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(54) EFFICIENCY AND EMISSIONS IMPROVEMENTS FOR NATURAL GAS CONVERSIONS OF EMD 2-CYCLE MEDIUM SPEED ENGINES

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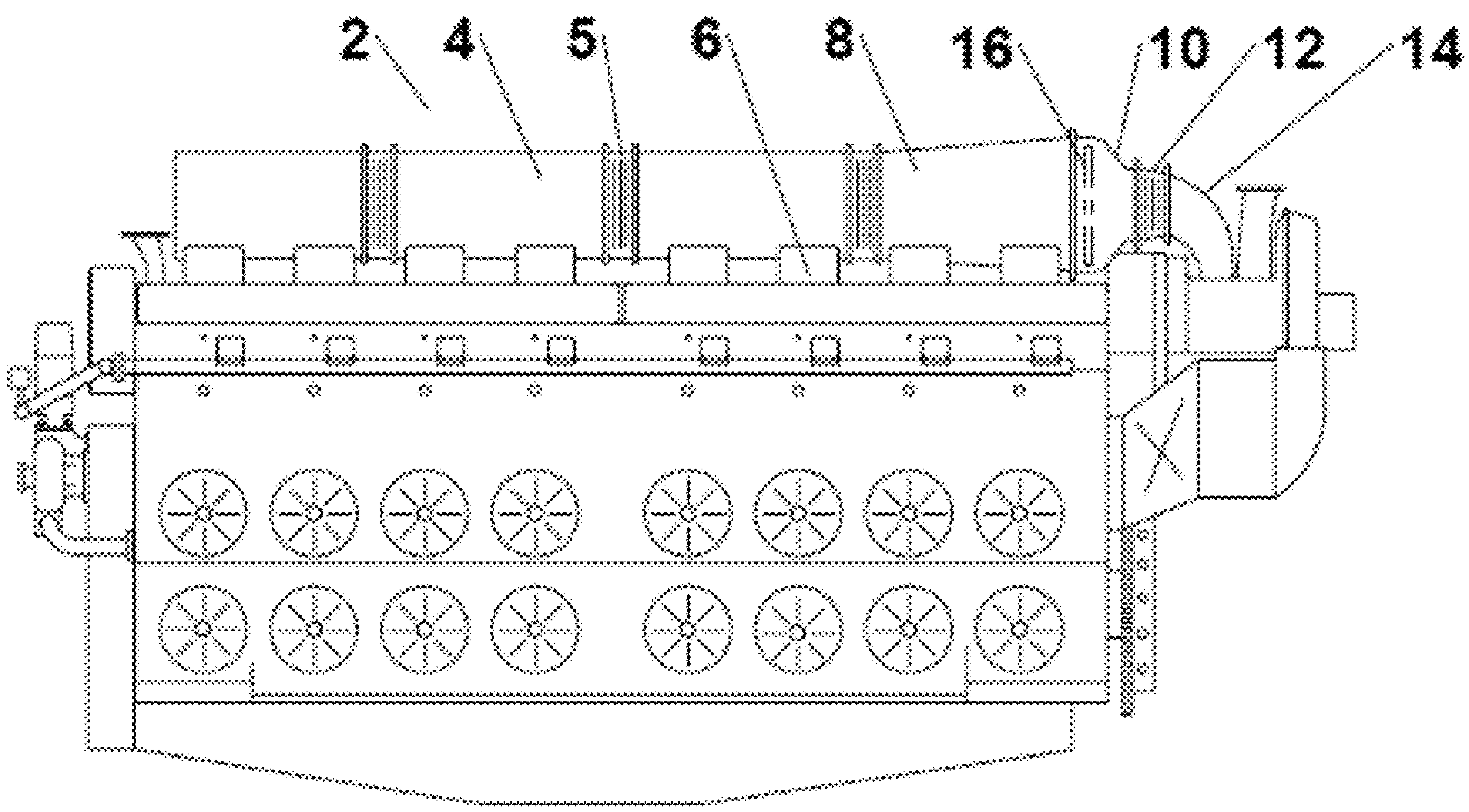
(57) **ABSTRACT**

A single substrate oxidation catalyst system including: a turbocharger adapter exhaust collector segment having a first end and a second end; a debris screen housing in fluid communication with the second end of the turbocharger adapter exhaust collector segment; and an oxidation catalyst substrate located in the in the second end of the turbocharger adapter exhaust collector segment, wherein the oxidation catalyst substrate slides into and out of position in the second end of the turbocharger adapter exhaust collector segment.



