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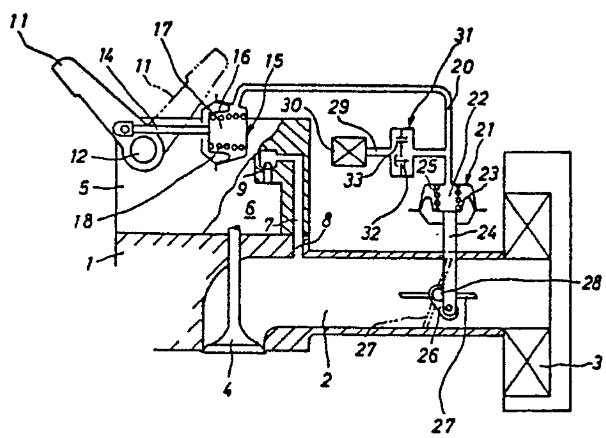
(54) Starting system for diesel engine.

(57) An intake flow limiting valve (27) for limiting suction air flow is installed in an intake passage (2) of a diesel engine. Engine starting torque is then reduced by throttling intake flow, using the intake flow limiting means (27), at the time of starting the engine.

FIG.1



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STARTING SYSTEM FOR DIESEL ENGINE

This invention relates to a starting system for a diesel engine.

In a conventional diesel engine, a valve has not generally been installed in a suction passage for throttling an intake flow.

Throttling of intake flow has been proposed in order to abate noise produced during idling or to improve ignitability upon starting. However, it has now been the practice to throttle the intake flow to such an extent that the starting torque can be reduced when starting the engine.

Since the intake flow has not conventionally been throttled as mentioned above, the starting torque has been large, and a strong force has been required for hand starting the engine, e.g. by pulling a rope or similar by hand, or else a starter motor of large capacity has been required for electric starting.

It was conventionally believed that due to a drop in the maximum air temperature in the combustion chamber during the compression stroke ignition would not be generated if the intake flow was throttled to such an extent that the starting torque could be reduced. Therefore, in the conventional proposal for throttling the intake flow for improvement of ignitability at time of starting, the intake flow was only throttled to the extent of decreasing engine charging efficiency by a few percent for the purpose of forcing suction air to raise the maximum temperature during compression. It did not include the concept of throttling the intake flow to the extent of effectively reducing the starting torque. Further, in the conventional proposal for throttling the intake flow in order to abate noise during idling, the intake flow has been considerably throttled for the purpose of securely continuing the idling. This conventional method has been applicable only to idling, not to reducing the starting torque at time of starting.

It has now become apparent from experiments conducted by the inventor that the maximum temperature during compression has not been lowered, as compared with a case where the intake flow is kept entirely unthrottled, even when the intake flow has been throttled to the extent of effectively reducing the starting torque, and the present invention has been made on the basis of this fact.

In the starting system for a diesel engine according to this invention, intake flow limiting means for limiting suction air flow is installed in an intake route of the diesel engine so that intake flow is throttled to the extent of reducing the engine starting torque upon engine start up.

When the first suction stroke of the intake flow is throttled by the intake flow limiting means, the force necessary for lifting the piston in the next compression stroke is small because the air volume in the combustion chamber is small. Moreover, the maximum temperature during compression is not lower than that obtained by unthrottling the intake flow. Therefore, in the case of hand starting the diesel engine, the force necessary for starting can be effectively reduced so that the starting operation may be done easily even by a weak operator. In the case of an electric starting diesel engine, since the cranking force can be reduced the capacity of the starter motor can be minimized so that engine weight and manufacturing cost can be decreased.

Furthermore, the engine starting torque can be reduced more effectively when the intake flow is throttled by the intake flow limiting means to the extent that the charging efficiency becomes approximately 70% or less than 70% of that obtained by unthrottling the intake flow.

It is not necessary to provide a throttle passage in the intake flow limiting means itself, and in accordance with a preferred embodiment of the invention, engine structure can be simplified if the intake route is made to communicate with a crank case interior in a freely opened and closed manner while the intake flow is throttled by the intake flow limiting means so that a breather passage for keeping the crank case interior pressure below a specified value is also utilized as a throttle passage for feeding air to a combustion chamber.

A specified practical embodiment of the system of the invention includes a valve for limiting suction air flow disposed in an intake route of a diesel engine, a pneumatic pump device interlinking with an exhaust decompression lever of automatic releasing type, a pneumatic actuator opening and closing the valve, a connecting passage interconnecting an air chamber of the pneumatic actuator with an air chamber of the pneumatic pump device, and a valve device comprising a check valve and a throttle which are both installed in parallel in the connecting passage so that the pneumatic actuator, owing to the functioning of the check valve, is not actuated when the decompression lever is operated, but is actuatd to close the valve upon return of the decompression lever, the valve being subsequently opened after a specified time has elapsed by air passing through the throttle.

