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 [21] Appl. No. **794,187**
 [22] Filed **Jan. 27, 1969**
 [45] Patented **July 27, 1971**
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 [32] Priority **Feb. 7, 1968, Aug. 7, 1968, Aug. 8, 1968,**
Dec. 5, 1968
 [33] **France**
 [31] **139,048, 162,147, 162,320 and 176,778**

[56]

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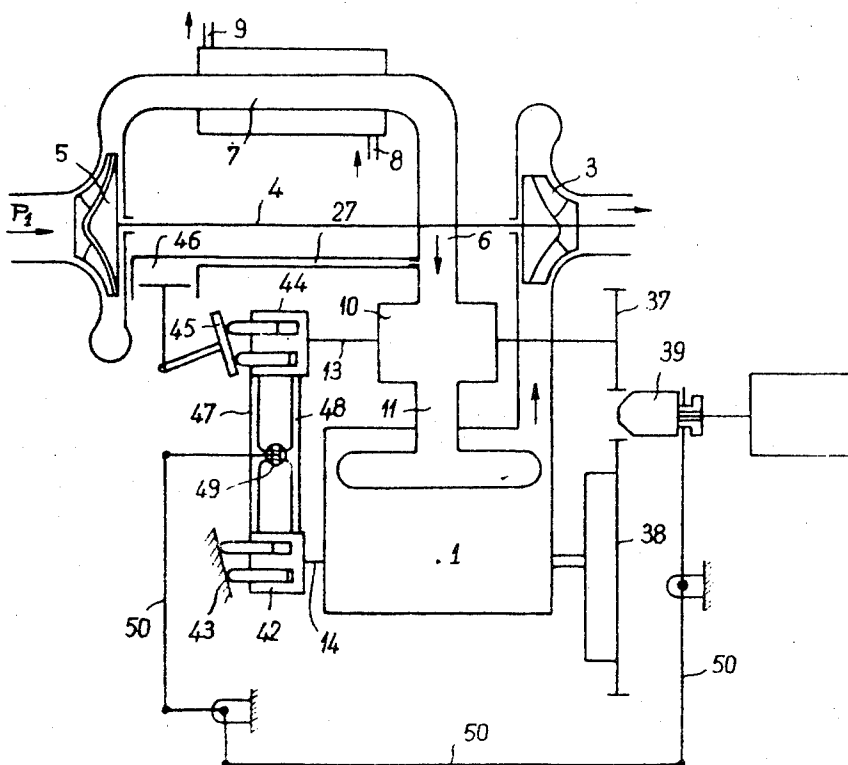
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[54] **COMPENSATED SUPERCHARGING DEVICES FOR**
COMPRESSION-IGNITION ENGINES
8 Claims, 12 Drawing Figs.

[52] U.S. Cl. **60/13,**
123/119
 [51] Int. Cl. **F02b 37/04**
 [50] Field of Search **60/13;**
123/119 C

ABSTRACT: A compensated supercharging device for a compression-ignition engine cooled by a liquid at a regulated temperature and already supercharged by a turbosupercharger, said device tending to correct the known inconveniences of this type of supercharging by regulating the temperature and pressure of the air delivered by said turbosupercharger before it is introduced into the engine for all speed and load conditions of said engine.



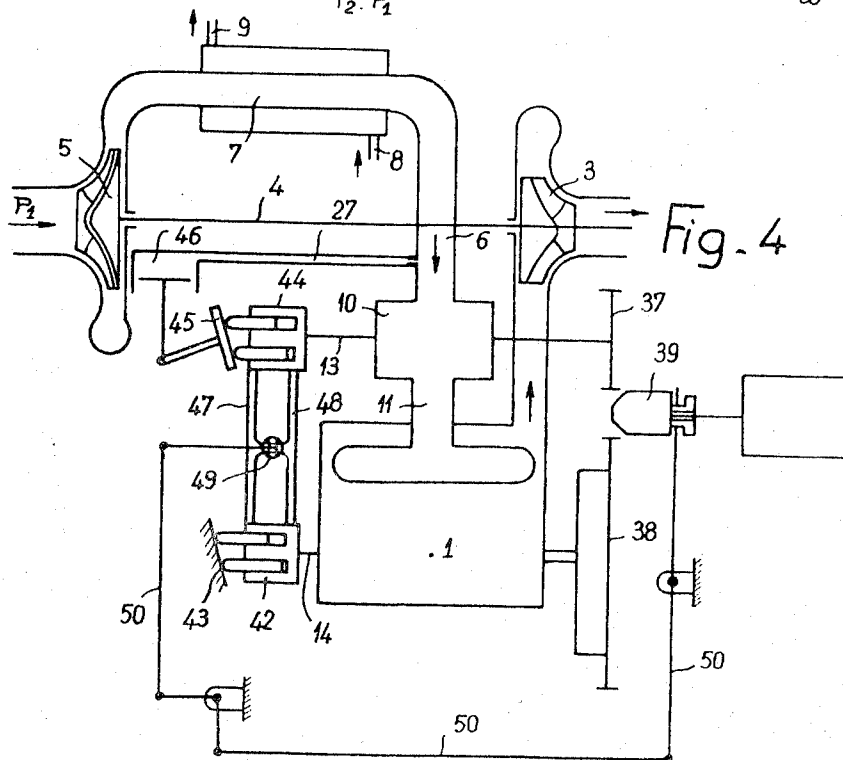
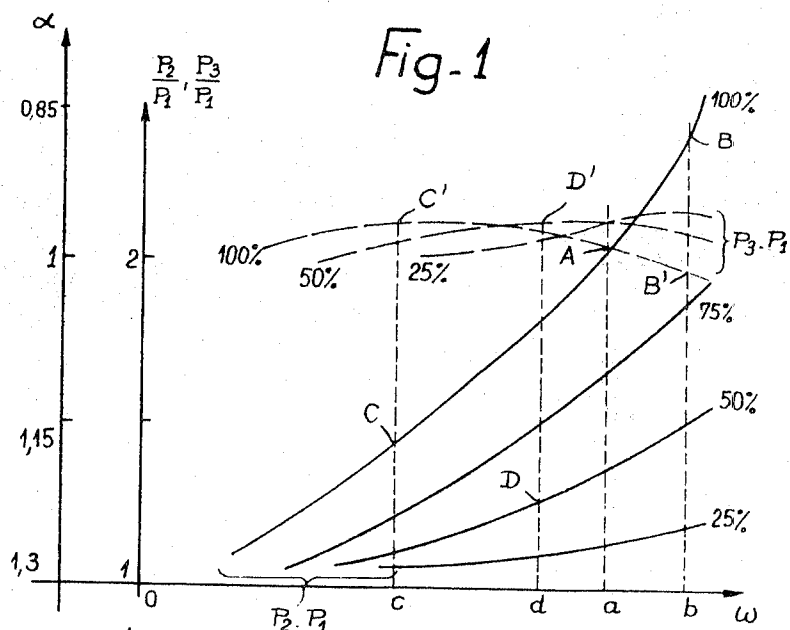