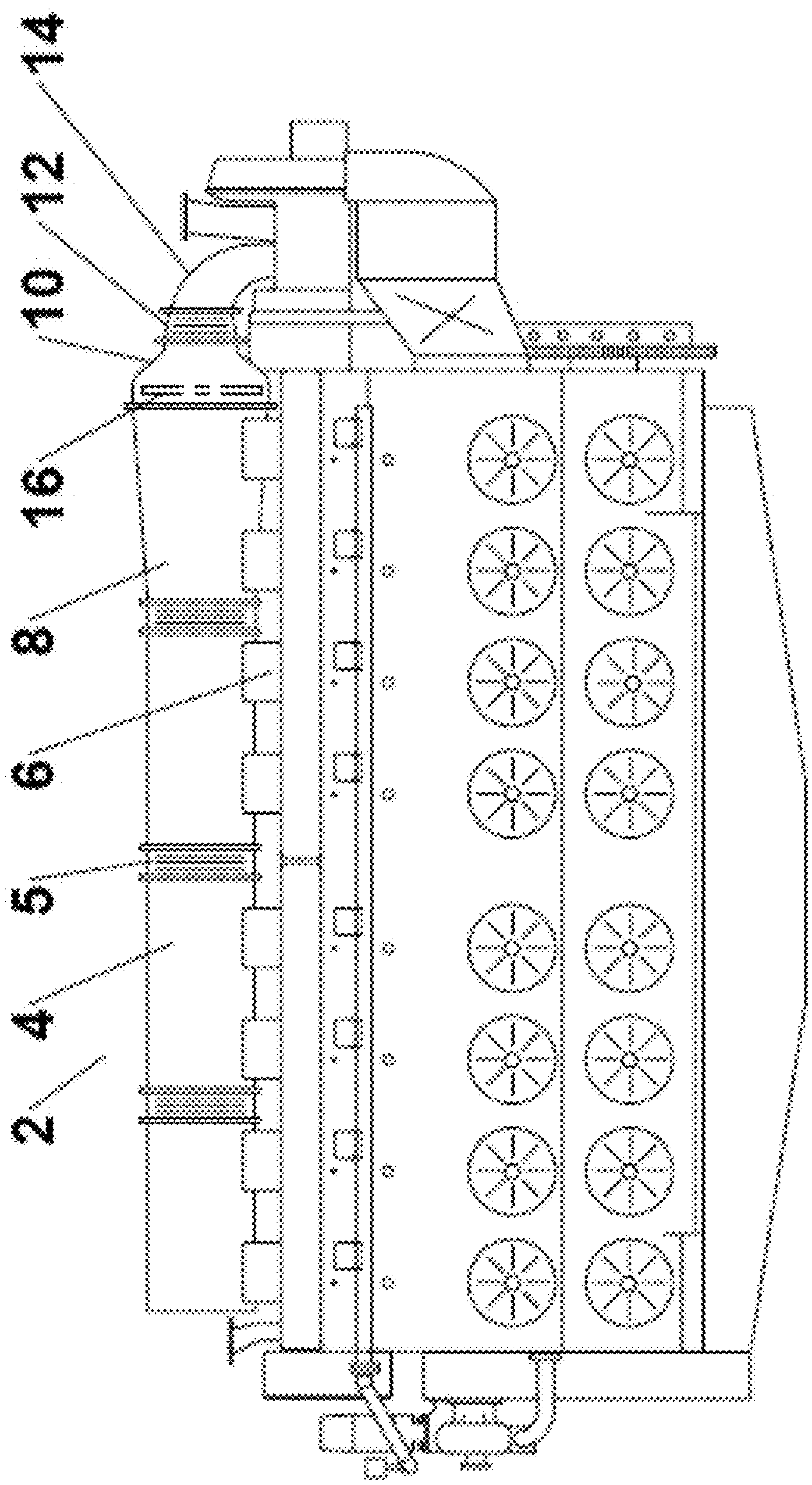


FIG. 1

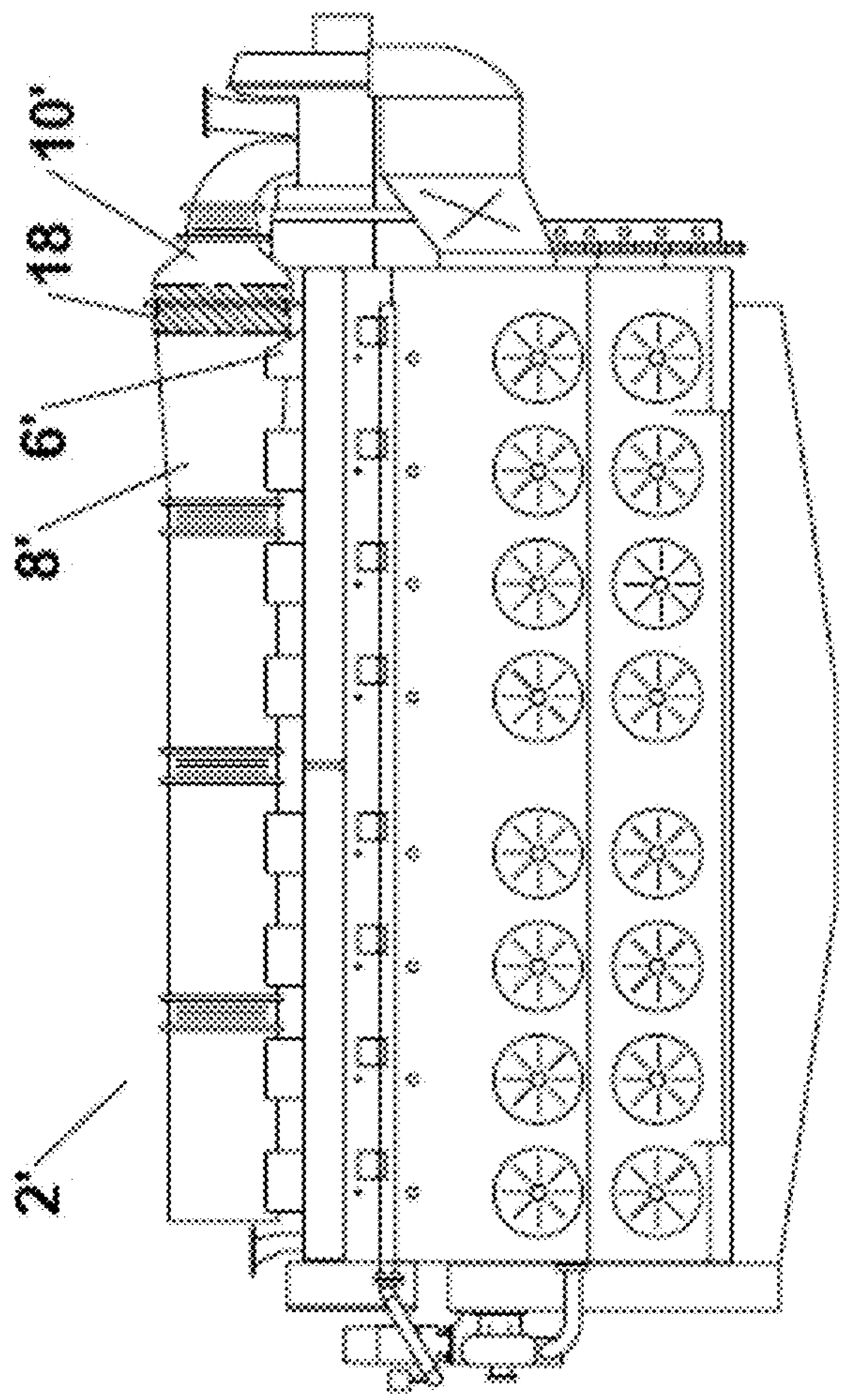


FIG. 2

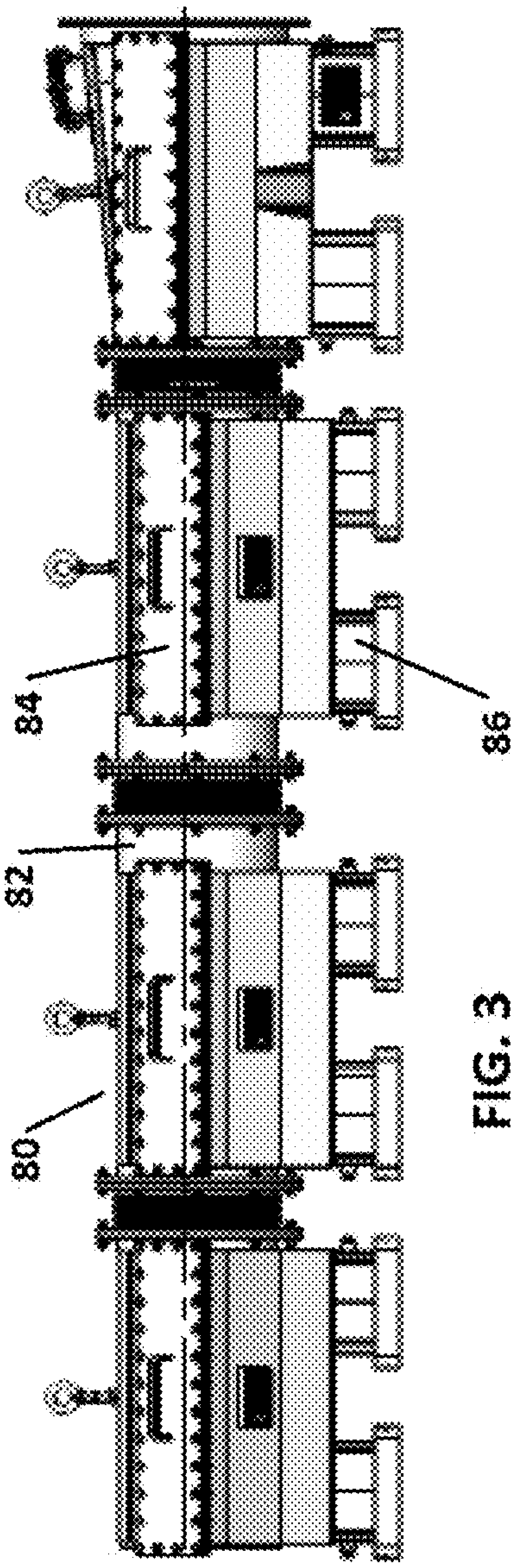