By throttling the intake flow in this way when starting the engine the engine torque upon start up can be reduced.

With the aforesaid system there is no possibility of lubricating oil etc. entering the pneumatic circuit, so there is no risk that the throttle of the valve device might be blocked by oil etc., because the pneumatic circuit for driving the valve opens to atmosphere only through the valve device and the other devices are entirely enclosed. Accordingly this system is superior in reliability and durability to others without these features.

Furthermore, malfunction and failure due to friction etc. do not occur because mechanical parts such as a link mechanism are not included in the system, so in this respect the system is also superior in reliability and durability. Moreover, the entire structure is comparatively simple, thus enabling inexpensive manufacture.

One further advantage is that since the valve can be closed automatically by the return action of the exhaust decompression lever, special closing operation of the valve is not required so that the starting operation can be carried out easily and quickly.

Fig. 1 is a schematic structural diagram of one embodiment of a diesel engine starting system in accordance with the present invention;

Fig. 2 is a plan view of a diesel engine equipped with the starting system show in Fig. 1;

Fig. 3 is a front view of the same diesel engine;

Fig. 4 shows explanatory diagrams of working conditions at the time of engine starting; and

Fig. 5 shows graphical representations of the relationship between intake throttle ratio and (a) charging efficiency, (b) maximum temperature during compression, and (c) maximum pressure during compression.

In Fig. 1, the diesel engine is shown as having an intake route which, by way of example, takes the form of an intake passage 2 formed in a cylinder head 1. An upstream end of the intake passage 2 opens to atmosphere through an air filter 3, whilst a suction valve 4 is disposed at the dowstream end of the intake passage 2. A bonnet (or cover) 5 is secured onto the cylinder head 1. A rocker arm chamber 6, formed within the bonnet 5, communicates with the intake passage 2 through a passage 7 formed in the bonnet 5 and a passage 8 formed in the cylinder head 1. A ball valve 9, which permits air to flow past only in a direction from the rocker arm chamber 6 to the intake passage 2, is disposed in the passage 7.

The passages 7 and 8 constitute a breather passage which connects the intake passage 2 with the interior of a crank case in a freely opened and closed manner so as to keep the pressure in the crank case below a specified value, and this breather passage also serves as a throttle passage for feeding air to a combustion chamber when a throttle valve, described later, is closed.

A shaft 12 for an exhaust decompression lever (abbreviated to "decomp lever" hereinafter) 11 of automatic releasing type (e.g. auto-return type) is rotatably mounted on the bonnet 5. A return spring 13 (Fig. 2) is fitted around the shaft 12. One end of a rod 14 is connected by a pin to the decomp lever 11 in the vicinity of the shaft 12. The other end of the rod 14 is connected to a diaphragm 16 of a diaphragm pump device 15, given as an example of a pneumatic pump device. A coil spring 18 urging the diaphragm 16 toward the rod 14 is installed in a chamber 17 of the diaphragm pump device 15. This diaphragm chamber 17 communicates with another diaphragm chamber 22, of a diaphragm actuator 21, which is given as an example of a pneumatic actuator, through a tube 20 constituting a connecting passage. One end of a rod 24 is connected to the diaphragm 23 of the actuator 21. A coil spring 25 urging the diaphragm 23 toward the rod 24 is installed in the diaphragm chamber 22 of the actuator 21. The other end of the rod 24 is connected by a pin to one end of a lever 26. The other end of the lever 26 is fixedly attached to a throttle valve 27, which constitutes intake flow limiting means and which is installed rotatably in the intake passage 2.

A branch pipe 29 branches off from the connecting pipe 20 and opens to atmosphere through an air filter 30. A valve device 31, which is composed of a check valve 32 and a throttle 33 connected in parallel with each other, is installed in the branch pipe 29. The check valve 32 permits air to pass only from the connecting pipe 20 side to the air filter 30 side.

Although not illustrated in Fig. 1, the rocker arm chamber 6 communicates with the inside of the crank case in generally known manner. Furthermore, when the decomp lever 11 is swung clockwise from its position shown by a solid line to that shown by a two-dot chain line in Fig. 1, an exhaust valve is pushed down and opened in generally known manner.

In the case of a hand starting engine, a piston is moved vertically by the pulling of a starting rope, the exhaust valve is further pressed down to a fully opened state, and the decomp lever 11 is swung counterclockwise by the biassing force of the return spring 13 and returned to its position shown by the solid line, as is commonly known.