Fig-2

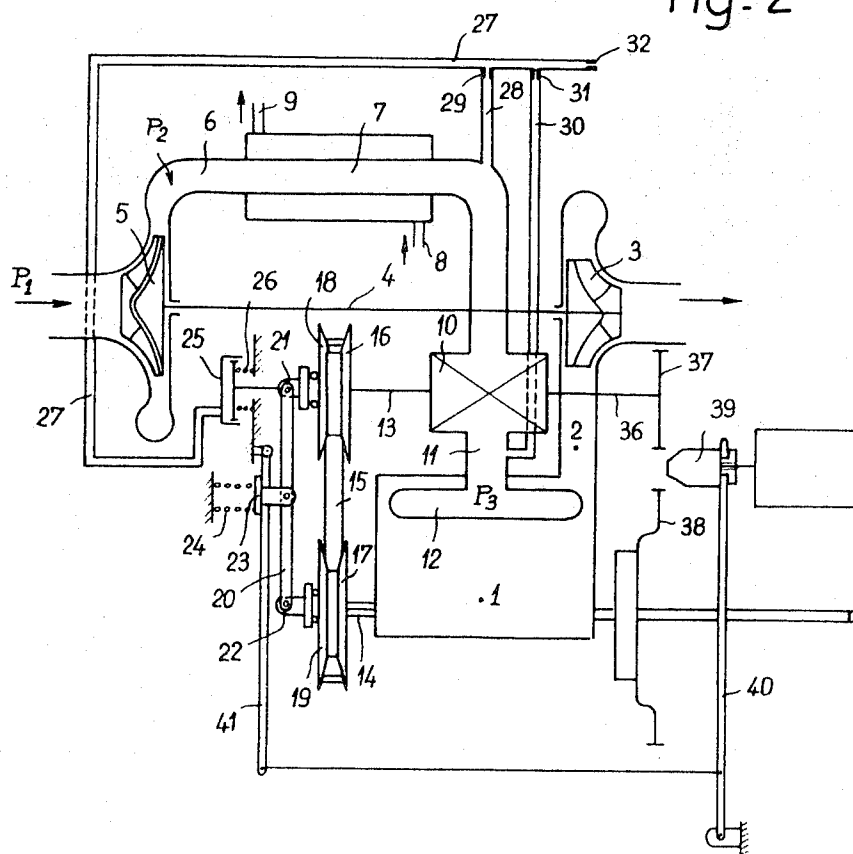
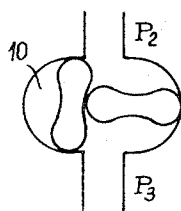
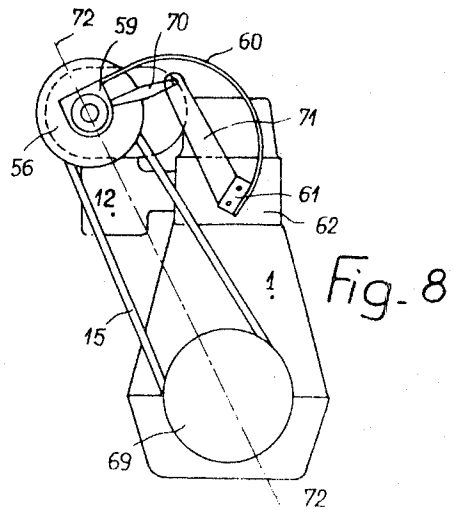
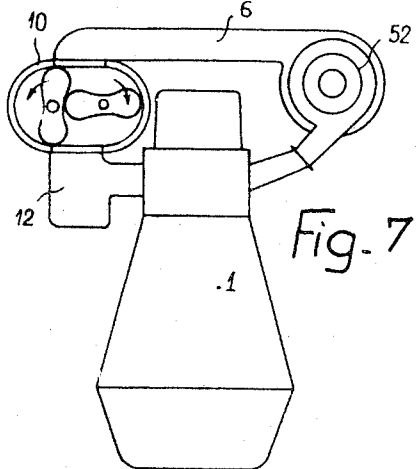
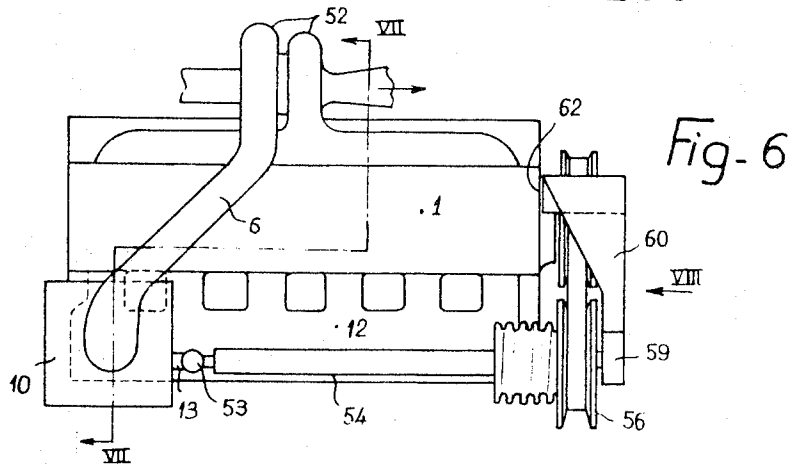
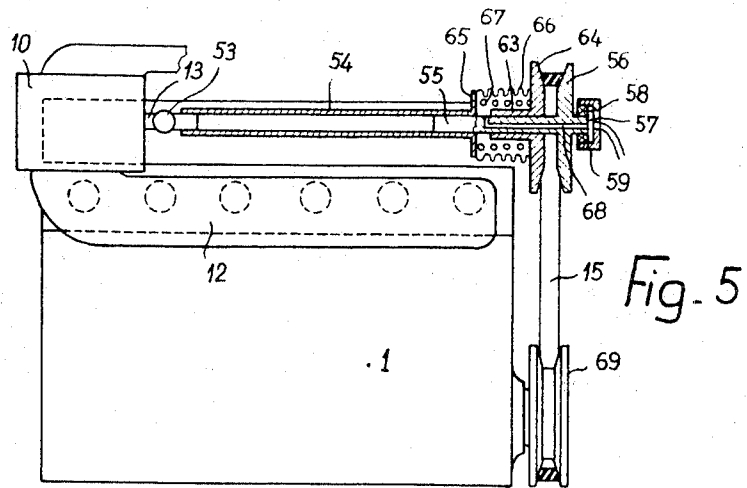