FIG. 3

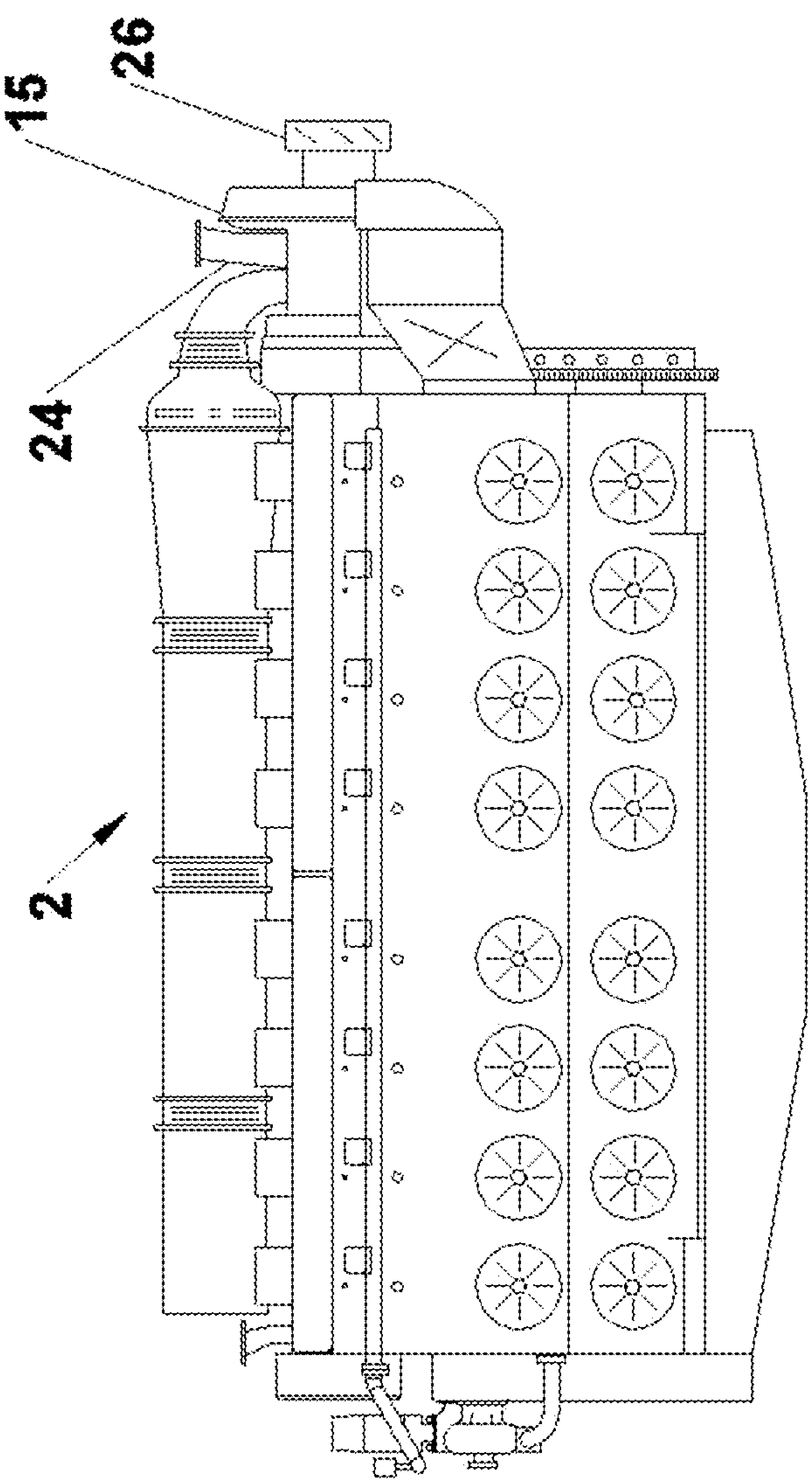


FIG. 4

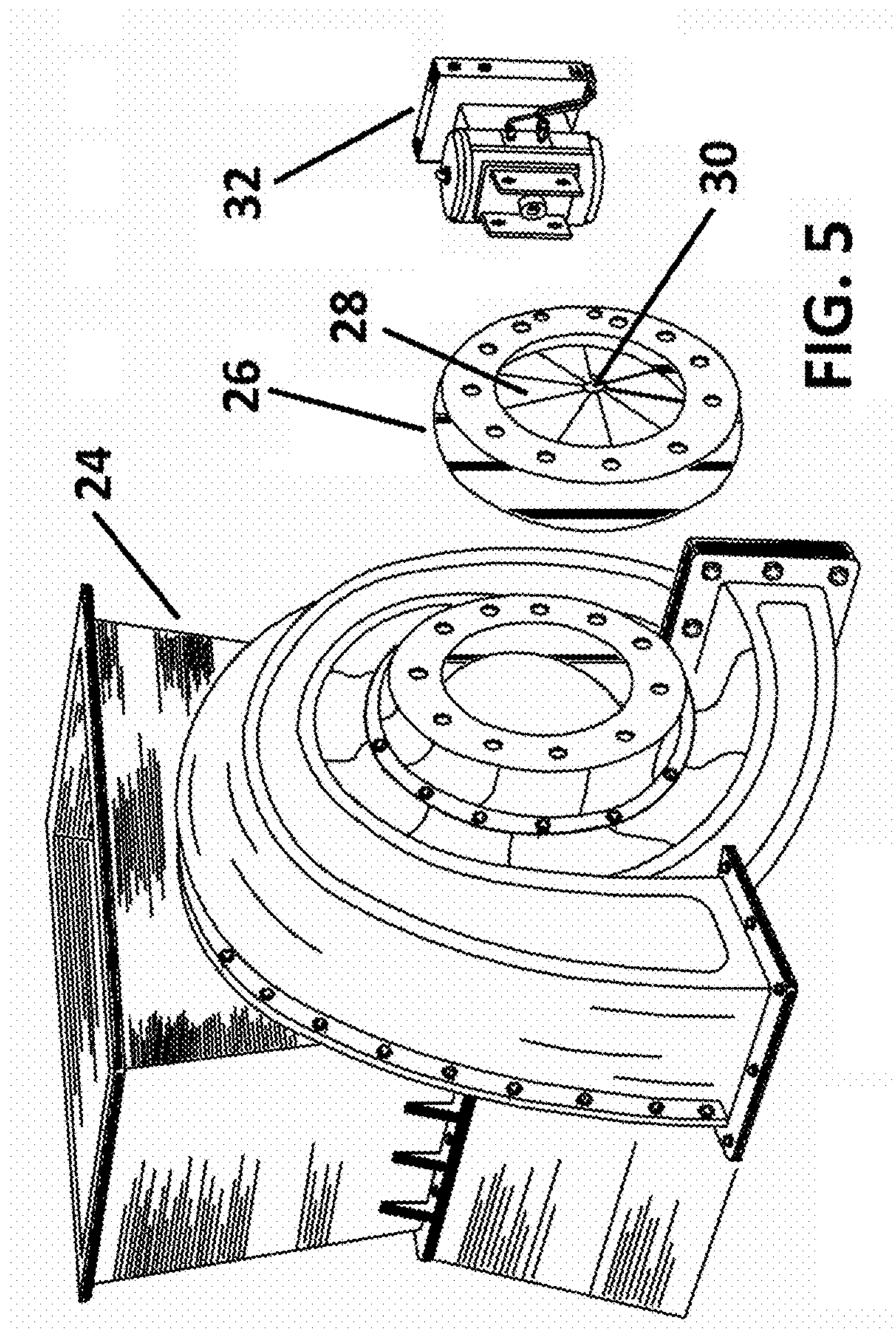


FIG. 5

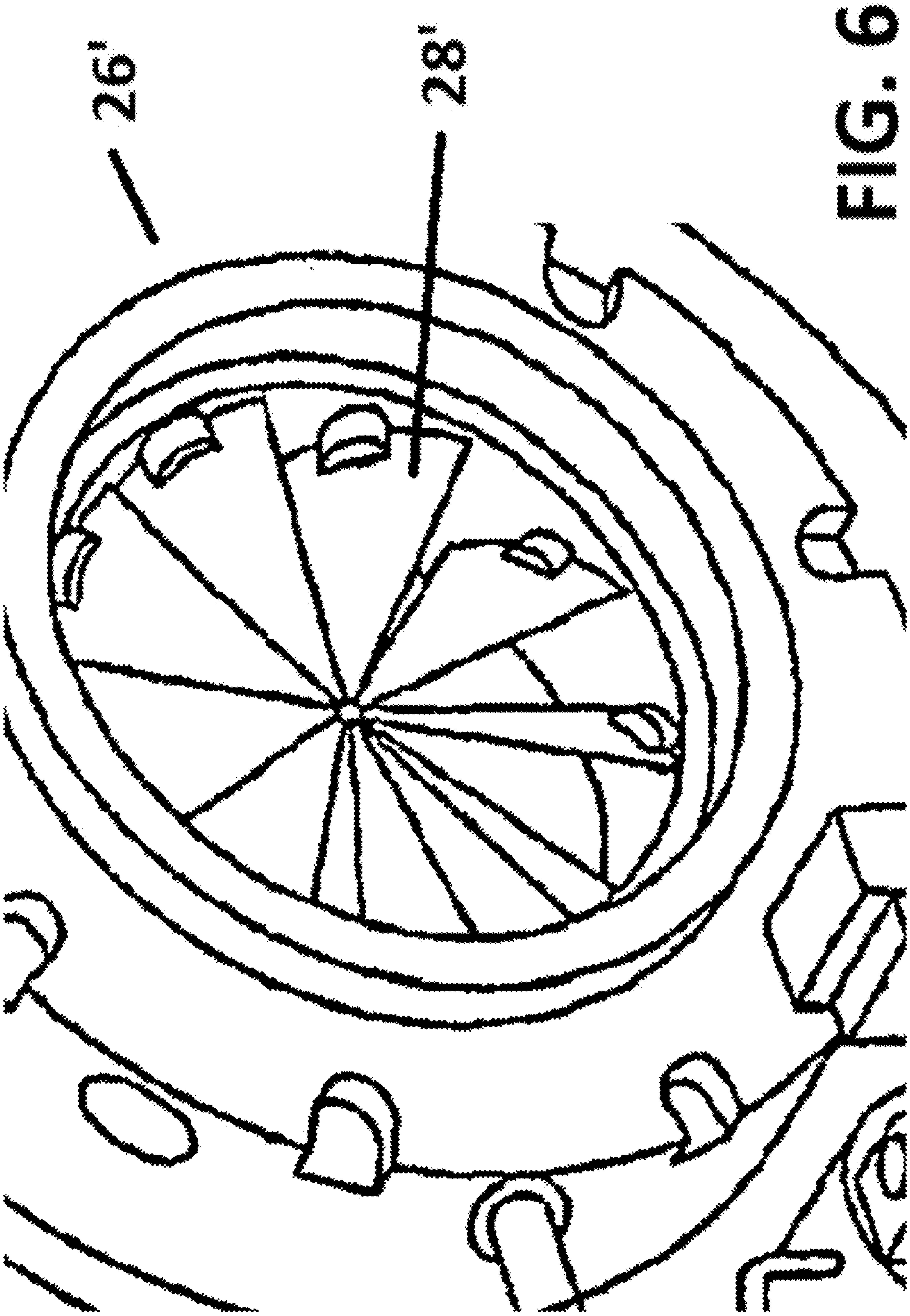