In this illustrated structure, when the decomp lever 11 is swung to the right side of Fig. 1, into its exhaust decomp position as illustrated by the two-dot chain line, the exhaust valve opens to provide the exhaust decomp state, the rod 14 is pushed to the right causing the diaphragm 16 of the diaphragm pump device 15 to distort against the bias of the coil spring 18, and air in the diaphragm chamber 17 is squeezed into the connecting pipe 20. However, the check valve 32 of the valve device 31 is opened thereby so that the air is released to atmosphere through the check valve 32. Accordingly, the pressure in the diaphragm chamber 22 of the diaphragm actuator 21 does not increase and the throttle valve 27 is kept opened.

When the starting rope is gradually pulled, the piston begins vertical motion, the exhaust valve is opened beyond the exhaust decomp state just as the piston starts the first exhaust stroke, and the decomp lever 11 is swung counterclockwise by the bias of the return spring 13 and returned to its position illustrated by the solid line. The rod 14 is thereby pulled toward the left, causing the diaphragm 16 of the diaphragm pump device 15 to distort, and pressures become negative in both the diaphragm chamber 17 of the diaphragm pump device 15 and the diaphragm chamber 22 of the diaphragm actuator 21 which are connected to each other by the connecting pipe 20. Since the check valve 32 is, in this instance, closed by the negative pressure, there is no chance for atmosphere to ingress through the check valve 32. When the pressure in the diaphragm chamber 22 of the diaphragm actuator 21 becomes negative the diaphragm 23 distorts against the bias of the coil spring 18, the rod 24 is pulled upward, and the lever 26 is swung counterclockwise; so that the throttle valve 27 is fully closed as illustrated by the two-dot chain lines. Thereafter, the pressure in the diaphragm chamber 22 of the diaphragm actuator 21 gradually returns to near to atmosphere owing to air flowing in through the throttle 33 of the valve device 31, as indicated by the graph in Fig. 4(b).

When the negative pressure in the diaphragm chamber 22 of the diaphragm actuator 21 rises up to a pressure P1 balancing with a setting load exerted by the coil spring 25, the throttle valve 27 starts opening and it is fully opened in the final stage as indicated by the graph of Fig. 4(c).

The point where the throttle valve 27 begins to open can be shifted to a point after completion of the first suction stroke, as shown in Fig. 4(c), by setting the sectional area of the throttle 33 of the valve device 31 to an appropriate value.

In Fig. 4, (a) represents the timing of the function of the decomp lever 11, (b) represents pressure change in the diaphragm chamber 22 of the diaphragm actuator 21, and (c) represents the changes in the opening of the throttle valve 27.

First fuel injection is carried out immediately before the first combustion stroke, but effective work is not performed, even if ignition occurs, because the engine is under the decomp state. When the suction valve 4 opens and the piston moves downward during the first suction stroke, pressure in the intake passage 2 becomes negative because the throttle valve 27 is closed, and the ball valve 9 is thereby opened to supply air from the rocker arm chamber 6 to the combustion chamber through the passages 7 and 8 and the intake passage 2. Since the sectional areas of the passages 7 and 8 are very small as compared with that of the intake passage 2, the air volume fed to the combustion chamber is, in this instance, smaller than that fed when the throttle valve 27 is opened so that the intake flow is throttled as the result.

In the second compression stroke, the decomp lever 11 has been returned and the exhaust decomp state is released to close the exhaust valve. A large force is not necessary to lift up the piston because the intake flow in the first suction stroke is small.

Fuel is injected and ignited immediately before the second combustion stroke. Since the throttle valve 27 is opened, as indicated by Fig. 4(c), on and after the second suction stroke, the intake flow is not throttled and engine speed increases to reach a fixed value in the same way as a conventional engine.

The relationships between suction throttle ratio (the ratio of intake flow sectional area with the throttle valve 27 fully opened and the intake flow unthrottled, to intake flow sectional area with the intake flow throttled) to engine charging efficiency, to maximum temperature during compression and to maximum pressure during compression, are as indicated by Fig. 5.

If the intake flow is throttled so as to lower the charging efficiency down to 70% or less in the first suction stroke, the maximum temperature during compression becomes approximately the same as that attained when the throttle valve 27 is fully opened, as indicated by Fig. 5(b). This indicates that ignition is reliably accomplished at second fuel injection.

