CHAPTER IX

UTILIZATION OF EXHAUST GAS ENERGY

1. The energy of exhaust gases

Modern vehicle engines have reached a high degree of efficiency except for the portion of the energy which leaves the engine, without being utilized, in exhaust gases. Approximately one third of the energy contained in fuel is converted into useful work, one third is removed by cooling, and the balance leaves the engine through exhaust gases.

Through increasing the compression ratio the thermal efficiency of the engine is improved and more fuel heat is utilized. Also by reducing thermal losses in the system the thermal efficiency of the engine is improved. This can be achieved by reducing the surface area of the combustion chamber by using a hemispheric shape and also by using a short stroke design. In air-cooled engines the amount of heat transferred by cooling represents only 0·8 to 0·5 (b.h.p.) which means a certain improvement in performance and less work done by the cooling system. Unfortunately the amount of heat drawn off by exhaust gases is not diminished, and is sometimes increased.

This encourages efforts to recover the energy from exhaust gases and convert it into useful work which can be accomplished in one of several ways.

2. Utilization of heat from exhaust gases

Exhaust gases from stationary engines can be used to heat water although the net gain of this arrangement is not substantial. In a power-station with seven diesel engines and a total output of 4700 kW exhaust gases were utilized to heat water in the boilers; from 1 b.h.p. of engine-output 1 lb/h (0·54 kg/h) of steam under a pressure of 163·5 lb/in^2 (11·5 kg/cm^2) was obtained. This steam was employed to drive two steam-engines whose output totalled 100 kW. Taking into account that the energy contained in exhaust gases roughly equals the performance of the engine, the utilized energy is slight. In latest-type equipment 3·3 lb (1·5 kg) steam/h can be obtained per b.h.p. of the engine performance where supercharged engines with elevated temperatures of exhaust gases are used.

With vehicle engines there are, however, other alternatives. The simplest and oldest method for utilizing heat was heating the vehicle. In buses or

diesel rail cars the exhaust piping is fitted under the seats inside the body and the whole of the car is directly heated. Care must of course be taken to prevent leaking of exhaust gases into the body. This method of using heat involves comparatively low losses.

3. Gas turbine (turbo-charger)

If the losses are to be kept at a low level, efforts must be made to utilize the energy directly, i.e. for the drive of a turbo-charger.

First attemps of this kind met with considerable difficulties with regard to the material to be used for the manufacture of the vanes, in particular with petrol engines with high exhaust temperatures. For this reason the development of turbo-chargers was discontinued and not taken up again until the use of diesel engines had become widespread. The temperature of exhaust gases from the latter engine is no more than 500 to 600 °C, as compared with 700 to 800 °C or more from petrol engines. The lower temperatures of exhaust gases didnot place such high requirements on the material for the turbine blades, and hence several satisfactory designs of exhaust gas driven turbines have become available. The output obtained can be employed for driving a supercharger to give an increase of some 50% in performance, without substantially increasing mechanical and thermal stresses.

A supercharger can now be mounted on any normal engine, as neither mechanical nor thermal stress of the engine are substantially increased. The increase in the performance is not obtained through an increase in peak pressures.

If a blower is employed for the supercharging of the engine, the timing of the valve gear and the exhaust layout of the engine must be modified. Emphasis must be placed on the least possible back-pressure and the greatest possible charging pressure. These requirements can be met through converging exhaust gases from two or three cylinders into one branch of the exhaust manifold. However, the exhausts of individual cylinders must under no circumstances overlap, as this feature would affect adversely the scavenging of the cylinders. Highest pressure, exceeding the charging pressure, is attained in the exhaust piping upon opening the exhaust valve. In the course of the exhaust stroke the pressure decreases, so that towards the conclusion of the exhaust stroke when the inlet valve of the cylinder is opened, the pressure in the exhaust piping is lower than the charging pressure. This provides for the scavenging of the cylinder. A portion of the charging air passes out through the exhaust port directly into the exhaust system. For the working cycle of the engine this air is lost, but it exerts a beneficial influence cooling the walls of the combustion chamber and the exhaust valve.