FIG. 6

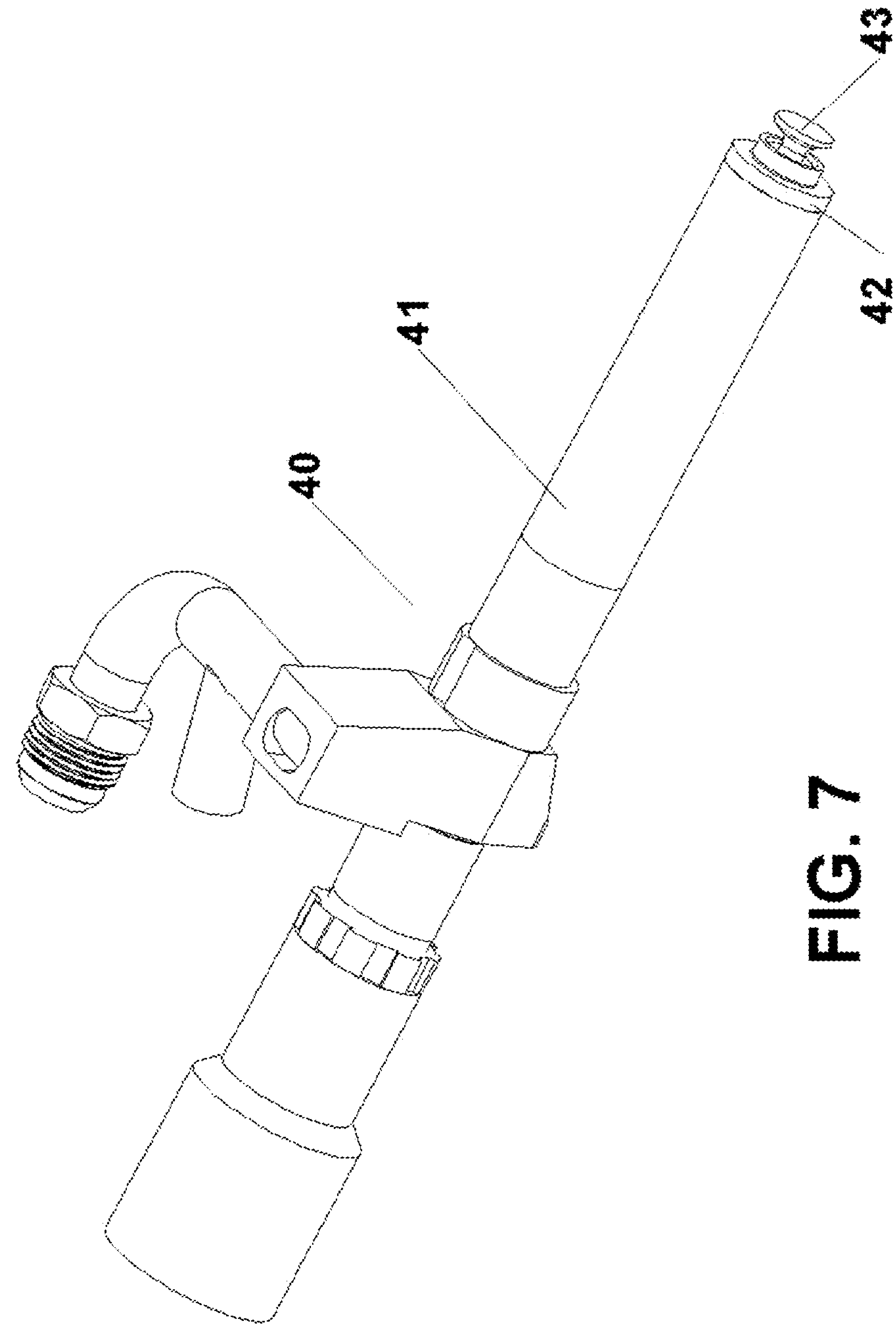


FIG. 7

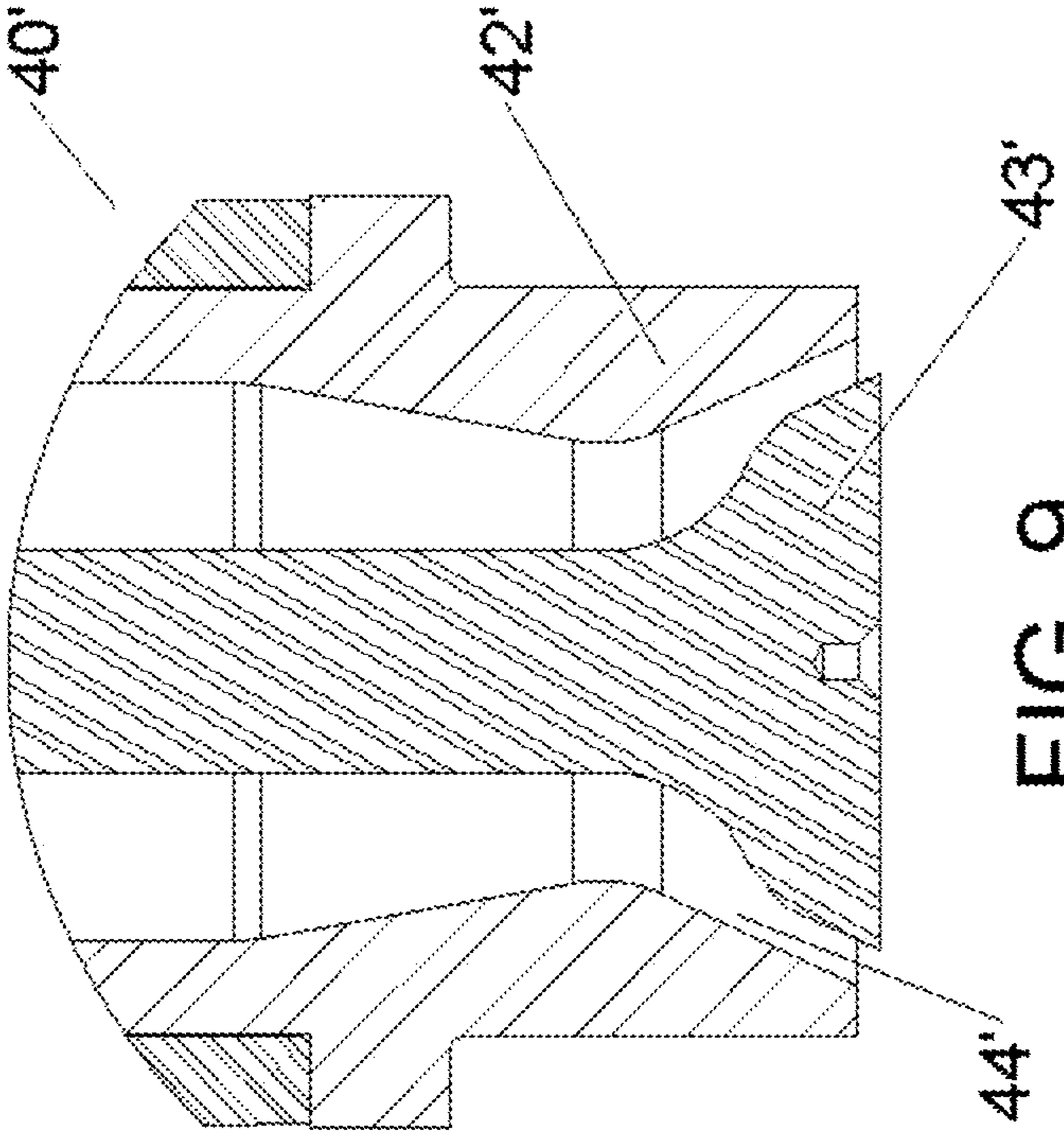


FIG. 9

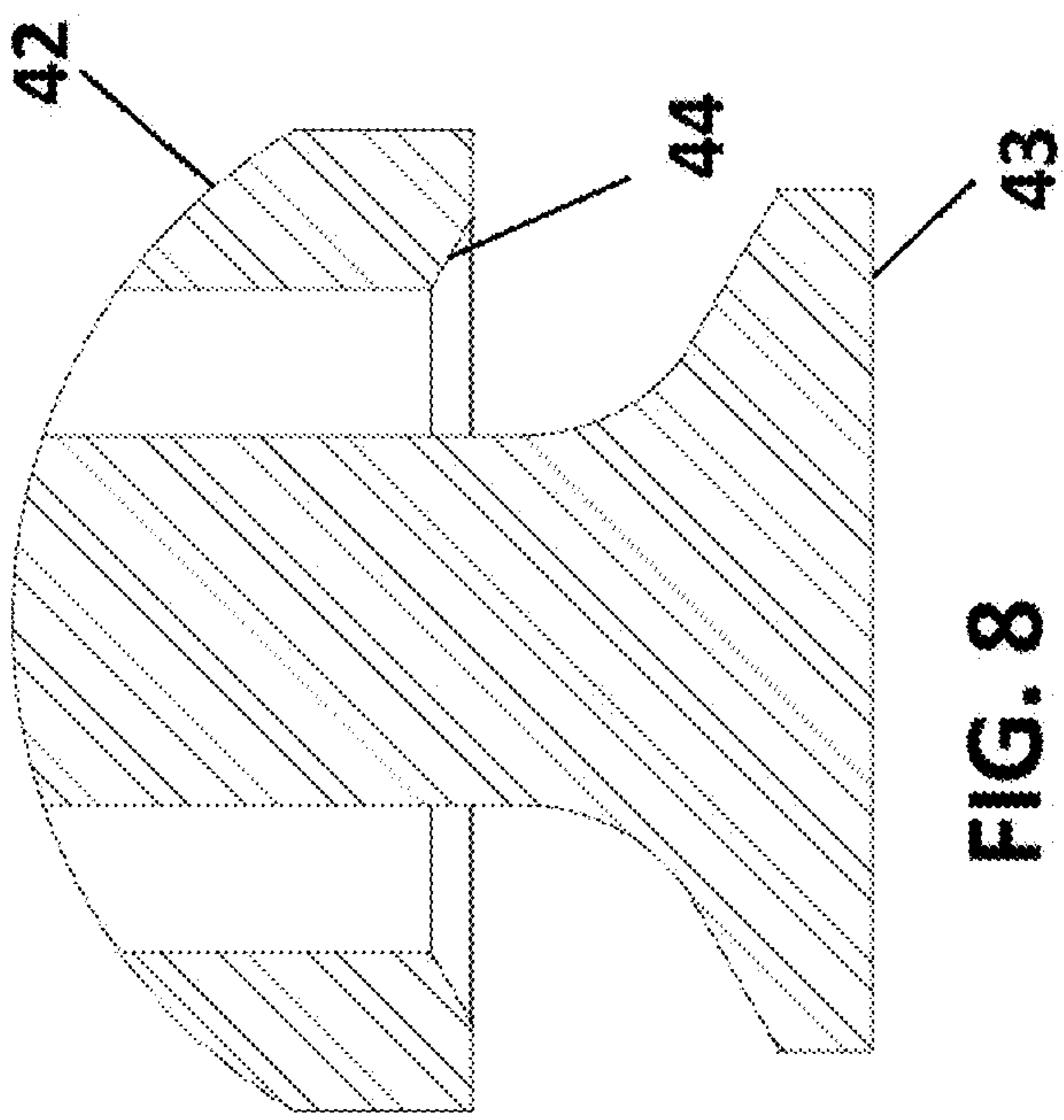