Also, the maximum pressure during compression is about 30 kg/cm^2, as indicated in Fig. 5(c), which is smaller by about 10 kg/cm^2 than that obtained when the throttle 27 is fully opened. Consequently, when the intake flow is not throttled in the first suction stroke as is the case in conventional systems, it is necessary to pull the rope with a very large force in the second compression stroke immediately before completion of pulling the starting rope. In contrast, if the intake flow is throttled so as to lower the charging efficiency down to 70% or less in the first suction stroke as disclosed in this embodiment, the rope pulling force can be lessened effectively. Naturally, the rope pulling force may be lessened much more by further throttling the intake flow within an ignitable range.

Since the intake flow is throttled in the first suction stroke by the throttle valve 27 as described above, the torque required for lifting the piston in the second compression stroke can be reduced effectively. Therefore, in case of a hand starting diesel engine, the force required for starting the engine can be reduced effectively and the starting operation can be done easily even by a weak operator. In case of an electric starting diesel engine, the capacity of the starter motor can be minimized so that the engine weight and manufacturing cost can be reduced. When the intake flow is throttled by using the throttle valve 27 so as to lower the engine charging efficiency down to about 70% or less compared to that obtained when the intake flow is not throttled, the engine starting torque can be minimized very effectively.

Since the passages 7 and 8 constituting the breather passage for keeping the pressure in the rocker arm chamber 6 below a specified value are utilized also as the throttle passage for feeding air into the combustion chamber during closing of the throttle valve 27, provision of an additional hole for use as the throttle passage in the throttle valve 27 is not necessary and manufacturing cost can, accordingly, be reduced.

There is no possibility of lubricating oil etc. entering the pneumatic circuit and consequently no risk that the throttle 33 of the valve device 31 might be blocked by oil etc. because the pneumatic circuit for driving the throttle valve 27 opens to atmosphere only through the valve device 31 and the other devices are entirely enclosed.

Furthermore, since the throttle valve 27 can be closed automatically by the return action of the exhaust decomp lever 11, special means for closing the throttle valve 27 is not required and the starting operation can be carried out easily and quickly.

The diaphragm actuator 21 is driven by negative pressure in the above-mentioned embodiment, but in other embodiments it could be driven by positive pressure, in which case the actuating direction of the check valve 32 of the valve device 31 would be reversed as compared with the above-mentioned embodiment.

The diaphragm pump device 15 and/or the diaphragm actuator 21 used as the pneumatic pump device and pneumatic actuator, respectively, in the above-mentioned embodiment could each be replaced by a bellows, for example.

Claims

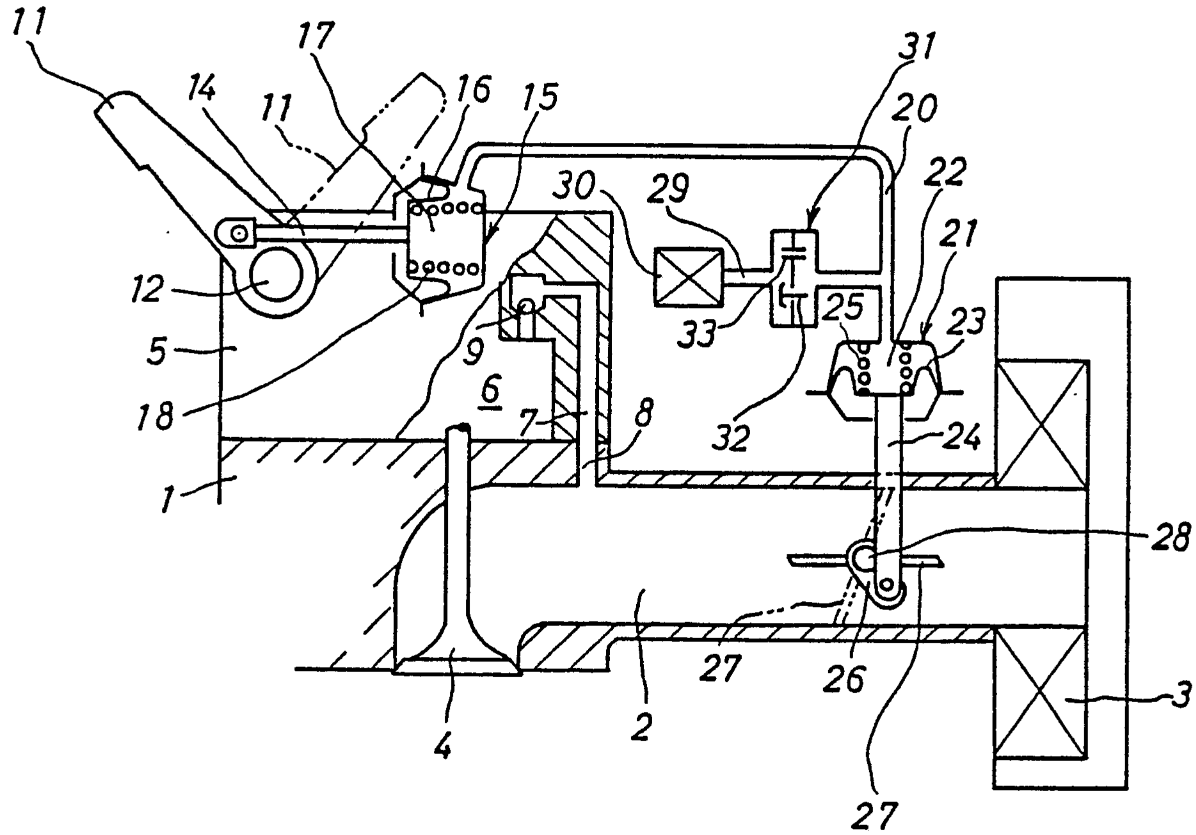
1. A starting system for a diesel engine wherein intake flow limiting means (27) for limiting suction air flow is installed in an intake passage (2) so as to throttle intake flow to such an extent as to reduce engine starting torque upon engine start up.