Fig-3





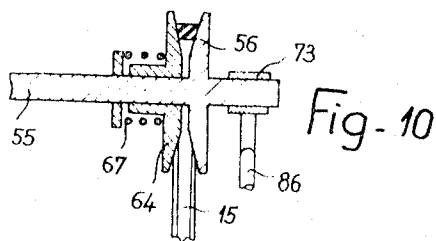
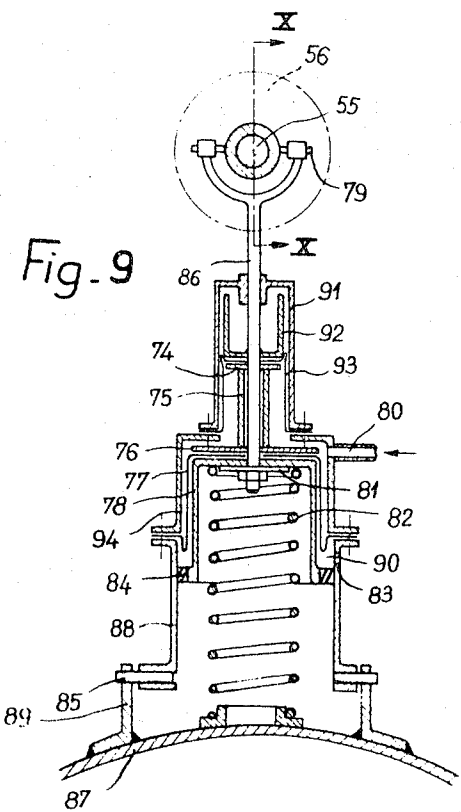


Fig. 11

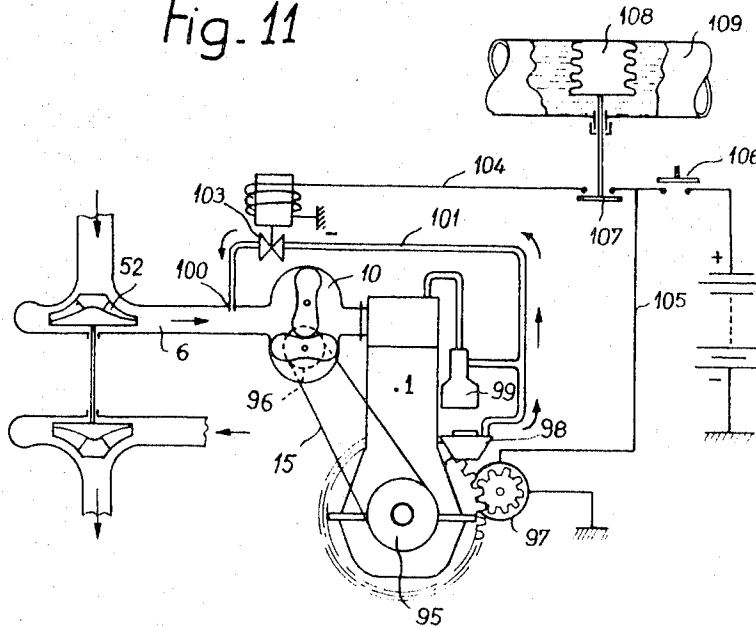
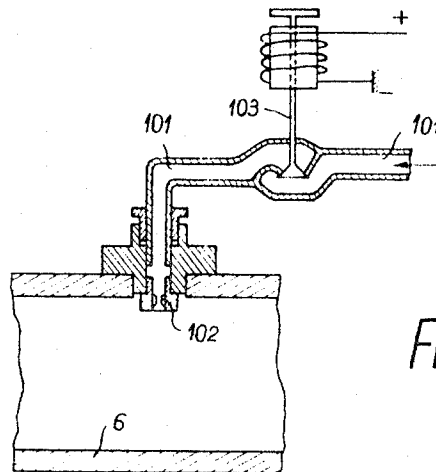


Fig- 12



COMPENSATED SUPERCHARGING DEVICES FOR COMPRESSION-IGNITION ENGINES

This invention relates in general to means for supercharging compression-ignition engines and has specific reference to devices for producing the compensated supercharging of engines of this character.

It is the chief object of this invention to provide a device capable, in a compression-ignition engine already supercharged by a turbosupercharger, or producing substantially unchanged temperature and pressure conditions for the air introduced into the cylinders, regardless of the conditions of operation of the engine. The ultimate object is to obtain controlled combustions in all cases, that is, a high-efficiency and low-consumption engine, without releasing smoke at low speeds nor emitting aldehydes at low-loads and when starting the engine, even in the case of multifuel engines, while preserving a torque curve ensuring a reliable driving power and without endeavoring to obtain a constant power output.

It is now current practice to supercharge Diesel or compression-ignition engines by using a turbosupercharger driven from the exhaust gases.

This technique is advantageous not only on account of the reduced overall dimensions and low weight of the means involved, but also in that it permits improving the engine power output by utilizing a substantial fraction of the energy usually lost with these gases, in spite of the counterpressure created by the turbine. A cooling system arranged between the supercharger and the engine further improves the engine cylinder filling.

In many cases the supercharging is completed by a more or less strong scavenging action by causing the opening times of the inlet and exhaust valves of a same cylinder to more or less overlap each other. Thus, the temperature of the pistons is reduced, so that they can withstand the considerable supercharging stress.

However, these various arrangements cannot be exploited as much as desired for the following reasons:

1. The turbosupercharger efficiency is higher at high engine speeds, so that the torque curve is deflected in a direction opposite to the usually desired direction;
2. At low load values the counterpressure may exceed the feed pressure, thus creating a counterscavenging likely to prove detrimental to the useful life of the inlet valve;
3. It is rather difficult to select a suitable volumetric ration combining a maximum pressure limitation and an easy starting.

It is the object of this invention to provide a device capable of preserving full load fuel conditions in all cases arising during the operation of a Diesel or compression-ignition engine already supercharged by a turbosupercharger, and more particularly of providing constant combustion conditions by regulating the induction air temperature and pressure, notably by expanding this air when the engine revolves at full-power and reinforcing the air supercharging when the engine operates at low speed or under low-load conditions.

The device according to this invention, which can be used in a compression-ignition engine equipped with cooling system utilizing a liquid as a heat transfer medium, and with a turbosupercharger driven from the exhaust gas and connected to an induction manifold or pipe of the engine, is characterized by the following features:

A temperature and pressure corrector is interposed in the scavenging duct and comprises a heat exchanger through which the cooling liquid of the engine is caused to flow, and a volumetric pump driven from the engine shaft through a variable-speed transmission adapted to be uncoupled. This transmission is so controlled through means responsive to force depending on the air pressure prevailing in the scavenging duct, that the corrector tends to increase the quantity of air fed downstream of the pump when this pressure is low and on the contrary to decrease this quantity of air when said pressure is high. An uncoupling and coupling control device may be as-

sociated with the transmission and with the starter motor so that the operation of the starter motor will disconnect the transmission and couple the pump to the driving pinion of the starting motor.