FIG. 8

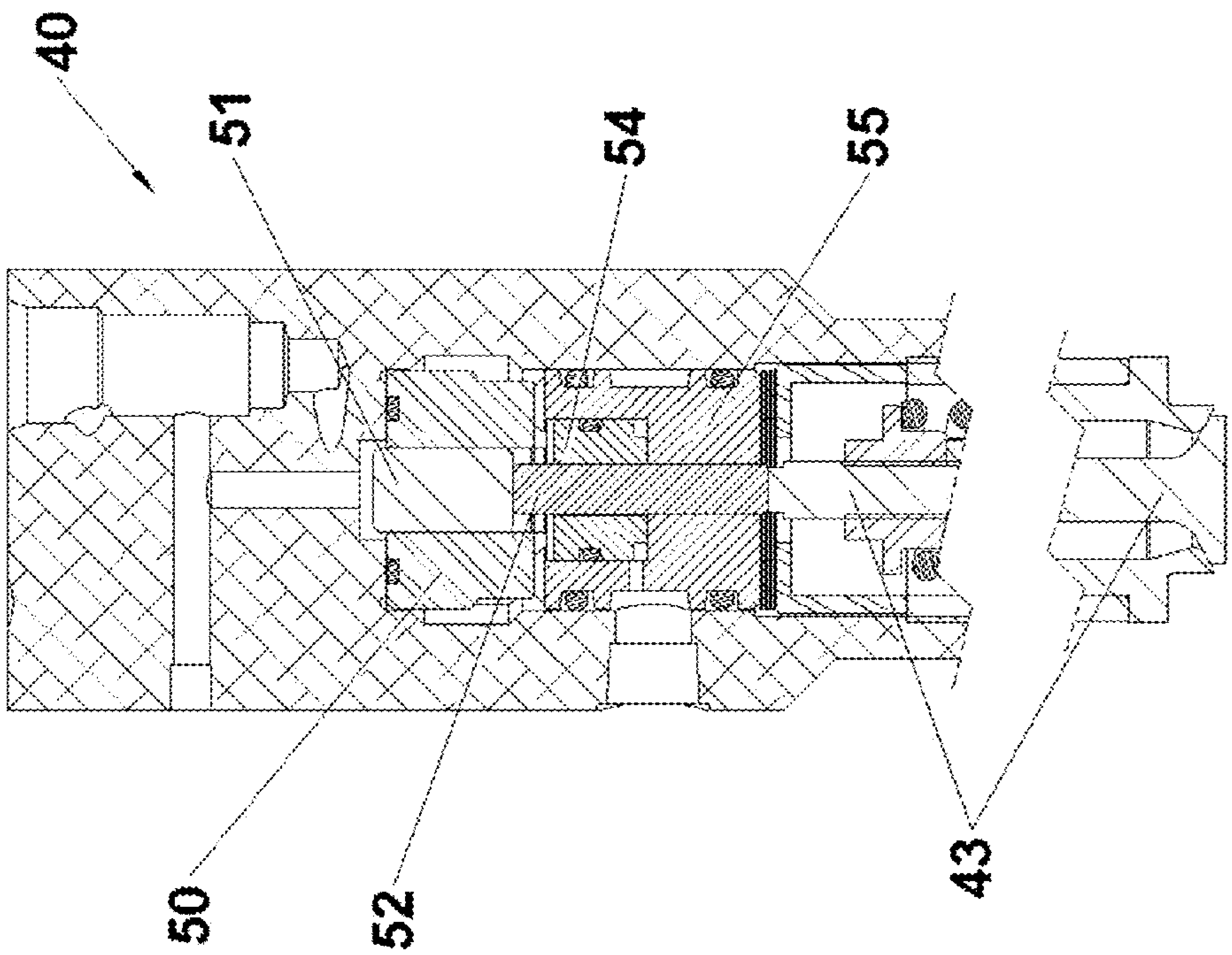
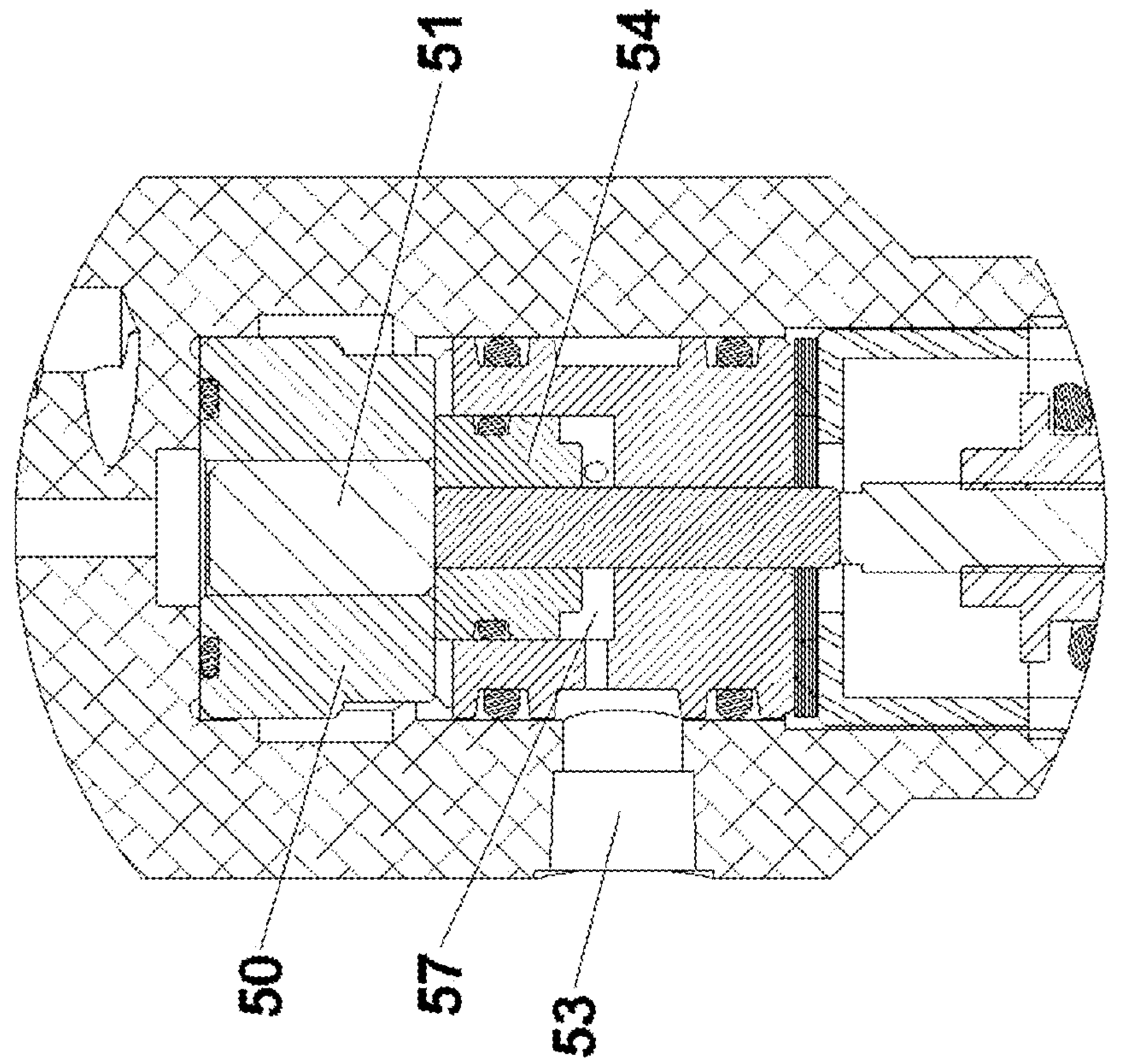
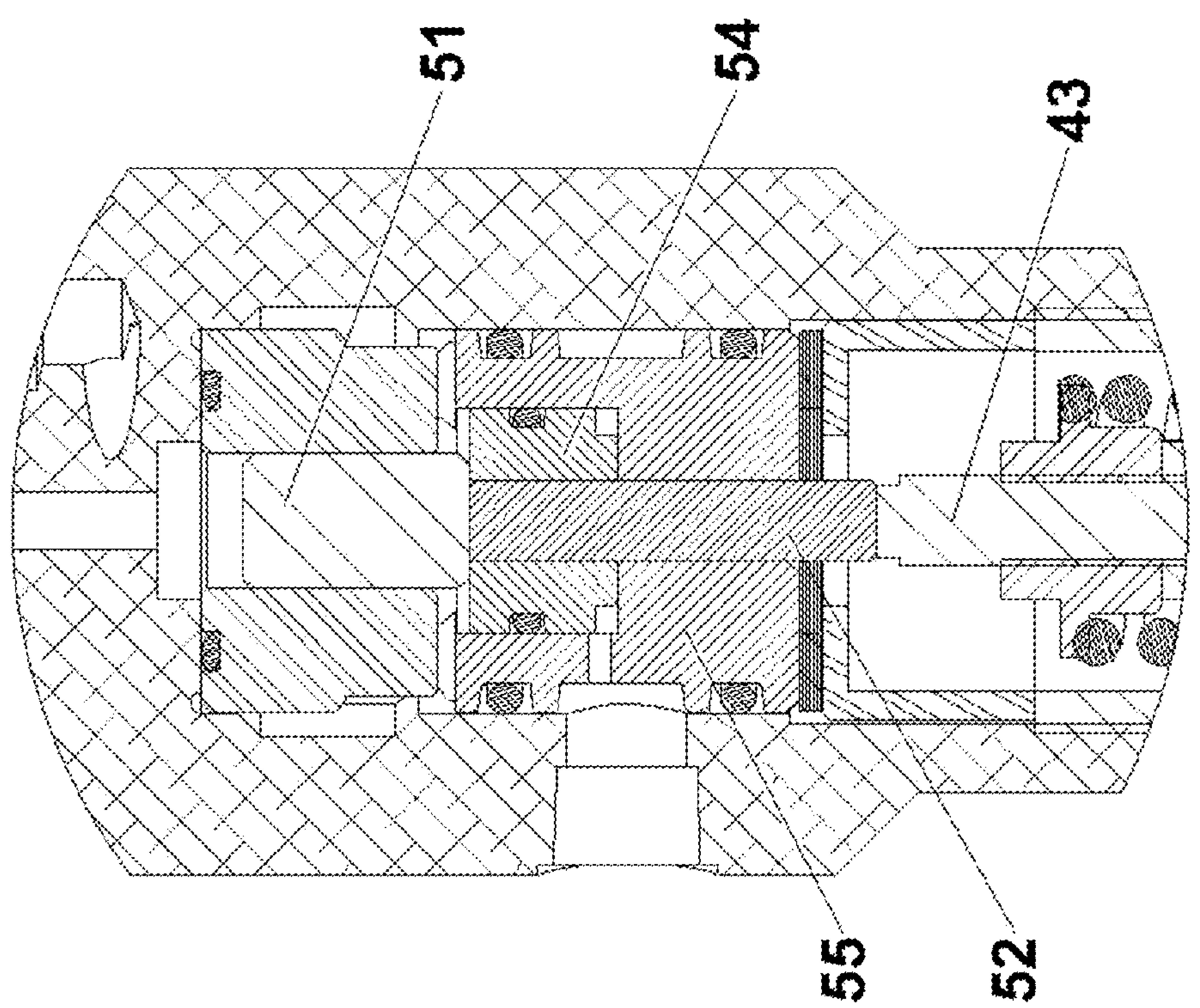