The scavenging air also lowers the temperature of the exhaust gases. It should be noted that the heat transferred by the scavenging air from the exhaust valve and the walls of the cylinder is not completely lost, as part of it will be converted into work in the turbine.

From the above it is evident that a four or six-cylinder engine must have at least two branches in the exhaust system, while eight- or twelve-cylinder engine must have three to four separate systems. If the thermal losses in the exhaust piping are to be cut down to the least possible amount, the blower must be mounted in such a manner as to reduce the length of the piping to the minimum. Besides, the piping must be thermally insulated. Fig. 133 shows a blower used for vehicle engines.

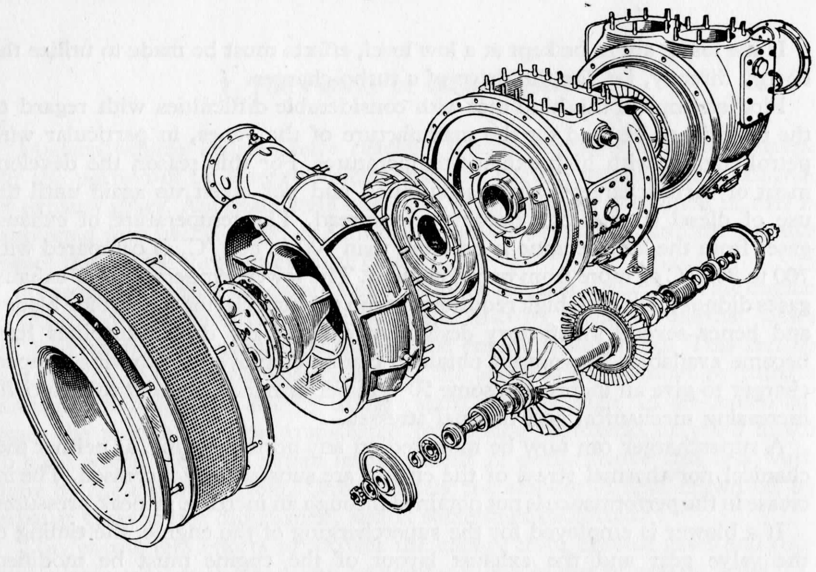

FIG. 133. Exhaust gas turbo-compressor used as a supercharger in compression-ignition engines.

Experiments have been made with much more powerful supercharging. In two-stroke engines with opposed pistons, supercharging by pressures of 2·3 or 6 atm has been tried. When the charging pressure of 6 atm was applied, the output of the engine equalled the input of the blower. Hence, the blower could be mechanically connected with the engine and the useful output was drawn only from the exhaust gas turbine. In the latter instance the reciprocating engine was actually replaced by a combustion turbine in whose design an engine-compressor was substituted for the combustion chamber. The situation is similar in the case of an engine with free pistons combined with an exhaust gas turbine. In that instance also the engine with free pistons is confined to the role of a generator of exhaust gases and the entire output required for the drive of the vehicle is generated by the turbine. By such an arrangement better thermal efficiency is attained than by a turbine provided with a combustion chamber.

For road vehicles the Eberspächer turbo-compressor using a centripetal turbine offers great advantages and permits the design of a small-dimensioned turbo-compressor. The overall layout of the latter is apparent from Fig. 134. This turbo-compressor is particularly suitable for air-cooled engines. Part of the air from the compressor is brought by a special duct to the bearings for cooling purposes and simultaneously prevents the exhaust gases from penetrating into the bearing area. The weight of the turbo-compressor will reach up to 35 lb (16 kg) for engines of up to 150—200 b.h.p. and 17·5 lb (8 kg) for engines of up to 100 b.h.p. Turbo-compressors of this type give a performance increase of some 30 %.