- 5 The corrector will thus act as a pressure-reducing device at high power outputs, and as a supercharger at low loads.

Thus, a turbosupercharger of the high pressure ratio type can be used for at high engine power output values the assembly comprising the supercharger, the heat exchanger and the pressure-reducing device acts as a real refrigerator which, in combination with a reasonable volumetric ratio in the engine, permits maintaining the pressure and temperature values, at the end of the compression stroke, within predetermined limits.

At low loads when the turbosupercharger gives neither sufficient pressures nor sufficient temperatures, the heat exchanger through which the engine cooling liquid adjusted at a constant temperature is caused to flow will then act as a reheater and the volumetric pump will further increase the temperature as it increases the air pressure, whereby the combustion conditions approximate those obtained under full load. This pressure increment further permits overcoming the exhaust gas pressure and avoiding any counter scavenging action. Of course, the conditions of operation of the volumetric pump must be adapted to vary gradually from one limit to another by passing through a neutral point at which it will neither compress nor expand the air, this neutral point corresponding, for example, to the conditions which would have been selected for supercharging the engine by means of a turbosupercharger if no corrector were provided.

The control means responsive to a force depending on the air pressure prevailing in the scavenging duct may advantageously be actuated by a pressure-responsive pick up element mounted in said duct upstream of the volumetric pump, in combination or not with a similar pick up element mounted downstream of said pump. The pick up element or elements may consist of a pressure-responsive capsule of the diaphragm or piston type branched off the manifold. The force of the movable member of the capsule is then utilized for varying the transmission ratio.

The correcting volumetric pump consists preferably of a gear-type supercharger (without excluding blade- or vane- or piston-machines) driven preferably through the medium of a variable-speed transmission of the type comprising variable-diameter pulleys and belts. A hydrostatic transmission may also be used with a variable-output pump and a fixed-output hydraulic motor. The main point to be adhered to is that this transmission, for reasons to be set forth presently, can be uncoupled automatically, for example in a first case by the movement of the flanges of two pulleys away from each other and in another case by operating a bypass valve in the hydraulic circuit. The extreme speed ratio will be at the most 1:1.7, so that the variable-speed transmission can be designed under the best possible conditions from the dual point of view of efficiency and simplicity.

Another important feature characterizing this invention lies in the particular use made of the corrector during the engine starting period. It was observed that at low engine speeds and loads, notably when idling is indulged in, the device according to this invention acted as a reheater. One might reasonably think that the same applies to the engine starting period, but in this case the centrifugal supercharger produces no pressure, the liquid in the heat exchanger is cold and the volumetric pump, notwithstanding the maximum step-up ratio provided by the transmission, does not rotate fast enough, in spite of its leakages, for producing a sufficient pressure and temperature increment; therefore, it is proposed to increase very strongly the rotational speed of the pump during this period by uncoupling same from the transmission and driving this pump directly from the starter motor, in conjunction with the engine and in the same fashion, by means of the starter motor pinion driving a toothed wheel mounted on the pump shaft and similar to the conventional toothed ring mounted on the

flywheel, but of a diameter so calculated that the momentary transmission ratio thus obtained between the engine and the pump is several times greater (e.g. 3, 4 or 5 times) than the maximum step up ratio provided by the variable-speed device. Thus, the pump speed will be sufficient to create a certain overpressure and more particularly an appreciable temperature increment, as the leakages are recycled. The volumetric pump can be uncoupled from the variable speed transmission by mounting the former of a freewheel device, but this arrangement would preclude its use as a pressure reducing device at high power outputs; preferably, a clutch associated with the variable-speed device will be used, this clutch being driven from the starter motor and constructed for example in the manner set forth hereinabove.

The power requirement of the starter motor is not necessarily increased since the resulting overpressure and the reheating of the piston crowns will reduce the necessary engine torque.

This invention is further concerned with means capable of improving the efficiency of the corrector during the engine starting periods as an alternative and a substitute for the corrector coupling and step up system described hereinabove, by eliminating or reducing the air leakages mentioned in the foregoing, which leakages are normally observed between the movable members and between these members and the case of the volumetric pump. To this end, an injection device fed with a surface-active liquid capable of adhering to said movable members and under a pressure greater than that prevailing in the scavenging duct opens into this duct upstream of the pump, this injection device comprising on the one hand a passage for said liquid which has a cross-sectional area calculated to cause the liquid output therethrough to momentarily stop any air leaks, and on the other hand, a stop of valve member permitting the flow of said liquid towards said duct only during the engine starting periods.

Thus, any air leaks from the pump are eliminated during the starting period by injecting upstream of the pump, with a moderate output, a liquid capable of adhering by surface-tension to the rotors and thus momentarily stop or cut off any leakages.

The above-mentioned surface-active liquid may consist of a suitable oil, but it is preferable to use the engine fuel proper. In fact, this liquid will finally penetrate into the engine and it is preferable to burn it without leaving any harmful deposit therein. The scavenging of this liquid efficiently atomized by the pump action will create a kind of vapor which, associated with the normal pump injection, promotes the fuel combustion.

Of course, this fuel injection should not continue after the engine has actually been started, and therefore it must be stopped immediately as the engine fires. Preferably, a suitable electromagnet-valve may be used for controlling this injection, this valve being energized to its open position in conjunction with the starter motor.

Finally, when the engine has warmed up the device becomes useless; it may even prove detrimental and cause in this case untimely preignitions.

It is therefore advantageous that the energizing circuit of the electromagnet valve be controlled by a thermostat responsive to the engine cooling water temperature, the circuit being open when the water is hot.

These multiple arrangements tending to keep the pressures and temperatures as constant as possible at the end of the engine compression stroke in all operating conditions are favorable to the engine operation as a multifuel engine.

It is important to note that what characterizes this invention in comparison to known combinations of centrifugal supercharger with volumetric supercharger as utilized for instance in certain large-sized two-stroke engines is that the corrector is not synchronized with the engine, that its controlled drive causes this corrector to operate as a pressure-reducing device and/or as a supercharger, that its position downstream of the heat exchanger disposed in turn downstream of the cen-

trifugal supercharger causes this device to operate now as a cooling device or as a reheater, and that when it acts as a pressure reducing device it returns a certain amount of energy to the engine.

