(12) UK Patent Application (19) GB (11) 2 110 298 A

(21) Application No **8228073**
(22) Date of filing **1 Oct 1982**
(30) Priority data
(31) **813071**
(32) **2 Oct 1981**
(33) **Finland (FI)**
(43) Application published
15 Jun 1983
(51) **INT CL³**
F02B 27/04 F01N 1/08
(52) Domestic classification
F1B F202 F222 F230 F236 F238 F239 F320 FB FL
(58) Field of search
No search possible

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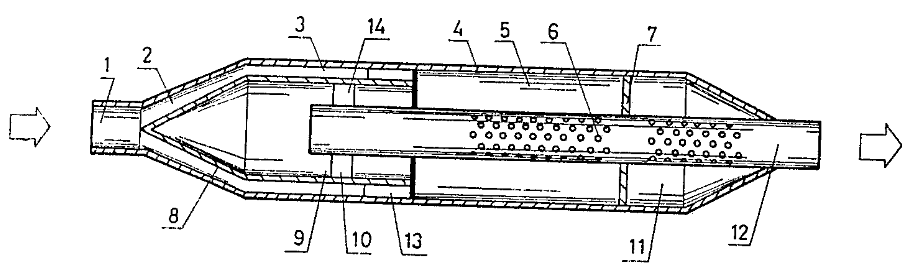
(54) **Internal combustion engine exhaust systems**

(57) A silencer mounted close to the engine exhaust manifold defines a conical flow passage 2 leading to an annular passage 3 containing vanes which set up a swirl in the gases entering the expansion chamber 5 before entering the outlet pipe 12 via the passage 9 and the perforations 6. The flow of gas in the silencer serves to reduce the pressure in the exhaust manifold.

Fig. 1



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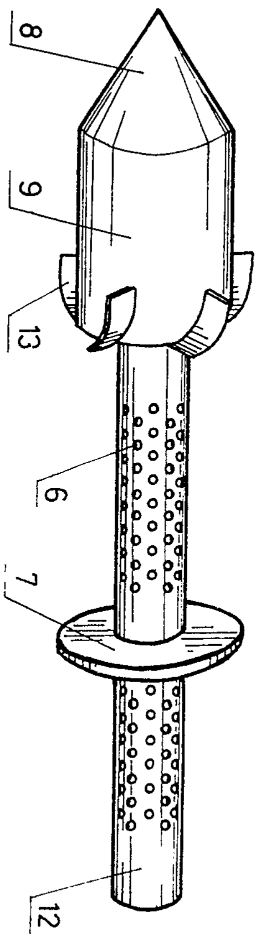

Fig. 2

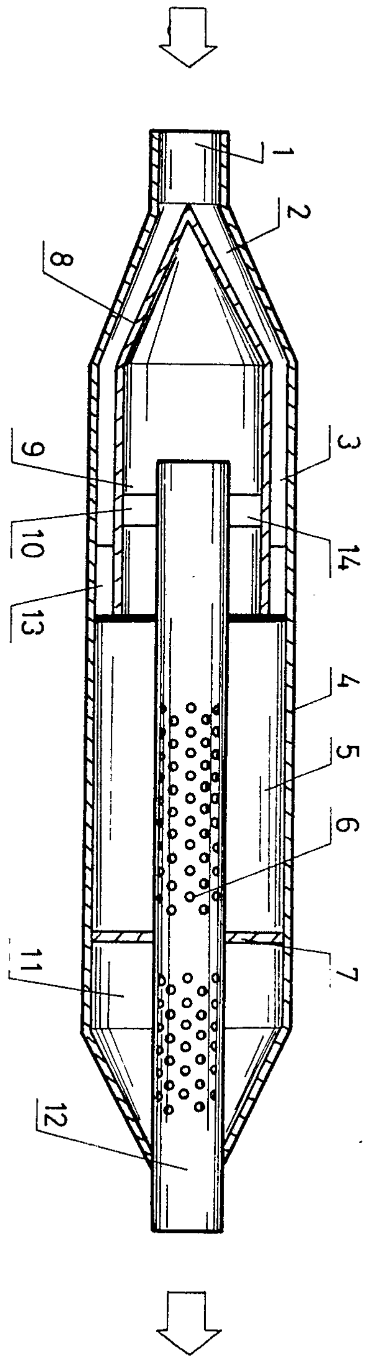

Fig. 1

SPECIFICATION

A process and a device for improvement in the performance and the efficiency ratio of an internal combustion engine

The subject matter of this invention is a process for the improvement in performance and efficiency ratio of an internal combustion engine, by improvement of the gas flow through the pipe connected to the engine exhaust manifold and the continuation to one or several successive silencer cases, together with the method for execution of the process device.