FIG. 10A



Valve Partially Open
FIG. 10C



Valve Full Open
FIG. 10B

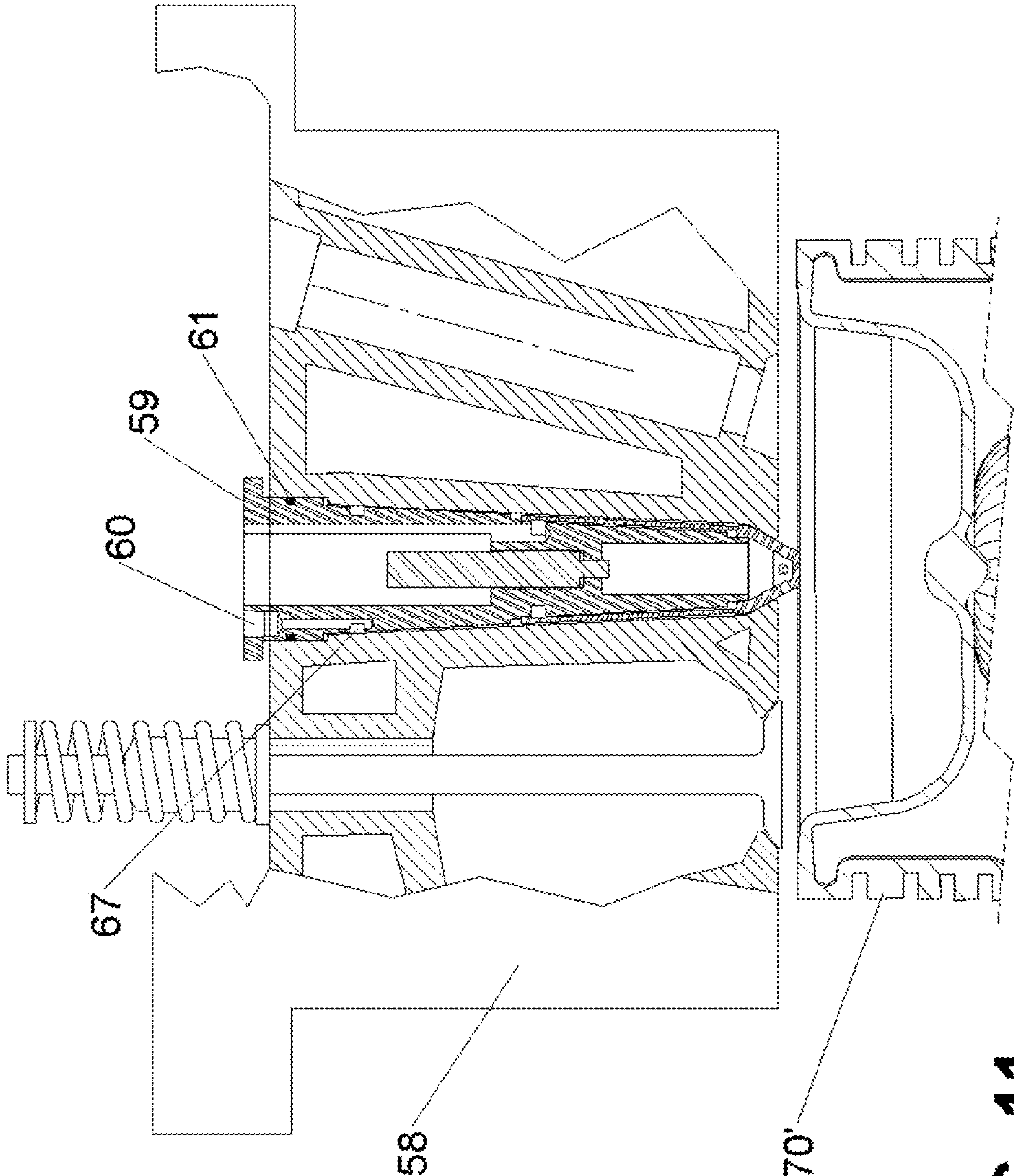
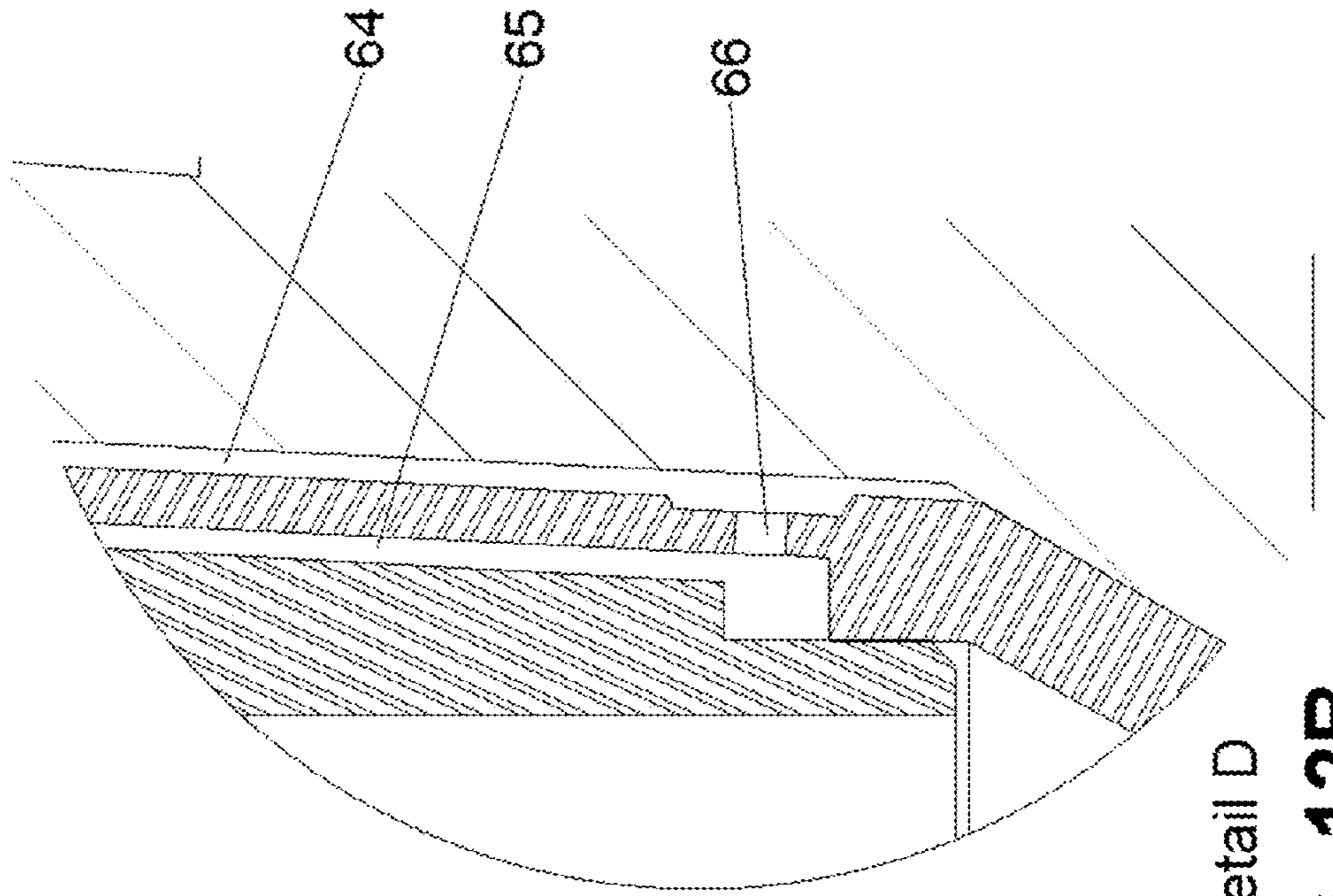