This invention is also concerned with various additional constructional details which will appear as the following description proceeds with reference to the accompanying drawings, in which:

FIG. 1 illustrates diagrammatically, as a function of the engine speed, the curves corresponding to the pressure ratios of the turbosupercharger under various loads corrected and uncorrected by the corrector of this invention;

FIG. 2 is a diagram showing the assembly with a variable speed pump drive of the belt-and-pulley type;

FIG. 3 is a cross-sectional view of the pump;

FIG. 4 illustrates the same general diagrammatic view as FIG. 2, the variable speed drive of the pump being of the hydrostatic transmission type;

FIG. 5 shows in elevational view a compression-ignition engine equipped with a modified type of variable-speed belt transmission having only one variable-diameter pulley;

FIG. 6 illustrates in the same engine as shown in FIG. 5 in plane view;

FIG. 7 is a section taken along the line VII-VII of FIG. 6;

FIG. 8 is a side elevational view of the engine as seen in the direction of the arrow VIII of FIG. 6;

FIG. 9 illustrates a modified form of embodiment of the variable drive illustrated in FIG. 2;

FIG. 10 is a detail view as seen along the section line X-X of FIG. 9;

FIG. 11 illustrates diagrammatically a compression-ignition engine supercharged and equipped with a starting injection device, and

FIG. 12 is a view showing on a larger scale the starting injection device proper.

Referring first to FIG. 2, it will be seen that the compression-ignition engine 1, preferably of the four-stroke cycle type, comprises an exhaust pipe 2 directing the exhaust gas towards the rotor 3 of a radial turbine driving via a shaft 4 the rotor 5 of a centrifugal supercharger. The scavenging duct 6 of this supercharger has inserted therein the corrector of this invention which comprises a heat exchanger 7 and a volumetric pump 10. Thus, the compressed air will be directed by the scavenging duct 6 firstly through the heat exchanger 7 in which the engine cooling fluid regulated at a constant temperature is circulated from an inlet 8 to an outlet 9, the through the volumetric pump 10 shown in cross-sectional view in FIG. 3. This pump is a conventional volumetric gear pump delivering combustion air to the engine 1 via the inlet pipe 11 and manifold 12. The pump shaft 13 is coupled to the engine crankshaft 14 by means of a variable-speed transmission of any known type, such as fluid, electrical or friction transmissions. The transmission illustrated is a pulley-belt arrangement comprising a V-belt 15 according to the known arrangement; the flanges 16 and 17 of the pulleys receiving this belt are keyed to shafts 13 and 14 respectively, and the pulley flanges 18 and 19 registering with the flanges 16 and 17 respectively and driven at the same speeds are slidably movable to a moderate extent in the axial direction whereby the diameter of the portion engaged by the belt 15 on each pulley can be varied. In the specific case contemplated, which requires but a moderate variation in the extreme speed ratios (e.g. from 0.85 to 1.5) the two flanges 18 and 19 can be moved simultaneously in opposite directions by a rocker 20 engaging with one end 21 and flange 18 and with the opposite end 22 the flange 19, for example by means of thrust bearings. This rocker 20 is fulcrumed intermediate its ends to a strap 23 urged by a spring 24; it is actuated at 21 by a control diaphragm 25 counteracted by a compression spring 26. This control diaphragm 25 is supplied with air under a variable pressure through a line 27 connected through a gauged air orifice 29 to a branch line 28 leading from duct 6 at the outlet of the heat exchanger and through another gauged orifice 31 to another branch line 30

leading from pipe 11 at the outlet of the volumetric pump 10, a third air orifice 32 being inserted in said line 27 and communicating with the external atmosphere.

The extension 36 of shaft 13 of the volumetric pump 10 carries a toothed wheel 37 having the same teeth as the toothed ring 38 of the engine flywheel, the pinion 39 of the starter motor being adapted to mesh simultaneously with both gears, provided that their teeth are designed with a suitable contour. The dimensions of these two gears are such that the speed step up ratio between the engine 1 and pump 10 is greater than the highest gear ratio provided by the variable-speed transmission incorporating the belt 15. The starter motor is equipped with means for uncoupling the variable-speed belt transmission; in this example, said means comprise a linkage shown in diagrammatic form as comprising levers 40 and 41 adapted to compress the above-mentioned spring 24 and thus simultaneously eliminate the belt-tensioning effort exerted by both flanges 18 and 19. Another means may consist in compressing said spring 24 by using an electromagnet energized in conjunction with the starter motor.

A hydrostatic transmission of the type illustrated in FIG. 4 may be substituted for the belt-type variable-speed transmission. In this case, the engine crankshaft 14 drives a barrel piston pump 42 having its output per revolution regulated by a fixed swashplate 43. The shaft 13 of pump 10 is then coupled to a hydraulic motor 44 also of the barrel piston type, of which the output per revolution is made variable by the variable inclination of the relevant swashplate 45. This variable inclination of swashplate 45 is controlled by a pressure-responsive control diaphragm 46 communicating with the scavenging duct 6 via line 27. These two hydraulic units are interconnected by passages 47 and 48 adapted to be interconnected in turn by a bypass line provided with a control valve 49 operable by means of a linkage (shown in diagrammatic form at 50) from the starter motor pinion 39. If desired, the movable swashplate 45 may also be mounted without any inconvenience on the hydraulic unit 42, only the control means thereof being modified accordingly.

The operation of the above-described arrangement will now be explained with reference to FIGS. 1 and 2, it being assumed that only air orifice 29 is operative.

Let us assume that P_1 is the air pressure at the inlet of the centrifugal supercharger, P_2 the pressure between this centrifugal supercharger and the volumetric pump, P_3 the pressure between this volumetric pump and the engine. In FIG. 1 the engine speed is plotted in abscissa against the pressure ratios P_2/P_1 and P_3/P_1 plotted in ordinates; the thick line curves designate the variation in the ratio P_2/P_1 at different load values, and the dash line curves show the variation in the ratio P_3/P_1 also at different load values.

The values α of the pump/engine speed ratio are also shown in ordinates.

With a maximum of 100 percent load, at an engine speed a representing substantially the three-fourths or the four-fifths of the maximum engine r.p.m. value, the pump/engine speed ratio is so selected that the volume generated by the pump corresponds to the engine cylinder filling capacity; in other words and to simplify the disclosure, let us assume that this ratio is 1.00. At this point the unit operates as if no corrector were provided; the pressures P_3 and P_2 are identical and the thick and dashline curves intersect each other at a point A corresponding to said engine speed a .

The ratio P_3/P_1 equals P_2/P_1 is relatively high; assuming that the temperature A at the outlet of the supercharger as in excess of 100° C., it is reduced to about 80° C. by the heat exchanger 7 and preserves this value in the manifold 11 as it penetrates into the engine.