In the context of present-day technology the economical efficiency ratio of petrol engines is extremely poor under partial loading, as for example, when driving in normal traffic conditions and this according to the predominant understanding, is mainly due to the relatively large effects of mechanical losses. This explanation is not however satisfactory when it has been verified in comparing diesel and petrol engines of the same power with one another that the economical efficiency of diesel engines is conclusively superior under partially loaded conditions. When it is considered that the friction surfaces of diesel engines with their larger swept volume are greater and their moving parts more massive, the poor performance of the petrol engine cannot be ascribed to the effects of mechanical losses.

It has been shown in performed researches that the poor performance of the petrol engine at partial loading is conclusively due to the fact that because of its quality and low compression pressure the combustion speed of the fuel mixture is clearly slower than when the engine is under heavy load. This can be seen for example on an indicator diagram taken at 20% load and about 2000 r/min, in which from a calculated value of the polytropic exponential, the middle and end part of the expansion is approximately 1. This proves that despite the expansion and thermal losses the gas temperature has stayed above constant, thus the process has received heat energy throughout the expansion period, in other words the combustion of the mixture has been carried on strongly throughout the expansion. The efficiency ratio is then exceedingly poor because the later the chemical energy of the combustion medium is converted through combustion into pressure energy the less chance there is for the piston to transfer it into mechanical work.

The knocking phenomenon also has a central significance in that it prevents increase in the compression ratio from the existing level, which would otherwise be the simplest way to raise the compression pressure and thus speed up the combustion process at part-loading. In addition to this the knocking phenomenon also prevents any meaningful lowering of the fuel grade octane number from the existing level, without the necessity of correspondingly lowering the compression ratio, which would then lessen the economy of fuel economy. Thus the knocking phenomenon fundamentally increases the problems associated with transference to combustion of lead-free petrol grades.

With respect to exhaust systems for use in passenger vehicles it must be concluded that not enough attention has been paid in the design of these exhaust systems for conventional vehicles to the size and nature of the back-pressure which they create, the obvious reason for this being that this pressure is relatively insignificant compared with the mean effective pressure of the engine. In designing silencers the main emphasis is placed on good muffling characteristics and thus silencers operating on the so-called impedance principle are commonly used. It is characteristic of these silencers that sound absorption is achieved by steep changes in cross-sectional surface of that duct through which the exhaust flow mainly occurs. A steep change in the cross-sectional surface of the duct will result in a steep drop in the gas flow velocity and as can be seen from a Carnot-Borda diagram the magnitude of the stroke (impact) loss is fundamentally dependent on the reading of the factor illustrating the change in cross-sectional surface area. Since the sound absorption attained, shown in the diagram in dB, also depends on the same factor, it can be stated that the higher the stroke (impact) losses the more efficient the silencing. The pressure shock associated with the stroke (impact) loss easily leads to vibration phenomena in the silencer casing and related sound effects, so that in these silencers a double case, sometimes provided with a further internal absorbent layer, is generally used.

A fundamentally influential factor is the nature of the exhaust gas flow. When the exhaust valve opens due to its high pressure ratio the exhaust gas first flows into the exhaust manifold at critical velocity and from there on through the connecting tube at high speed towards the silencer. There the gas rushes onto the gas already present in both exhaust manifold and connection pipe, this resulting in the gas becoming denser and a consequent increase in pressure and deceleration of flow velocity, so that the continually expanding portion of the dynamic energy in the gas is converted into pressure energy. The progressing velocity of the pressure wave thus formed as well as the prevailing pressure are dependent on the circumstances and the pressure wave cannot be compared in its characteristics with a sound wave. Because of the high velocity of the pressure wave the stroke (impact) losses are high and associated with a steep rise in pressure which can easily result in gas return flow occurring, i.e. a partial regression of the pressure wave. This phenomenon and the decelerating influence of the stroke (impact) losses on the gas flow through raise the prevailing back-pressure in the exhaust manifold.

