressor cylinder
b. Diesel cylinder
c. Inlet port
d. Exhaust valve
e. Propellant-gas piping
f. Gas turbine

Figure 18. - Propellant-gas process with crank-type propellant-gas generator.



a. Diesel engine
b. Diesel piston
c. Compressor piston
d. Rack
e. Gear wheel
f. Propellant-gas piping
g. Receiver
h. Gas turbine

Figure 19. - Propellant-gas process with floating-piston-type, propellant-gas generator.