FIG. 11



Detail D

FIG. 12B

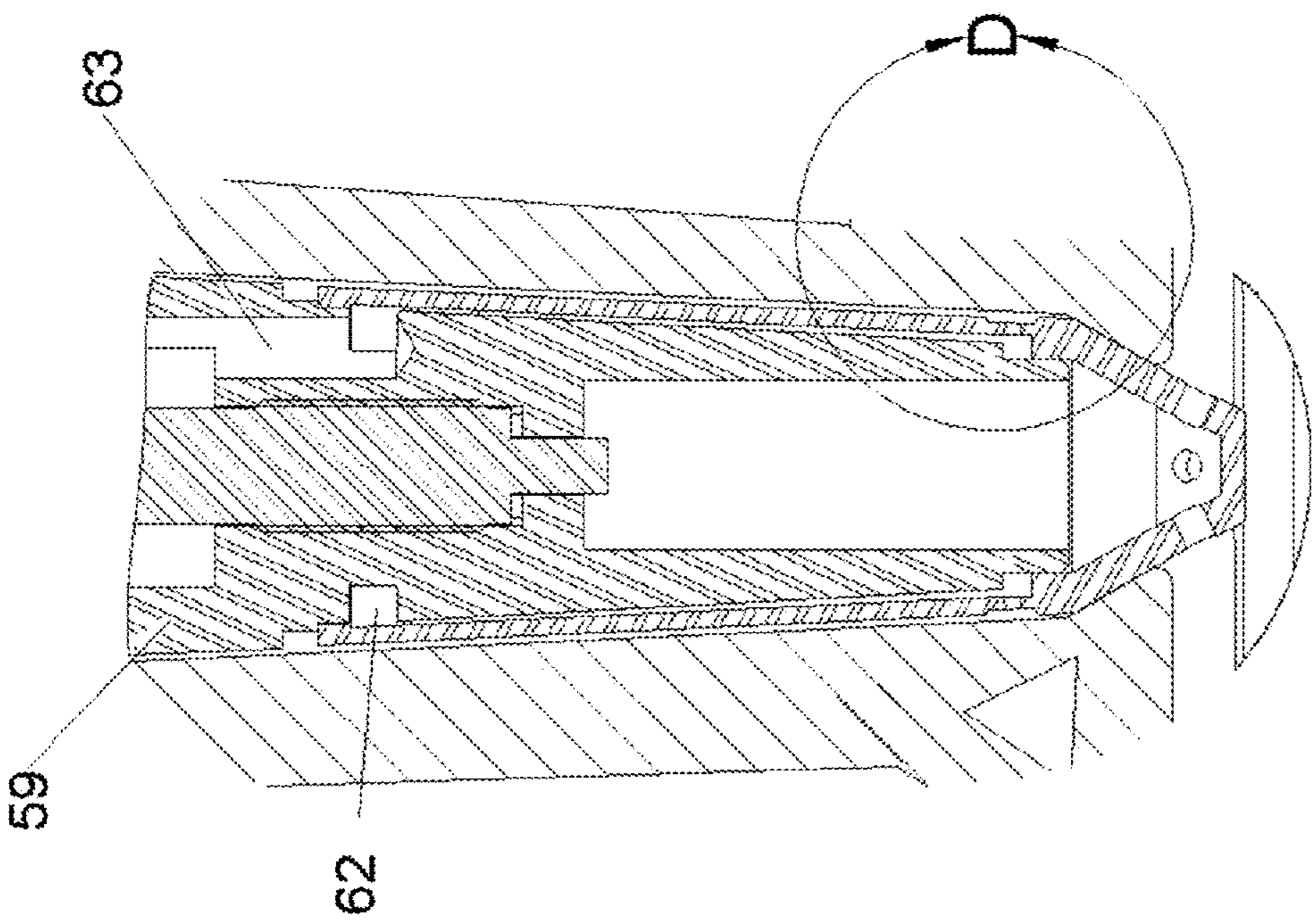


FIG. 12A

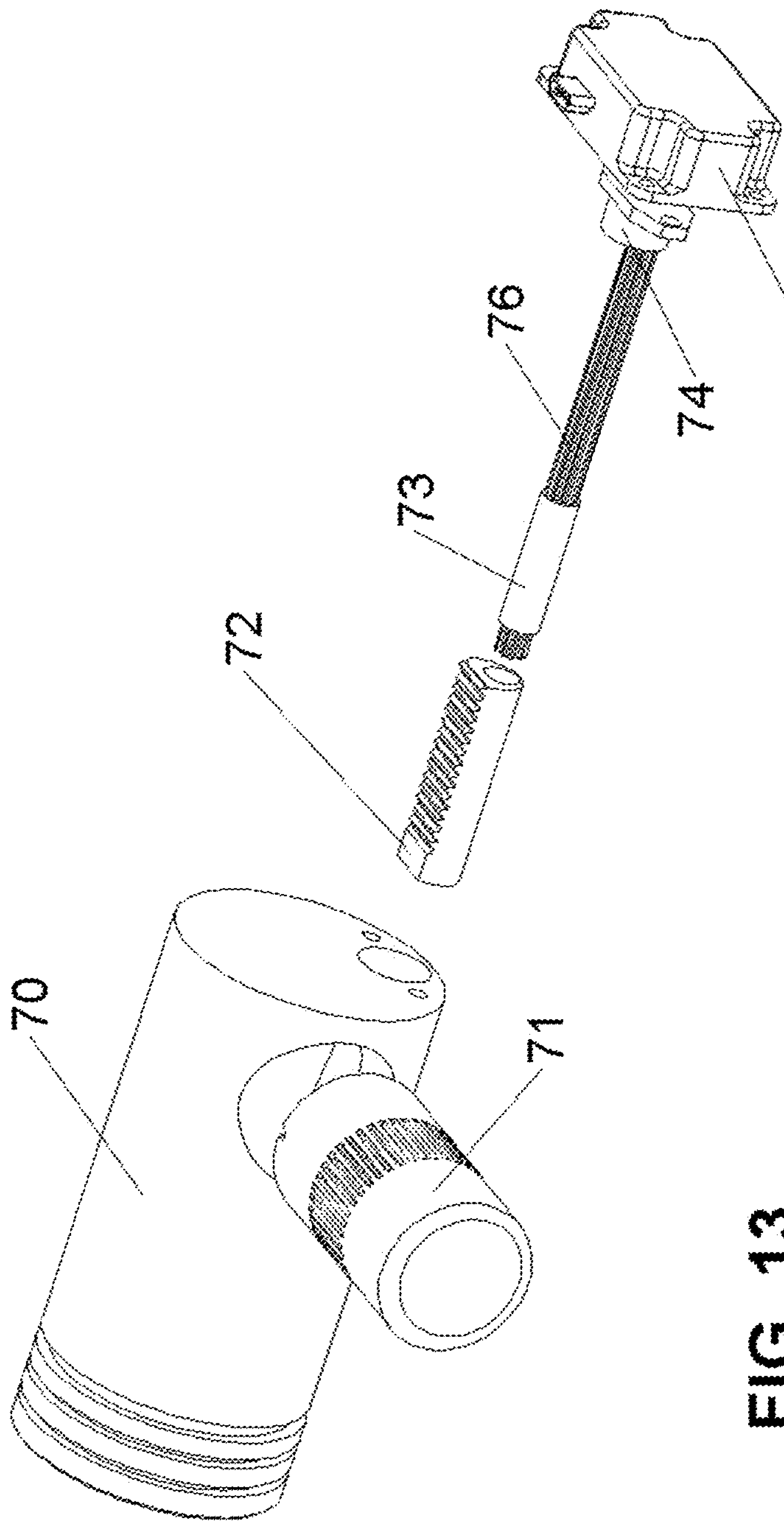


FIG. 13

EFFICIENCY AND EMISSIONS IMPROVEMENTS FOR NATURAL GAS CONVERSIONS OF EMD 2-CYCLE MEDIUM SPEED ENGINES

CROSS-REFERENCE TO RELATED APPLICATIONS

[0001] This application claims the benefit of U.S. Provisional Application No. 61/790,771 filed on Mar. 15, 2013, the entirety of which is incorporated herein by reference.

BACKGROUND OF THE INVENTION

[0002] The first portion of the background is most closely related to a single element preturbine oxidation catalyst element for EMD turbocharged engines using twisted exhaust runners. Oxidation catalysts (OC) are used to reduce the emissions of unburned hydrocarbons (HC), carbon monoxide (CO) and certain types of particulate matter (PM). Of the aftertreatment systems used on lean burn engines, this is the simplest system as it is completely passive and practically maintenance free. In the art these are also commonly referred to as diesel oxidizing catalysts (DOC). When dealing with both diesel engines and gas engines the shortened term oxidation catalyst (OC) is the more appropriate term.