From this it follows that when considering the usual exhaust pipe construction a considerable back-pressure prevails in both exhaust manifold and in the cylinder combustion space at the beginning of the induction stage when the inlet valve opens. It has been shown by experimentation that this pressure creates a strong flow of the residual gas into the inlet manifold as the inlet valve opens. The quantity of residual gas which flows into the inlet manifold naturally depends on the prevailing pressure difference and also on the length of time between opening of the inlet valve and closing of the exhaust valve.

The residual gas intermixed with the induced charge lowers the combustion characteristics, slows down the rise in pressure and so diminishes engine efficiency and the engine efficiency ratio. The residual gas also noticeably increases the temperature of the induced fuel thus diminishing the thermal efficiency ratio and fundamentally increasing the possibility of engine knock. The effect of the fuel intake temperature on the susceptibility of the engine to knocking is a result of the time lapse before the charge has attained its ignition temperature—the length of induction—which is the combustion stage characteristic to the knocking phenomenon, and is decisively dependent on the fuel octane number and in addition on the prevailing pressure and temperature in the charge at the end of the compression stage. It has also been shown in the graphical results of various published researches (Jost, Teichmann, Tizard and others) that a relatively small drop in temperature noticeably lengthens the induction time and thus fundamentally diminishes the tendency of the engine to knock.

In summating the foregoing it can be concluded that a fundamental improvement in the efficiency ratio of a petrol engine operating under partial load presumes the development of a solution which will lessen the relative quantity of residual gas to be mixed with the induced charge, thereby improving the combustion characteristics and lowering the tendency of the engine to knock.

In order to be able to compare the knocking tendencies of various engine designs with one another a working formula has been drawn up based on the results of research work. When used with conventional exhaust pipe designs the formula gives as a guideline the factor 17.2, at which the engine still operates knock-free. If the compression ratio is higher than normal the quantity of residual gas in the cylinder combustion space is relatively speaking smaller since the volume is smaller and therefore the guideline number can be slightly higher. The same applies to engines with direct fuel injection in which the fuel jet ties up heat energy and thus lowers the temperature of the mixture.

In accordance with the invention the solution is characterised in that by using an exhaust construction the quantity of residual gas to be mixed with the cylinder induction charge is lessened and the knock tendency of a petrol engine is reduced to a level such that the engine values can be classified from the formula

$$\frac{(100+D) . e}{RON} = gn$$

in which D=piston diameter in mm., e=compression ratio, RON=petrol octane number (researched value) and gn=indicated tendency of the engine to knock, these all giving a guideline factor of 18.5. or, if the RON value is 92 or less, the guideline number is higher, when the efficiency ratio of a partially loaded petrol engine is decisively improved. This is in the first instance due to the fact that there is relatively less residual gas to be mixed with the induced charge. This is the result of the lower pressure level prevailing in the gas at the time of inlet valve opening, as a consequence of which less residual gas flows into the inlet manifold and less remains in the combustion space when the exhaust valve closes. The knocking tendency of the engine is then reduced to the extent that in engine applications the compression ratio value and the fuel octane number and also their common ratio can be chosen with a considerably higher guideline number and in terms of a petrol engine the value of the guideline number can be at least 18.5. In practice this means that for current petrol grades the compression ratio can be raised 1—1.5 units from present levels, or by keeping the compression ratio at the current level a move can be made to using petrol grades with an octave rating of 8—10 units lower than presently in use. Naturally a compromise between these alternatives can also be made.