2. A starting system for a diesel engine as claimed in claim 1, wherein intake flow is throttled by the intake flow limiting means (27) so as to lower engine charging efficiency down to approximately 70% or less compared to that attained when the intake flow is not throttled.

3. A starting system for a diesel engine as claimed in claim 1 or 2, wherein breather passages (7, 8) connecting the intake passage (2) with a crank case interior in a freely opening and closing manner to keep pressure in the crank case interior below a specified value, are also utilized as throttling passages for feeding air into a combustion chamber when the intake flow is throttled by the intake flow limiting means (27).

4. A starting system for a diesel engine comprising a valve (27) for limiting suction air flow disposed in an intake passage (2) of the diesel engine, a pneumatic pump device (15) interlinking with an exhaust decompression lever (11) of automatic releasing type, a pneumatic actuator (21) opening and closing said valve (27), a connecting passage (20) interconnecting an air chamber (22) of the pneumatic actuator (21) with an air chamber (17) of said pneumatic pump device (15), and a valve device (31) comprising a check valve (32) and a throttle (33) which are both installed in parallel in the connecting passage (20) so that, owing to the functioning of said check valve (32), said pneumatic actuator (21) is not actuated when said decompression lever (11) is operated, but it is actuated to close the valve (27) upon return of the decompression lever (11), said valve (27) subsequently being opened by air passing through said throttle (33), and so that by throttling the intake flow in this way when starting the engine, the engine torque upon start up can be reduced.

FIG.1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31
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12
33
25
23
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18
7
8
32
24
1
28
27
26
27
4
2
3
27

FIG.4

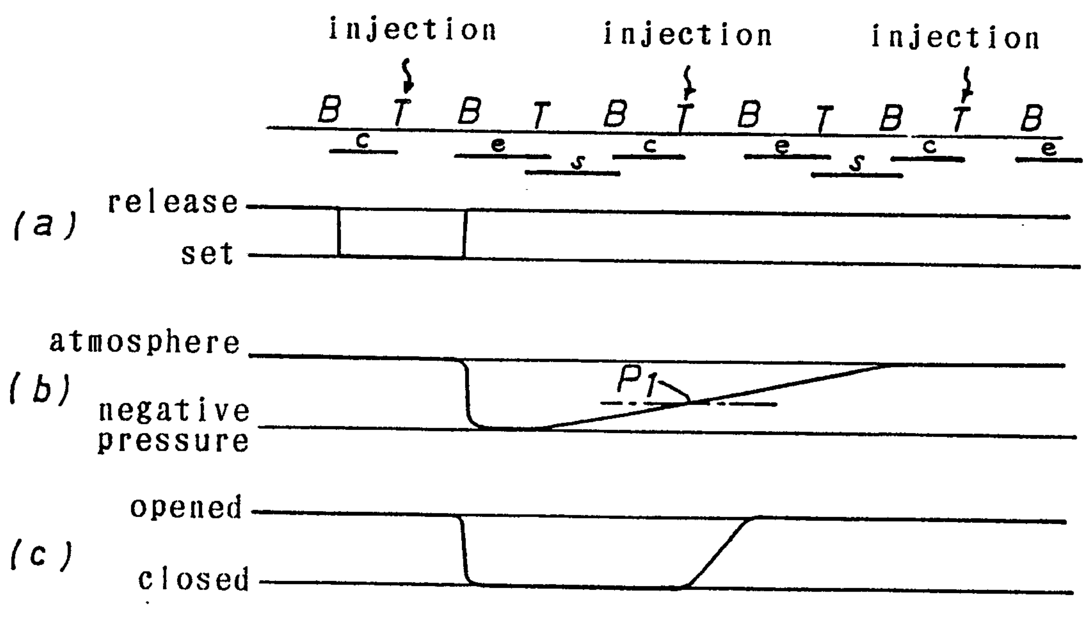

injection
injection
injection
B
T
B
T
B
T
B
T
B
T
B
c
e
s
c
e
s
c
e
(a)
release
set
atmosphere
(b)
P1
negative pressure
opened
(c)
closed