If the engine speed increases, the ration P_2/P_1 increases with the pressure P_2 . Assuming that the air jet 31 were not perforated, the increment in pressure P_2 would be transmitted via duct 28, jet 29 and line 27 to the diaphragm 25 which would load the thrust bearing 21 while unloading the thrust bearing 22 by means of a rocker or compensator 20, thus causing a

reduction in the pump/engine speed ratio α , for example down to 0.85 at speed b . The volume generated by the pump is lower than the engine filling capacity; pressure P_3 (at point B') is lower than the pressure P_2 (at point B, FIG. 1). The output temperature of the supercharger, which was for instance 150° C., is reduced to about 100° C. at the outlet of heat exchanger 7, and then to about 80° C. in the manifold 11 after the expansion of the volumetric pump 10. Thus, from the point of view of combustion, the conditions thus obtained approximate the operating conditions at point A.

If in contrast thereto the engine speed decreases the ration P_2/P_1 decreases with the pressure P_2 . The control diaphragm 25 receiving this pressure as before relieves the thrust-bearing 21 while loading the other thrust-bearing 22, thus augmenting the pump/engine speed ratio α for example up to 1.15 for an engine speed c . The volume generated by the pump is greater than the engine cylinder filling capacity; the pump acts as a second stage of a supercharger. The pressure ratio P_3/P_1 (at point C') is higher than pressure ratio P_2/P_1 (at point C). The temperature at the outlet of the first supercharger stage, which may have been 50° C., is increased by the heat exchanger to about 60° C. and then to about 100° C. by the second supercharging stage 10. Thus, normal operating conditions are restored. In all the cases contemplated the scavenging action takes place normally.

The operation takes place as in the preceding example when the engine speed is d with a load reduced for example by 50 percent, the pump operating as a supercharger with a pressure P_3/P_1 at D' considerably greater than pressure P_2/P_1 at D. The temperature, after the two reheatings by the exchanger 7 and the pump 10, attains a relatively high value approximating 100° C.

The combustion conditions are nearly the same as at full load. Moreover, the counterscavenging effect usually feared at low loads cannot occur due to the high pressure P_3 at D'.

In the above-description of a typical mode of operation, P_2 is selected as a single criterion of the control-diaphragm operation; in some instances one may introduce correcting elements from pressure P_3 by properly gaging the air orifices 29 and 31. This kind of constancy in the combustion conditions permits so selecting these conditions that the engine can be operated as a multifuel engine without any risk of overstressing it.

The energy transmitted through the variable speed transmission of the pump circulates in one or the other direction according to whether the pump operates as a supercharger or as a pressure-reducing device. In no case high-power values are transmitted; however, in the supercharging direction these values are the highest when the engine power output values are the lowest. It should be noted that in this case the transmitted power is partly returned on the pistons. Since the above-defined ratio P_3/P_1 remains always relatively low, the supercharger efficiency is satisfactory and the actual loss very moderate. This loss is amply compensated by the combustion improvement, so that the overall efficiency of the engine at low engine loads is satisfactory. The reduction in fuel consumption in all engine-operating conditions constitutes one of the objects of this invention, but nevertheless the first objects are on the one hand the improvement in the power-to-weight ratio and the possibility of operating the engine on any fuel.

When starting the engine the operation is as follows:

The starter drive pinion 39 is moved to the left as seen in FIG. 2 so as to mesh both with gear ring 38 and toothed wheel 37, their teeth being shaped accordingly; at the same time it reacts against the strap 23 through linkage 40, 41 while compressing spring 24. The V-belt 15 is no longer tensioned against one of its pulleys. The pump 10 is driven only by toothed wheel 37. The pump 10 may be so designed that it does not produce any additional torque for "unsticking" the engine; as the latter begins to rotate the considerable gear ratio provided by the pump drive causes the pump to create a certain pressure P_3 in pipe 11, of course with considerable leakage and a certain reaction torque. The temperature in said

pipe 11 increases to a substantial degree, considering however the compression and the recycling of air leaks. Therefore, the cylinder walls and the lubricating oil are heated, and the temperature at the end of the compression stroke is increased very strongly. Immediately as the engine is started, the engine torque available at the gear ring 38 assisted by the force of spring 24 will push back the starter drive pinion 39 in the known manner while the V-belt 15 causes the pump 10 to be driven again under normal conditions.

In the arrangement shown in FIG. 4, the functions of the variable-diameter pulleys 16—18 and 17—19 are provided by the hydraulic pump 42 and motor 44, the hydraulic motor 44 being of the variable-capacity type. When the engine is started, the linkage 50 associated with the starter drive pinion 39 opens the bypass valve 49 to short circuit the pump 44 and the corrector 10 is then driven through the toothed wheel 37 as in the preceding example.

In order to simplify, and reduce the cost of, the V-belt and pulley variable-speed transmission described hereinabove and illustrated in FIG. 2, a single variable pulley transmission illustrated in FIGS. 5 to 8 of the drawings may be used.

According to this invention, this alternate form of embodiment is characterized by the following specific features:

The variation in diameter for controlling the transmission ratio by means of this belt and pulley device applies only to the driven pulley driving the pump;

This driven pulley acts at the same time as a belt-tensioning device;

The belt-tensioning pulley is responsive to a spring blade acting at the same time as a resilient pulley-supporting bracket;

The shaft carrying the driven pulley of the transmission is coupled through a single universal joint to the pump shaft and its length is relatively long so that the maximum amplitude of its swivel movements cannot exceed $\pm 3^\circ$ in order to minimize on the one hand the angular amplitude of driven member of this universal joint and on the other hand the angular variation of the plane of said pulley. Therefore, a single ball bearing is sufficient for supporting the shaft at the pulley end and besides the radial reaction at the pump shaft is negligible, so that it is not necessary to fit a relatively large rolling-contact bearing in this pump, as contrasted with conventional arrangements wherein the belt tension is exerted on the driven end of the pump;

The angular variation in the shaft position is attended by a very moderate torsion of the spring blade, so that the stress produced in this plane is very slight;

The variation in the plane of the driven pulley of the transmission is kept within very reasonable and permissible limits;

Finally, the pneumatic system controlling the movement of one flange of this pulley in relation to the other is mounted directly on the pulley with its return compression spring, the control air being supplied thereto through a flexible pipe.

As will be seen in FIGS. 5 to 8 to four-stroke compression-ignition engine 1 is supercharged by a turbosupercharger 52 connected through a scavenging pipe 6 to to one orifice of a Roots-type volumetric pump 10. The other orifice of this pump communicates with the induction manifold 12 of the engine. The heat exchanger of the corrector, normally interposed in this pipe 6, is not shown.