Practical experiments carried out on several test engines have shown that with exhaust systems designed in accordance with the invention the fuel economy of an engine can be noticeably improved even in cases where the compression ratio is not increased. The relative saving in fuel consumption is then the greatest when the engine is partially loaded which is the result of the common influence of various factors. Due to the low temperature of the induced charge the thermal efficiency is better from the outset and because there is less passive residual gas in the cylinder and combustion characteristics are better. The combustion of the mixture as well as the pressure formation are consequently faster, resulting in a beter shaped indicator diagram adds to the quantity of mechanical work done. In consequence of the foregoing, for example when driving in normal traffic conditions, there is a lower average effective pressure level prevailing in the inlet manifold than normally used in a conventional exhaust system, which progresses vapourisation and thus speeds up the combustion process still further. A noticeable practical advantage is the possibility of using petrol grades of 84—86 RON in an engine which with a conventional exhaust system requires the use of a petrol grade of 92—96 octane. This has been proven in tests with several test engines and it opens new possibilities to move to the use of lead free petrol grades.

In the solution afforded by the invention the heat, pressure and kinetic energy in the gas flowing out of the cylinder and into the exhaust manifold after the exhaust valve has opened are utilised by a suitably constructed exhaust system to achieve strong and timed pressure changes so that when the exhaust valve opens a compression peak is created by the gas flow, this pressure peak being immediately succeeded by a strong pressure drop which lasts until the pressure peak created by the following exhaust valve opening. In this way the exhaust gas flow can be accelerated into the exhaust manifold and from then on into the exhaust system piping, from both that cylinder the exhaust valve of which has created the pressure and also that cylinder in which during the successive pressure drops both inlet and exhaust valves are continuously open. As the result of tests it has been confirmed that by using the solution according to this invention, a notably lower pressure level exists in the exhaust manifold between the pressure peaks created by the exhaust valves opening than is the case with other exhaust manifold constructions.

To realise the process according to the invention the exhaust manifold is connected by means of a pipe of suitable length to a case the construction and volume of which are such that the gas flow accelerated by the pressure peak causes the least possible pressure loss within its confines and which when a fast pulsating flow is in question sufficiently slows down the return flow of the gas. An essential part of the construction is a swirl chamber into which the flow channel is led and which because of its location and construction creates a strong swirling flow, this noticeably slowing down the return flow of the gas. The formation of the swirl flow can be additionally strengthened by the use of vanes which guide the flow in a mainly tangential direction and which because of their shape and location add to the slowing down effect of the construction on the gas flow.

According to the invention is an exhaust pipe system to additionally operate in cases where there is a pulsating gas flow so that no stroke (impact) losses arising from pressure wave reverberation occur, nor rise in pressure occurring in the silencer as a result of gas return flow.

This is based on the flow channel construction of the connection pipe leading to the swirl chamber of the silencer and additionally that the gas which has flowed through the chamber takes part in the high speed swirling action. The gas coming from the connecting pipe thus flows on the cross-sectional surface in a widening channel the expansion leading trom this suitably adding to the gas flow velocity. The construction of the channel also causes ring-like gas flow through its cross-section, the outer dimension of which increases to the same value as the outside swirl flow diameter of the chamber, this taking place in a plane at right angles to the direction of progression, the outer diameter of the swirl chamber being suitably double that of the connecting pipe and of circular section. The direction of flow changes at this stage under the influence of the guide vanes so that as it flows in the prevailing swirl flow of the chamber the average emergent angle is about 40°.

When pressure waves arrive continuously the swirl flow that they maintain is also continuous and the speed of the participating gas is about the same as the average speed of the pulsating gas flow entering it, so that the flow speed only momentarily diminishes, in other words when the gas flow diminishes the swirl flow speed. The lessening of speed thus caused and also the change of direction compensates however for the increase in speed previously caused by swirling so that only an insignificant part of the kinetic energy in the gas is transformed into pressure energy as it flows from the connection pipe into the swirl chamber. As a consequence of this no reverberation occurs in the pressure waves, neither are pressure shocks nor sound effects to be observed. A fundamental concept is that the ring-like gas flow is directed under the influence of several equispaced guide vanes into a swirling flow. These vanes, because of their shape and location, prevent the gas from penetrating into the channels between them so that no return flow can occur in the short time which elapses between the arrival of one pressure wave and the next. The solution acts so efficiently in this respect that the cubic capacity of the silencer can be selected as two or three times that of the cylinder volume before a pressure rise caused by the relatively small volume would cause any significant flow return. It has been confirmed experimentally that if the swirl chamber volume is suitably enlargened the silencer case can be, if so desired, located in close proximity to the exhaust manifold before the operation of the manifold exhaust pipes is significantly affected. This enables the utilisation of the solution according to the invention of cleansing catalytic reactor for the exhaust gas. When the requirement for an efficiently operating catalytic reactor is a sufficiently high exhaust gas temperature the reactor is located at the end of a connecting pipe which is only 0.5 to 0.8 m long.