FIG.2

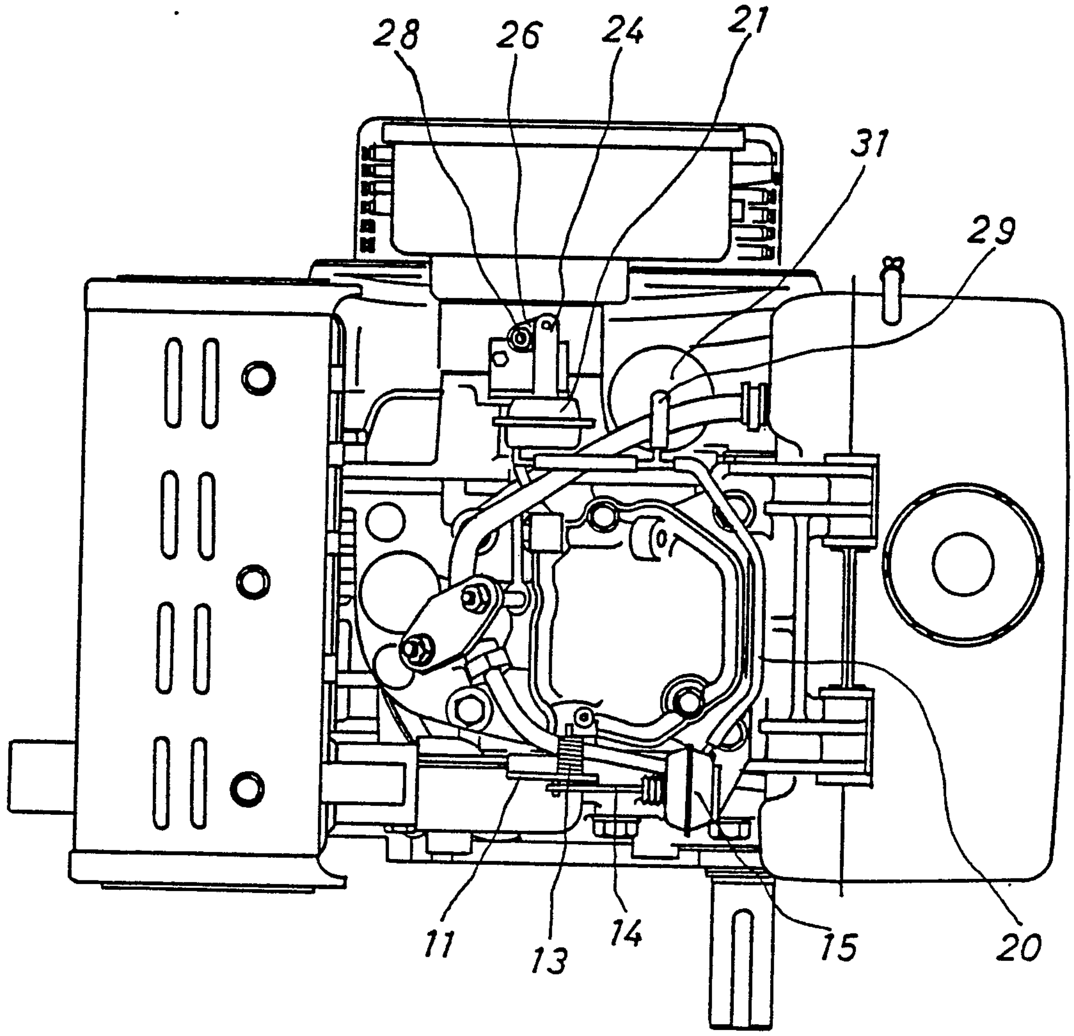


FIG.3