The pump shaft 13 is connected through a universal joint 53 to a hollow shaft 54 rigid with a shaft 55 carrying the right-hand flange 56 of the driven pulley, as shown in FIG. 5. The stub end 57 of this shaft 55 is mounted in a single ball bearing 58 carried by a case 59. This case 59 is secured to the outer end of a spring blade 60 having its other end anchored by means of a bracket 61 to the wall 62 of the engine cylinder-block (FIG. 8).

A sleeve 63 rigid with the left-hand flange 64 of the pulley is slidably mounted on shaft 55. The tubular shaft 54 has formed

at its pulley end an integral flange to which one end of a control diaphragm or bellows 66 is welded, the opposite end of this control diaphragm being welded to the left-hand flange 64. A compression spring 67 is enclosed in the control diaphragm and constantly urges the sliding or movable flange 64 against the V-belt 15.

The inner chamber of control diaphragm 66 communicates via an axial passage 68 of shaft 55 with the line 27 of FIG. 4 or with lines 27, 28 and 30 of FIG. 2. The pressure obtaining in the control diaphragm 66 assists the spring 67 in the lateral clamping of V-belt 15 between the pulley flanges. As can be seen, the flange 65 of shaft 54, the control diaphragm 66 and the pulley flange 64 constitute a pressure-responsive diaphragm or bellows like the devices 25 and 46 illustrated in FIGS. 2 and 4 respectively.

The V-belt 15 is mounted on the other hand on a grooved pulley 69 keyed on the crankshaft of engine 1.

The spring blade 60 is cut to a shape imparting a substantially uniform resistance thereto, so that it can clear the pulley 56, 64 while exerting a constant force substantially in the plane of this pulley.

To maintain the movement of the driven pulley in the position shown by the axis 72 in FIG. 8 a link 70 is pivoted at one end to a very light bracket or arm 71 and at the other end to the bearing case 59.

This alternate form of embodiment of the variable transmission operates as follows:

When the air pressure created for regulation purposes within the control diaphragm 66 through a flexible pipe increases so as to strongly compress the V-belt 15 by moving the pulley flange 64 toward its companion flange 56 while compressing the spring 60, the belt 15 engages a greater diameter of these pulley flanges 56, 64 so that the pulley tends to move downwards, i.e. towards the engine crankshaft. Under these conditions the shaft 54 moves but through a very small angle easily accepted by the universal joint 53, without appreciably altering the momentary angular velocities involved. Similarly, the median plane of this pulley 56, 64 varies by the same angular amplitude also accepted by the belt 15.

It will be noted that the acceptance of these small angular movements is attended by the following advantageous simplifications:

A single universal joint is used,

A single ball bearing is necessary,

The supporting bracket is merged into the spring blade (the guiding system 70, 71 being extremely simplified),

A single inlet joint for supplying air under pressure to the inner passage of shaft 55, the means for tensioning the flange 64 revolving bodily with the pulley.

It may be noted that as a counterpart of these various advantageous features the device may be attended by the following inconveniences:

The air pressure in the pulley control system adds itself to the force of return spring 67 exerting a pressure against the pulley flange 64; this pressure is uselessly important when the minimum relative distance of pulley flanges 56 and 64 is minimum (i.e. when the belt engages the largest pulley diameter and the reaction torque of the supercharger attains its lowest value and can entail a premature wear and tear of belt 15);

The belt-tensioning spring blade 60 is abnormally sensitive to the vertical accelerations "awakening" the inertia of this pulley. Therefore, some type of damping device is necessary for avoiding detrimental variations in the belt tension, belt slippage and, consequently, a premature belt wear and tear.

According to another modified form of embodiment of the present invention, which permits eliminating the two inconvenience set forth hereinabove, the transmission ratio variation is again obtained by using the action of compressed regulation air against a single pulley 56, 64 having its movable flange 64 urged by a spring 67 towards the companion flange

56 while the pulley shaft 55 is connected through a universal joint and supported as close as possible to the pulley, and on the opposite side thereof, by a single ball bearing 73 (see FIGS. 9 and 10). However, in this case a fluid-actuated cylinder is so pivotally mounted between the engine and the bearing that the movable member or piston of this cylinder causes the distance between centers of these pulleys, and therefore their transmission ratio, to vary as a function of the regulation air pressure.

Advantageously, the fluid-actuated cylinder will be of the compensated type wherein the working chamber is supplied with regulation air, the movable member of this cylinder being urged by spring means acting in a direction opposite to of said regulation air. In this case, the use of a helical compression spring as a substitute for the spring blade and its light bracket permits constructing a lighter assembly mounted directly through pivot means on the engine case or block.

A pneumatic damper may be interposed between the fixed and movable members of the fluid-actuated cylinder in order to damp out the movements of the movable member.

Finally, the use of rolling diaphragms will eliminate any problem concerning the fluid tightness in the working chamber of the fluid actuated cylinder.

As illustrated in FIGS. 9 and 10 showing a typical form of embodiment of this alternate construction, the pulley comprises a fixed flange 56 rigid with shaft 55 revolving in the single bearing 73 pivotally mounted by means of trunnions 79, and an axially movable flange 64 urged by a return spring 67 against its companion flange 56 so as to clamp the V-belt 15.

The trunnions 79 of the swiveling bearing 73 are solid with the piston rod 86 to which the upper piston 92 is welded; the upper rolling diaphragm 93 is adapted to roll on this piston 92 during the downward movement of this piston within the upper cylinder 91.

This rolling diaphragm 93 is clamped in a fluidtight manner against the bottom of piston 92 by means of a washer 74 and a nut screwed on the lower end of rod 86 and adapted to assemble the lower piston 78, the lower diaphragm 77, the washer 76, the distance-piece 75, the washer 74 and the aforesaid diaphragm 93 by tightening against said piston 92. The outer edge of diaphragm 93 is clamped in a fluidtight manner between the lower flange of cylinder 91 and the upper flange of the intermediate cylinder 94. The outer edge of the lower diaphragm 77 is clamped also in a fluidtight manner between the connecting flanges of the intermediate cylinder 94 and of the lower cylinder 88.

The fluidtight chamber bounded by diaphragms 77 and 93, their clamping washers 74 and 76, distance-piece 75 and intermediate casing 94 is supplied with regulation air pressure through a flexible pipe connected to port 80. This pressure compensated by spring 82 is used for adjusting the vertical position of the pistons, rod 86 and pump 10, by varying the distance between the centers of this pulley and of the fixed driving pulley, and therefore the transmission ratio between these pulleys.

The spring 82 centered with respect of the bottom of the lower piston 78 by a washer 81 is centered at its opposite end by a shallow neck secured directly to the case 87. By thus causing the spring 82 to react directly against the case 87 the trunnions or pivot pins 85 are relieved from undue stress and subjected only to the forces produced by the regulation air.