Because of this and the construction of the catalytic reactor the back pressure in the exhaust manifold is noticeably higher than usual, this deteriorating the performance of the engine and adding markedly to the engine tendency to knock. Because of this latter a lower than normal compression ratio is used in the engine and this itself also lowers fuel economy. By locating a silencer constructed according to this invention between the catalytic reactor and the exhaust manifold, or by constructing a pre-chamber according to this invention into which the gas flows from the connecting pipe and then on into the actual catalytic reactor, the average back pressure in the exhaust manifold is noticeably lowered.

As illustration of this invention the accompanying drawing Figure 1, represents a silencer in longitudinal cross-section and Figure 2 depicts the location of the vanes which improve the swirl flow.

The constructional parts are numbered as follows:

1 entry duct. 2 conically-shaped flow channel between casing and inner cone piece, 3 annular flow channel between the inner conically ended nozzle piece and the silencer casing, 4 casing, 5 swirl chamber, 6 perforations, 7 separating wall, 8 nozzle cone, 9 channel between inner cylindrical part of nozzle piece and flow exit pipe, 10 cylindrical rear part of nozzle piece, 11 expansion space, 12 flow exit pipe, 13 nozzle vanes, and 14 support piece between nozzle piece and flow exit pipe.

The silencer operates as follows: gas flows from the entry duct 1 at high speed into a conically-shaped divergent channel 2 the annular sections of which increases in the direction of flow so as to bring about a slight expansion i.e. a part of the gas pressure is converted here into kinetic energy. The gas continues its flow in constant section annular channel 3 and meets the vanes 13 located between the casing 4 and the rear end of the nozzle piece 10, these vanes changing the flow direction so that a strong tangential swirl flow is set up in the chamber 5. The simultaneous pressure increase taking place in the swirl chamber accelerates the gas flow by way of the flow channel 9 and the perforated pipe 6 to the flow exit pipe 12 and then on to the continuation of the exhaust pipe.

Because of the pulsating nature of the gas flow when the construction according to this invention is used, a strong drop in pressure result immediately in the exhaust manifold and the connection pipe, the pressure difference thus created having a tendency to cause return flow of the gas via channels 3, 2 and 1 from the swirl chamber 5 and thence on into the exhaust manifold. The direction of swirl flow in the chamber 5 and the position of the vanes 13 which created the swirl flow constitute however a relatively high resistance to this return flow, this also being increased by the dimension of the channel which causes a slight compression in gas flowing in that direction with a consequent partial conversion of the kinetic energy of the gas into pressure energy and a slowing down of the gas flow. The silencer thus operates strongly in cases where there is a pulsating gas flow and when the duration of different stages is in the order of milliseconds thus resembling a flow return valve, causing a flow resistance that is fundamentally dependent on the direction of gas flow.

It is characteristic to the construction that the form of the vanes 13 is so selected that the change in direction of gas flow increase evenly whereupon the secondary swirl flow remains minimal. This is achieved by using a curvature of the vanes 13 corresponding to the graphical curve of which there is a standard second derivation.

It is obvious that the invention is not limited to the aforementioned examples of application and that many other constructions can be put forward according to the following patent claims.