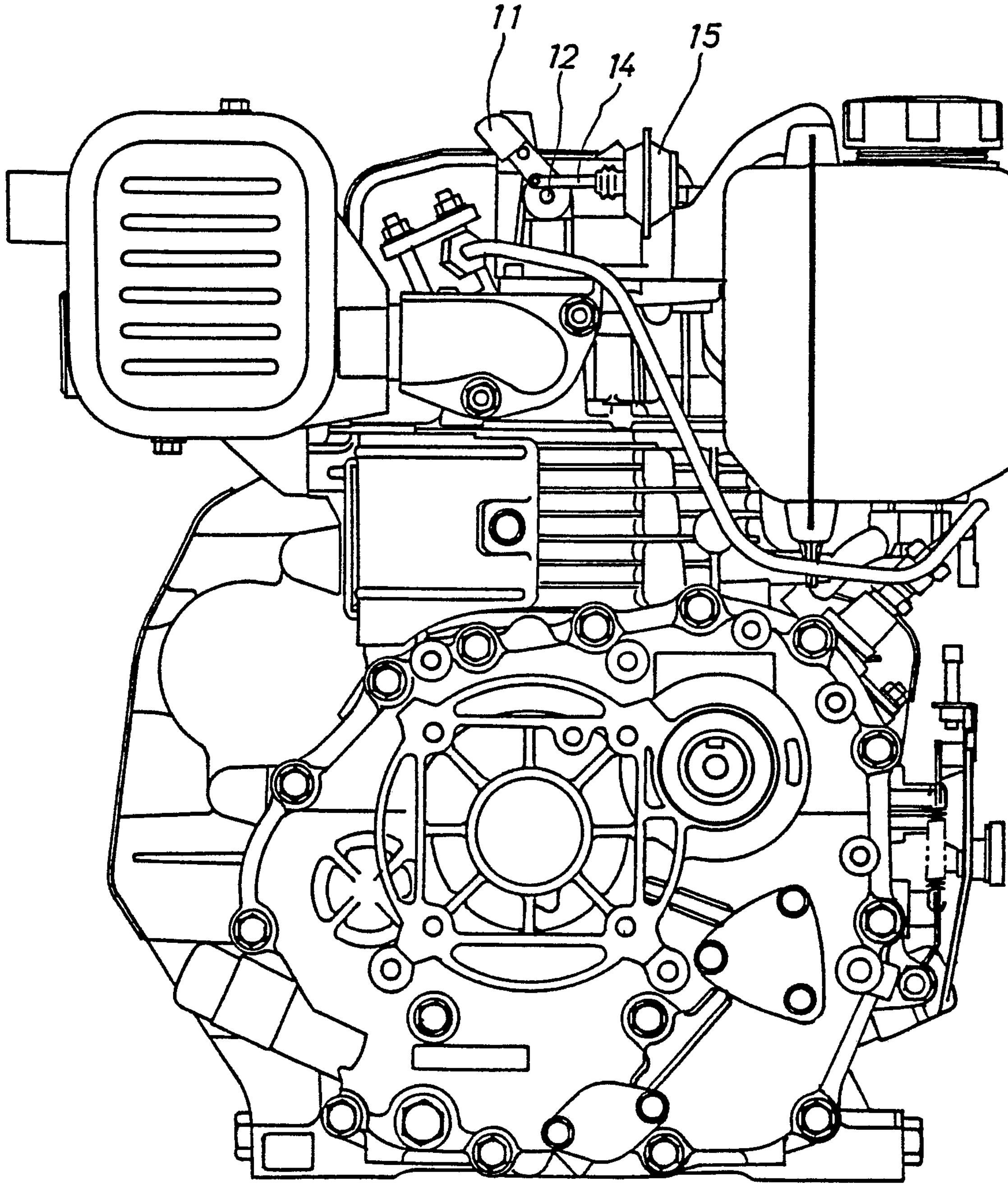
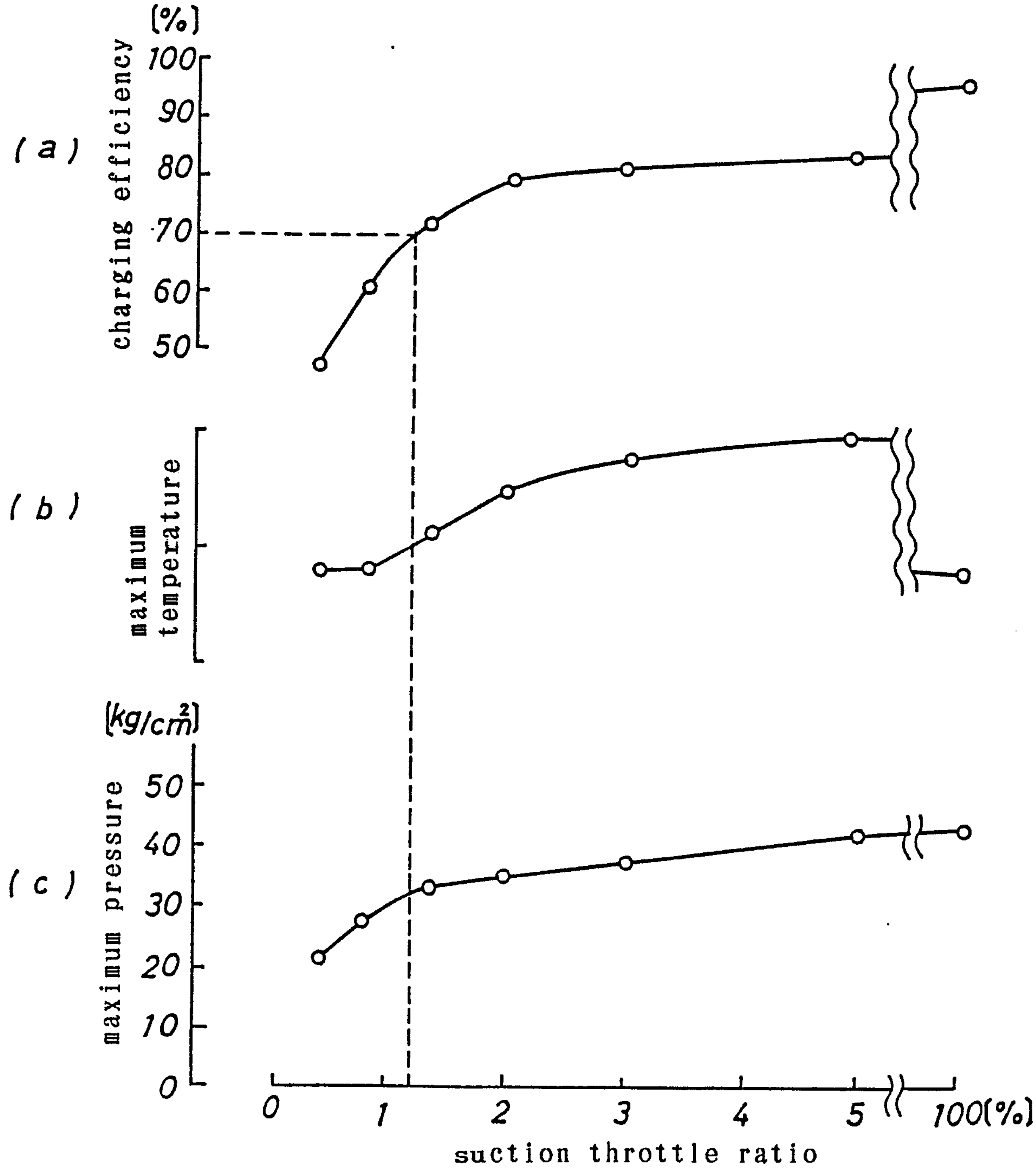