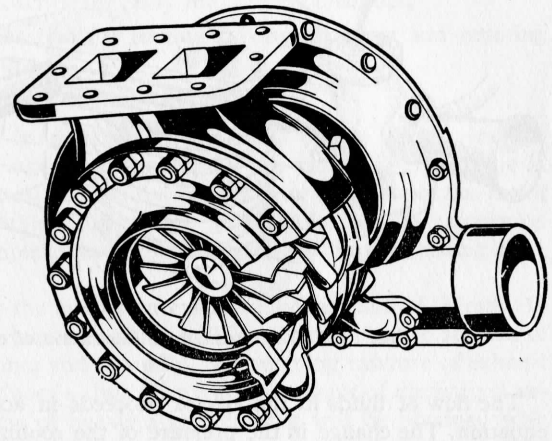

FIG. 134. The Eberspächer turbo-compressor.

Application of the exhaust gas turbine is not confined to compressor driving. It may also be used for driving a fan supplying cooling air. In the latter instance not only is there a gain in work used for the drive of the fan, but the cooling is automatically controlled. When the engine is fully loaded the exhaust gas turbine output is at its highest, so is the speed, and the maximum of cooling air is supplied. With the same speed of the engine and a diminished load the speed of the turbine will be reduced, and the cooling will be proportionate to the lower performance of the engine. The obvious advantages of this system have not been fully exploited because of comparatively expensive equipment.

4. Exhaust gas ejectors

The energy of the exhaust gases can be used directly, without the intermediary of any rotational device, for the suction of cooling air. This can be achieved by the use of an ejector which is a simpler device than an exhaust turbine connected with a fan. There is no lag in operation and it is absolutely reliable.

The ejector uses the kinetic energy of one fluid to draw in another. The layout of an ejector is apparent from Fig. 135. Exhaust gases are brought from the engine by pipes to the mouth of the ejector where the cooling air is entrained and a mixture of this air and exhaust gases leaves the end of the ejector to the atmosphere. The ejector consists of a branch of the exhaust piping (nozzle), a cylindrical mixing chamber, to which a conical extension called the diffuser is sometimes added. The whole equipment is therefore extremely simple, and as it contains no moving parts it does not require servicing.

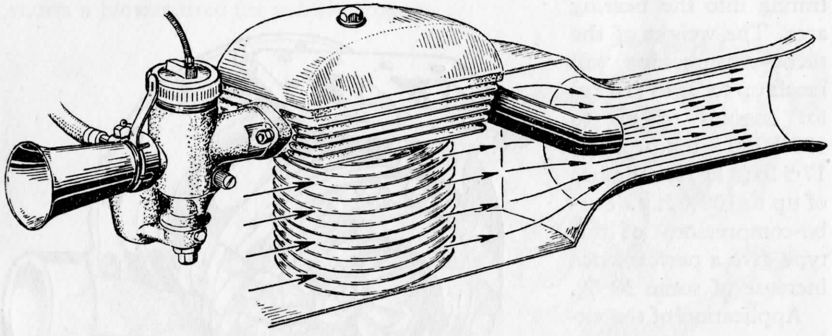

FIG. 135. Diagrammatical illustration of ejector cooling.

The flow of fluids in the ejector proceeds in accordance with Bernoulli's equation. The change in the pressure of the cooling air upon entry into the ejector amounts to:

$$p_0 - p_1 = \frac{1}{2}\gamma_1 \cdot v_1^2;$$

Assuming that the momentums of inlet and exit air are constant, the change in the pressure obtaining in the mixing chamber can be established. This, however, only holds true if the friction of gas against the walls of the mixing chamber is disregarded. The increment in air-pressure will then equal:

$$p_2 - p_1 = \frac{1}{A}\left[\frac{W_e \cdot v_e}{g} + \frac{W_a \cdot v_1}{g} - \frac{W \cdot v_2}{g}\right]$$