Claims

1. A method of increasing the performance and efficiency ratio of an internal combustion engine by improving the gas flow therethrough by means of a pipe connected to an exhaust manifold and its continuation to one or more successive silencer constructions, in which the quantity of residual gas to be mixed with the induction charge into a cylinder of the engine is reduced, thus lowering the temperature and reducing the engine tendency to knock, by constructing the exhaust system such that when the engine values are placed in the formula

$$\frac{(100+D)\,.\,e}{RON} = gn$$

in which D=piston diameter in mm, e=compression ratio, RON=the petrol octane number (researched value), and gn=a guideline number indicating the engine tendency to knock, a guideline value of 18.5 or larger is given with a combustion fuel that researched octane number of which is 92 or lower.

2. A method according to claim 1 in which when the cylinder exhaust valve opens into the exhaust manifold the pressure and kinetic energy of the discharged gas are utilised by suitable exhaust pipe system construction to create intensive and thus timed pressure changes in the exhaust pipe system so that when the exhaust valve opens the pressure peak in the exhaust manifold caused by the flowing gas is immediately followed by an intense pressure drop that lasts until the pressure peak caused by the next opening of the exhaust valve.

3. A method according to claim 1 or claim 2, in which when the exhaust valve is open the construction of the exhaust pipe system is such that the system creates such a pressure that the flow of residual gases into the inlet manifold is essentially inhibited when the inlet valve opens, the temperature thus being lowered and increasing the relative portion of the new mixture in the cylinder induction charge.

4. A method according to any one of claims 1 to 3, in which the exhaust gas flow is caused to expand over an increasing annular surface so that its velocity is suitably increased and then directed into a circular motion taking place normal to the axial flow of the gas.

5. An exhaust pipe system for carrying out the method according to any one of claims 1 to 4, in which an exhaust manifold is connected to a silencer by a connection pipe of suitable length, the silencer being of such construction and volume that the gas flow accelerated into it by the pressure peak results in as small as stroke (an impact) loss within its confines as possible, which creates a strong swirl flow in its chamber and

which fundamentally slows down the return flow of the gas which is part of the swirl flow.

6. An exhaust system according to claim 5, in which in the silencer case is an entry duct furnished with an extended expansion space which is fitted with swirl-forming elements and opening into a swirl chamber from which the gases pass on into a flow exit pipe.

7. An exhaust system according to claim 6, in which in the expansion space a swirling gas flow is set up under the influence of the swirl-forming elements which results in an annular swirl flow over its whole length.

8. An exhaust system according to claim 6 or 7, in which behind the entry duct is an internal nozzle cone, the space between it and the silencer case expanding annularly at least from the duct end and the cylindrical rear portion of which has, on its surface or at its position, vanes arranged at suitably equidistant spacing to create a swirling action, their pitch, shape and attitude relative to the swirl flow being such that the return flow of the gas is essentially inhibited during the interval time in which there is a higher pressure prevailing in the swirl chamber than in the entry duct.

9. An exhaust system according to claim 8, in which the swirl-forming vanes are set at an angle to the axial flow of the gas, the slowing effect of which as the gas moves into its swirling pattern corresponds to the increased flow speed in the expansion space.

10. An exhaust system according to claim 8 or claim 9, in which the shape of the swirl-forming vanes is such that the change in flow direction they bring about is evenly accelerated so that the secondary swirling formation remains a minimum.

11. An exhaust system according to any one of claims 8 to 10, in which the conical nozzle is open to the rear and an exit flow pipe is open from the front end, is perforated with holes at intervals and extends beyond its wall.

12. An exhaust system according to any one of claim 5 to 11, in which the silencer is located in front of a catalytic reactor, the function of which is for cleansing the exhaust gas, or it is constructed with a pre-chamber and situated in the proximity of the exhaust manifold.

13. An exhaust system according to claim 12 dimensioned such that the silencer is operatively situated at a distance in the range of from 0.3 to 0.8 metres from the exhaust manifold.

14. A method of increasing the performance and efficency ratio of an internal combustion engine, substantially as hereinbefore described with reference to the accompanying drawings.

15. An exhaust pipe system substantially as hereinbefore described with reference to the accompanying drawings.

Printed for Her Majesty's Stationery Office by the Courier Press, Leamington Spa, 1983. Published by the Patent Office, 25 Southampton Buildings, London, WC2A 1AY, from which copies may be obtained