FIG. 5


(%)
100
90
80
70
60
50
(a)
charging efficiency
(b)
maximum temperature
(c)
[kg/cm²]
50
40
30
20
10
0
maximum pressure
0
1
2
3
4
5
100(%)
suction throttle ratio

Europäisches Patentamt

EUROPÄISCHER RECHERCHENBERICHT

Nummer der Anmeldung

EP 90 30 0723

EINSCHLÄGIGE DOKUMENTE

Kategorie	Kennzeichnung des Dokuments mit Angabe, soweit erforderlich, der maßgeblichen Teile	Betrifft Anspruch	KLASSIFIKATION DER ANMELDUNG (Int. Cl.5)
X	PATENT ABSTRACTS OF JAPAN Band 7, Nr. 177 (M-233)(1322), 5. August 1983; & JP - A - 58 79634 (NISSAN) 13.05.1983	1	F 02 D 9/02 F 02 N 17/08
A	idem ---	2	
A	EP-A-0 028 745 (KNORR-BREMSE) * Seite 1, Zeilen 18-23; Seite 3, Zeile 32 - Seite 4, Zeile 12 * ---	1-3	
A	DE-A-3 024 731 (ISUZI) * Seite 8, Zeilen 2-19; Figur 1 * -----	4	
			RECHERCHIERTE SACHGEBIETE (Int. Cl.5)
			F 02 D F 02 N

Der vorliegende Recherchenbericht wurde für alle Patentansprüche erstellt

Recherchenort	Abschlußdatum der Recherche	Prüfer
BERLIN	08-05-1990	NOVELLI B.

KATEGORIE DER GENANNTEN DOKUMENTE

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