The change of pressure in the diffuser can be expressed by the equation:

$$p_3 - p_2 = \frac{1}{2}\gamma_2 \cdot v_2^2 - \frac{1}{2}\gamma_3 \cdot v_3^2$$

The actual conditions prevailing in the ejector are far more involved and cannot be expressed by simple equations. A rough outline of the rating is given in the references [39]. The symbols employed in the foregoing equations signify:

p denotes the pressure of gas,
v „ „ velocity of gas,

v_e denotes the velocity of exhaust gases leaving nozzle,
γ „ „ specific gravity of exhaust gases,
g „ „ gravitational acceleration,
W_e „ „ amount by weight of exhaust gases,
W_a „ „ amount by weight of cooling air,
A „ „ cross-section area of the mixing chamber,
1 „ „ subscript for entry into mixing chamber,
2 „ „ subscript for leaving mixing chamber and entering diffuser,
3 „ „ subscript for leaving diffuser.

Losses arising through friction against the walls of the ejector are not negligible and have to be taken into account for the rating. Every change in the flow entails certain losses, and the flow of gas from exhaust nozzles is not even, but pulsating. All these factors are responsible for the calculation becoming complicated. Empirical coefficients established through testing have, therefore, to be applied.

The action exercised by the ejector on exhaust gases is twofold. Firstly, in the chamber before the ejector vacuum will be formed for the suction of cooling air across the engine, and secondly, the outgoing mixture of exhaust gases and cooling air produces a drag. For a given amount of discharged exhaust gases and a given size of nozzle a greater drag will be obtained with ejectors having a small cross-section area and high velocity of discharge. The dimensions of the ejector are small, but the ejector is more sensitive to maintaining the supposed values.

If the ejector is employed for drawing the cooling air, it is not the drag of the ejector that is of importance, but the amount and the pressure of drawn air. Generally, the ejector can draw either small amounts of air with great pressure heads, or large amounts of air with small pressure heads.

For cooling air-cooled engines ten times the amount of air is required compared to the amount required for the air-fuel mixture. A widely spaced finning requires the supply of large quantities of low-pressure air, whereas a design providing for a dense finning will call for smaller quantities of high-pressure air. Ordinarily, a pressure head 3″ to 6″ (of 75 to 150 mm) water column is required. The cross-section of the mixing chamber is usually eight to ten times greater than the cross-section of the exhaust nozzle.

When laying down the size of the exhaust nozzle it must be taken into account that if the cross-section of the nozzle is less than the flow area in the exhaust valve at full opening, the back pressure in the exhaust piping reduces the performance of the engine. Therefore, the size of the exhaust nozzle should be somewhat greater than the flow area in a fully opened exhaust valve.

If a separate pipe is fitted from each cylinder to the mouth of the ejector, the pulsating effect of the exhaust will have been made use of, but only for the time of the exhaust of the corresponding cylinder. For the rest of the time the exhaust pipe unnecessarily blocks the useful cross section of the ejector. It is, therefore, preferable to conduct the exhaust from two or three cylinders, with regularly spaced ignition, into one exhaust pipe. A more even flow from the nozzle and a better utilization of the cross section of the ejector will thus be attained. However, the exhaust periods of individual cylinders must not overlap, so that the back pressures in the exhaust system do not obstruct the scavenging of the cylinders.