The chamber 90 between the lower cylinder 88 and the piston 78 which is bounded at its ends by the diaphragm 77 and the piston packing 84, communicates with the free atmosphere through a gauged orifice 83 and acts as a dashpot or pneumatic damper for damping out the oscillations resulting from the piston movements.

The assembly, except the spring 82, is pivotally supported by said trunnions 85 bearing on brackets 89 rigid with the engine case 87.

This invention is also concerned, as an alternate solution to the problem of, and a substitute for the coupling and step up transmission 23, 24, 37, 39 to 41, and 37, 39, 49 and 50 illus-

trated in FIGS. 2 and 4 respectively, with a device adapted to eliminate or at least reduce internal leakages in the pump 10, which is illustrated very diagrammatically in FIGS. 11 and 12.

Referring first to FIG. 11 it will be seen that a compression-ignition engine 1 is supercharged by means of a turbosupercharger 52 and a Rootes pump 10 driven at a variable speed by the pulley 95, belt 15 and pulley 96.

The heat exchange device of the corrector is not shown.

The volumetric supercharger is driven from the engine crankshaft by the variable-speed transmission causing this supercharger to generate an output-volume now greater, now smaller than the cubic capacity of the engine cylinders.

The variable speed drive is not illustrated. This device may be of any suitable and known type, for example a V-belt variator, or a hydraulic pump and hydraulic motor assembly of the barrel-piston type, etc.

Conventionally, the engine is equipped with a starter motor 97, a fuel feed pump 98 and a fuel injection pump 99. In the pipe or duct 6 connecting the turbosupercharged 52 to the pump 10 a fuel injection is produced at 100, i.e. slightly upstream of the volumetric pump 10.

The fuel is fed via a pipeline 101 and the output is adjusted by properly selecting the orifice diameter of the fuel jet 102 as well as the pressure at the outlet of the feed pump 98 (FIGS. 11 and 12).

Under normal operating conditions this output is cut off by the electromagnet valve 103 closed when energizing current is fed to its winding. This current is supplied through an electric conductor 104 connected to the starter motor supply conductor 106 via a switch 107 responsive to a thermostatic control device 108. This thermostatic control device may consist preferably of a bellows capsule immersed in the engine cooling liquid flowing through the duct 109, as shown.

This device operates as follows:

Assuming that the engine is inoperative and the water temperature rather low, contact 107 is closed. When the switch 106 controlling the energization of starter motor 97 is closed the electromagnet valve 103 opens as the engine performs its first revolutions. The fuel pump 98 delivers a pressure depending on the force of its spring, and a fuel output is directed through the jet 102. Thus, the pump 10 is driven from the engine with the maximum transmission ratio but at a very low speed.

The fuel mixed with air penetrated into the pump 10 and wets the inner walls of its stator as well as the rotor surfaces. The clearances between stator and rotors and between rotors, which are normally of the order of 0.004 inches, practically cancel each other and the volumetric ratio between the pump 10 and engine 1 creates the expected overpressure. The final pressure and temperature in the engine combustion chambers become sufficient for causing the first combustion, inasmuch as the latter is promoted by the slight vapor caused by this additional fuel injection.

Of course, the size of the orifice of jet 102 is selected after proper tests and also as a function of the temperature at which the switch 107 is to be operated.

We claim:

1. A compensated supercharging device for a compression-ignition engine cooled by a liquid at a regulated temperature and already supercharged by a turbosupercharger, said device tending to correct the known deficiencies of this type of supercharging by regulating the temperature and pressure of the air delivered by said turbosupercharger before it is introduced into the engine for all speed and load conditions of said engine, said device comprising a duct connecting said turbosupercharger to said engine, a heat exchanger in said duct, means for circulating an engine cooling liquid through said heat exchanger at a regulated temperature, a volumetric air machine mounted in said duct between said exchanger and said engine, transmission means operatively connecting said volumetric air machine to a crankshaft of said engine so that the ratio of the speeds of said air machine and said engine will vary continuously between two limits, one of said limits

producing an output volume of the machine which is greater than the engine cylinder cubic capacity so as to produce an over pressure and the other of said limits producing an output volume of said machine which is lower than that of said engine cylinder cubic capacity so as to expand the air pressure, a control diaphragm operatively connected to said transmission means to control said speed ratio variation in response to a reference pressure resulting from a predetermined combination of pressure prevailing upstream of said volumetric air machine and pressure prevailing downstream thereof, said speed ratio variation being controlled inversely to changes in said reference pressure.

2. A device as set forth in claim 1 in which said supercharger comprises a high efficiency turbosupercharger having the highest possible pressure ratio at maximum engine speed.

3. A device as set forth in claim 1 wherein said volumetric air machine comprises a gear pump having at least two teeth.

4. A device as set forth in claim 1 in which said transmission comprises a belt-type variable-speed transmission having at least one variable-diameter grooved pulley, a swivel shaft means connecting said pulley to volumetric air machine, said swivel shaft means comprising a bearing, a universal joint and a resilient support, said swivel shaft means being relatively long so that the maximum amplitude of its swivel movements cannot exceed $\pm 3^\circ$, the movements of the pulley flanges towards and away from each other being controlled by said reference pressure.

5. A device as set forth in claim 1 in which said transmission is of the hydraulic type and comprises a hydraulic pump and hydraulic motor assembly of the piston-barrel and swash-plate

type, the speed variation resulting from the action exerted by said reference pressure on the inclination of said swash plate.

6. A device as set forth in claim 3 further comprising special starting means provided when said gear pump is rendered inoperative as a consequence of internal leakages at very low engine speeds, notably at engine starting speeds, said starting means being adapted, during the rotation of a starter motor of said engine, to inject upstream of said gear pump a liquid capable of adhering by surface tension to the internal surfaces of said pump so as to temporarily seal the leakages thereof, said injection means comprising a jet disposed at the inlet end of said duct, an electromagnetic valve and a liquid supply, the circuit for energizing said electromagnetic valve being established from a storage battery and said starter motor and comprising a switch responsive to a thermostatic control diaphragm capable of opening said circuit when the temperature of the engine cooling liquid has reached a sufficient value.

7. A device as set forth in claim 6 in which said sealing liquid comprises engine fuel supplied from the engine fuel feed pump.

8. A device as set forth in claim 3, further comprising means for minimizing the influence of leakages by increasing very considerable proportions the drive ratio of said gear pump, said means momentarily uncoupling said transmission and substituting therefor a direct step up gear providing a higher gear ratio, said uncoupling being controlled by a drive pinion of a starter motor through a linkage, the direct step up gear being obtained by the meshing action of said starting motor drive pinion on a tooth wheel mounted on a shaft of said volumetric air machine.

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