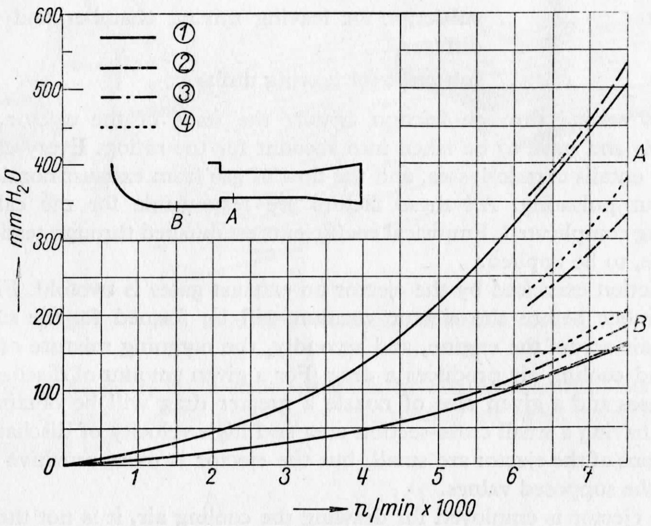

FIG. 136. Measured under pressures in different ejectors.

In a given ejector, joint exhaust piping from three cylinders will yield a greater vacuum than three independent branches each from one of the three cylinders. The branches of exhaust nozzles should open approximately on the level of the ejector orifice. Experiments with even flow nozzles do not reveal an impairing of the performance when the nozzles reach into the ejector branch. However, the useful length of the mixing chamber is diminished. It is of importance to ensure that the nozzles are distributed evenly over the area of the mixing chamber and that the gases are discharged in a direction parallel to the axis of the ejector. If exhaust gases come up against the wall of the mixing chamber, losses are incurred and the efficiency of the ejector is lowered. The diameter of the bevel to the inlet edge of the ejector can reach one sixth of the diameter of the ejector. A greater diameter than this unnecessarily extends the length of the ejector.

The length of the mixing chamber is related to its diameter and to the number of nozzles. At a given diameter of the ejector, the greater its length the

more even is the distribution of the velocity of the gases in the outlet area, and the performance of the ejector improves. But excessively long ejectors imply great losses through friction of the gases against the walls and a drop in the overall performance. The optimum length of the mixing chamber, as established by tests is, for single-nozzle ejectors, four to eight times the diameter. Considering that exhaust gases leave the nozzle at a certain angle,

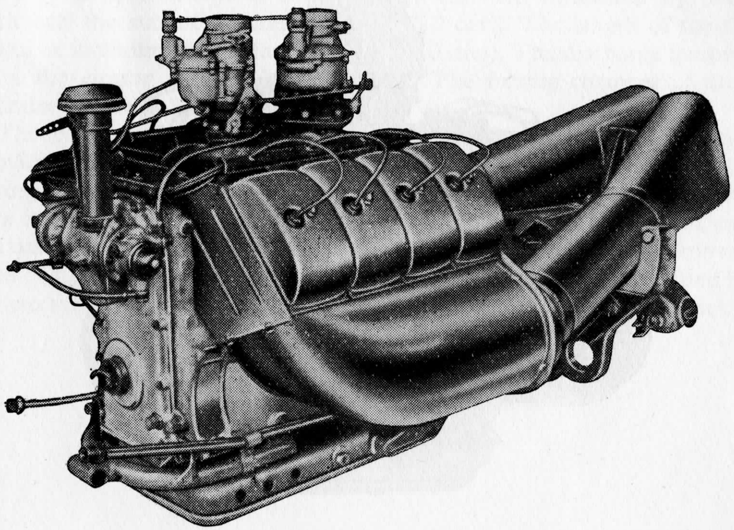

FIG. 137. The T 603 engine with ejector cooling.

the length of the ejector is dependent on the distance of the edge of the nozzle from the wall of the mixing chamber. Therefore, if several nozzles evenly distributed over the cross-section are used, the optimum length of the ejector will be smaller than in the case of a single nozzle. The length of the ejector can be reduced by forming the mouth of the nozzle in the shape of a star, etc. Similar arrangements are responsible for increasing the contact surface between exhaust gases and air, and the efficiency of the ejector is thus enhanced.

The efficiency of the ejector will not be raised through fitting a greater number of nozzles, such as one nozzle from each cylinder, if the discharge is not regular. For low pressure heads a short mixing chamber will be sufficient. If an expanding diffuser is linked to the mixing chamber, the length of the mixing chamber can be reduced. Tests have proved that a cylindrical shape for the mixing chamber with a constant cross section is entirely satisfactory and that there is no need to apply a Venturi tube.

The efficiency of the ejector is improved by the use of a conical diffuser fitted behind the mixing chamber. An apex angle of 8° has proved the most con venient. In other than circular diffusers the increment in the surface of the cross section of the diffuser should be such as to correspond to a circular cross section with an angle of 8°. A limited length of the ejector combined with a greater angle will produce a greater flow at the expense of the drag. If the

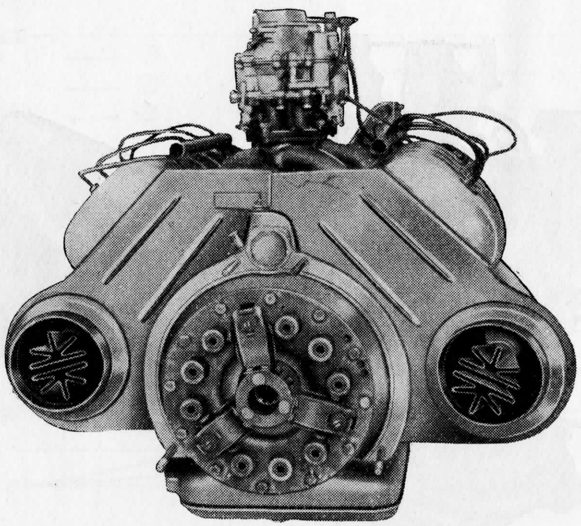

FIG. 138. Star-shaped tail ends of the exhaust pipe.

area at the delivery end of the diffuser is only slightly greater than the area at the intake, the optimum angle of widening will exceed 8°.

If the energy of the exhaust gasses is to be exploited thoroughly, the exhaust piping must be short and not exposed to excessive cooling. This gives difficulties of silencing. The Tatra works experimented with ejector-type cooling for racing cars, where noise is unimportant, while at high speeds the economy in input to be used for the fan is substantial and the dependable operation of the ejector is most welcome.

After the first promising tests with the T 603 engine, where the ejector worked in series with the axial fan, thorough tests have been arranged on a two-cylinder engine with cylinders of 75 mm bore and a stroke of 72 mm. The results of such tests are illustrated in Fig. 136. Different diameters of mixing chambers were experimented with and by means of a telescopic arrangement their length could be modified while the mechanism was in operation. Also different shapes of end-sections have been tried. Based on the results of experiments, ejectors have been constructed both for the two-cylinder T 605

car and for the eight-cylinder T 607 car. These ejectors have proved outstanding in operation and will cool the engine safely even at maximum performance.

The final arrangement of the T 603 ejector-cooled engine is shown in Fig. 137. The engine has a 75 mm bore and 71 mm stroke. At 7500 rev/min and a compression ratio of 12 : 1 the performance of the engine is 200 b.h.p. On each side of the engine one ejector is mounted, with a mixing chamber 140 mm dia. The end-section of the exhaust is double star-shaped as illustrated in Fig. 138. Groups of two cylinders are each connected by one pipe with half the star of an area of 1·5 in^2 (10 cm^2). The length of the mixing chamber including the diffuser is 33·4″ (850 mm). The discharge temperature from the ejector is 160 °C at full load. The mixing chamber of the two-cylinder engine is 3·5" (90 mm) dia.

The ejector type of cooling might also be used for stationary engines, providing there is sufficient room for fitting silencers behind the ejector. The saving in fan energy will give lower fuel consumption. The cooling control is to a large extent automatic, for at a full load the greater amount of exhaust gas draws in an increased quantity of air. It is possible that with a convenient type of finning a silencer, possibly of the Burgess type, could be fitted before the ejector, thus making this type of cooling acceptable for road vehicles.