[0003] A very common use of an OC has been in the aftertreatment of diesel truck engines where the OC is placed downstream of the turbocharger, and upstream of the diesel particulate filter (DPF). Because locomotives have become so tightly packaged, there is minimal room for a downstream OC in the locomotive application. A solution to this called a V-Cat has been patented and developed by Miratech. With a V-Cat system the OC is built into the exhaust manifolds on the engine upstream of the turbocharger, hence a pre turbine OC. For packaging reasons, this system had a single OC substrate for each cylinder.

[0004] ASME Paper JRCICE2007-40060 titled 'Exhaust Emissions From a 2,850 kW EMD SD60M Locomotive Equipped With a Diesel Oxidation Catalyst' describes the application and testing of prototype V-Cat system on a 16 cylinder EMD engine in an SD60M locomotive.

[0005] The primary parameter that determines the emissions reduction efficiency of an OC is its temperature. Results of the V-Cat testing in the ASME paper indicate the CO reduction efficiency reaches 90% at around 200 C, the peak HC reduction efficiency is approximately 50% at 320 C.

[0006] Not only does the preturbine placement of this OC system on the turbocharged EMD engine offer a solution to the packaging problem, preturbine placement of the OC has several other benefits.

[0007] As the efficiency of the OC is affected by temperature, its pre turbine placement will substantially increase its overall operating temperature. Notch 4 has a temperature drop across the turbo of 41 C. Notch 4 is when the preturbine temperature finally reaches 310 C where the OC starts to reduce HC at 50% efficiency. If the OC was downstream of the turbo in this case, it would be operating at the mid 40% range. At notch 8, the temperature difference across the turbo is 137 C.

[0008] These increases in preturbine temperature will be even more important in the future when methane becomes a regulated emission. Typical OC systems do not efficiently remove methane until 400 C. With a preturbine OC system this would start at notch 4, with a downstream OC this may not start until notch 7. When hydrocarbon emissions are regulated for natural gas engines, they regulate only non-methane hydrocarbons. In natural gas engines the methane component of total HC is typically close to 90%. Although methane is not a criteria pollutant with direct human health risks, air agencies are paying more attention to methane emissions as a potent greenhouse gas with regulations for it pending in the near future.

[0009] One interesting finding in the ASME report is that the preturbine OC actually increased the turbo inlet temperature and overall engine efficiency because of the energy released when it oxidized the CO, HC and PM matter. This led to an actual increase of engine thermal efficiency at some points even though the additional back pressure of the OC would typically cause a decrease in efficiency due to reduced air flow.

[0010] Another advantage of a preturbine OC over a downstream OC is the effect of the OC system back pressure. Downstream of the turbo, whatever back pressure the OC causes would be multiplied by the pressure ratio of the turbo turbine. So if the OC caused a pressure drop of 1.1 kPa and the turbo had a pressure ratio of 2.7, the back pressure increase in the exhaust manifold would be 2.9 kPa. In the case of the preturbine DOC, the OC pressure drop is not multiplied.

[0011] As noted, the V-Cat system tested in the ASME paper had a single OC substrate per cylinder. This system is now in production and sold exclusively by EMD the manufacturer of the EMD engines. Because of the way the exhaust manifold segments are tightly packaged across the top of the engine and there is only a short 4 inch length of common exhaust plenum before the turbo charger inlet plumbing, there was no easy way to package a single large substrate that all of the exhaust would flow through equally. The solution was to have a single OC substrate for each cylinder and therefore each cylinder would experience the same pressure drop and the engine would run smoothly. If a single large substrate was attempted and each cylinder was affected differently, performance would suffer as some cylinders would get more intake air than others.

[0012] In the report the pressure drop was measured across one substrate with the engine running, the measured pressure drop was 1.1 kPa. While accurately measured, this pressure drop is not representative of the instantaneous pressure drop that affects the scavenging of the cylinder and how much intake air is brought into the cylinder. Because the exhaust valves are open substantially and flowing exhaust for less than a ⅓ of the crank rotation, it is likely that this measured average pressure is actually ⅓ of what would be allowed with a single substrate for all of the cylinders when the exhaust pulses are all combined together into one average exhaust manifold mass flow.

[0013] Early versions of this single substrate per cylinder OC system suffered substrate failures that were attributed to the pulsing effect of the exhaust gases flowing rapidly through the substrate for only ⅓ of the crank rotation. This resulted in the substrates breaking up into small pieces and flowing through the exhaust manifold towards the turbo inlet. Fortunately the EMD engine has a built in debris screen installed in front of the turbo inlet to prevent material such as this from damaging the turbine blades. Later designs of the preturbine OC system overcame this problem by adding additional material and substrate supports to enhance the durability of the substrates.

[0014] While the existing preturbine OC solution for the EMD engine solves the packaging problem, it would be preferable if a more economical and simpler single substrate solution could be found that did not have to replace every one of the existing exhaust manifolds.

[0015] The second portion of the background is most closely related to adjustable inlet guide vanes for improved emissions in EMD locomotives. Two aftertreatment systems have been developed and tested for emissions reductions in EMD powered locomotives, and both test programs noted a spike in particulate matter (PM) emissions for notch 6 engine loading. Miratech has developed and patented a preturbine diesel oxidizing catalyst (DOC) system call the V-Cat, testing results were published in ASME Paper JRCICE2007-40060 titled 'Exhaust Emissions From a 2,850 kW EMD SD60M Locomotive Equipped With a Diesel Oxidation Catalyst'. This system was focused on reducing PM emission and from Notch 3 to Notch 8, the system efficiency averaged over 55% except for Notch 6 where the reduction plummeted to approximately ½ that value at 27%. Overall this system reduced PM by 52%.

[0016] Engine, Fuel and Emissions Engineering has trademarked its Compact SCR and the final report documenting its system on a Metrolink passenger locomotive is available on their website at www.efee.com. Unlike the preturbine V-Cat system, the Compact SCR system was located downstream of the engine turbocharger exhaust outlet and its primary function was to reduce oxides of nitrogen (NOx). It has a secondary function of reducing PM and was capable of reducing PM by 61% on the locomotive duty cycle. The testing with the Compact SCR resulted in a similar PM emissions spike at Notch 6 as seen in the V-Cat DOC testing. Further the NOx reduction efficiency of the Compact SCR system at throttle setting of idle through Notch 2 were very low.

[0017] The notch 6 increase in PM emissions and the low load reduction in NOx reduction efficiency are due to two different characteristics of the EMD 2 stroke locomotive engine. The notch 6 PM increase is due the engine air fuel ratio starting to be less lean than optimum which decreases combustion efficiency of the diesel spray and increases soot which is a major part of diesel PM. On the other hand the low load reduction in SCR efficiency is because the engine air fuel ratio is becoming too lean and the exhaust temperature is very low.

[0018] These varied air fuel ratios are a function of the design of the turbocharged EMD 2 stroke engine. The EMD system has a unique combination supercharger and turbocharger. It is driven by the engine geartrain through a one way clutch up until the point that there is enough exhaust energy to drive the turbocharger faster than the gear train. The point where the turbo spools up is typically notch 7 and that is where the boost builds up and the engine runs a leaner air fuel ratio that produces less PM. At very low loads the engine is also at low RPM, but at these lower speeds the intake ports are open for a longer time giving the reduced boost pressure more time to drive fresh air into the cylinder. Also at these lower loads the engine actually needs less air because it is making less power and consuming less fuel. This causes the engine to take in even more air than is needed and this excess air in the combustion chamber lowers the exhaust temperature. At idle this problem is at its worst as the low RPM allows a long time for scavenging and the minimal engine fuel consumption further drives down the exhaust temperature. From a peak exhaust temperature over 500 C at notch 8, the exhaust temperature is just over 110 C at idle and 160 C at Notch 1.

[0019] Energy Conversion Inc. (ECI) in Tacoma, Wash. has had to overcome an additional problem in its conversion of these EMD 2 stroke engines to natural gas. In order to prevent detonation at high loads with natural gas, it was required to lower the compression ratio of the engine. Lowering the compression ratio at idle exacerbated the low RPM combustion temperature issue and in order to get lower emissions at idle, ECI incorporated a bank idling system where it only injected fuel into the cylinders on one side of the engine. This allowed each cylinder to operate with twice the amount of fuel and generate twice the amount of power. Every two minutes the engine would swap banks and run on the other half of the engine.

[0020] In addition to the bank idling technique, ECI devised an inlet throttle system to restrict the amount of air that the engine took in at idle to further increase the combustion temperature and increase the exhaust gas temperatures. This system had a set of rotating vanes pointing inward from a ring. This ring would be in front of the turbocharger compressor and had an open and closed setting. In the open setting the vanes would turn so that they were lined up in the direction of air flow and offered minimal resistance to the airflow. In the closed position an air actuator would rotate the vanes almost 90 degrees until the vanes touched and closed off the air passage except for the small round opening left over at the tips of the vanes.

[0021] This inlet restriction system developed by ECI is similar to variable angle inlet guide vanes used on some gas turbine engines and large stationary compressor equipment. When the guide vanes are in the neutral position they have no effect on the compressor upstream of them. When the guide vane are rotated from the neutral position they will add swirl to the flow, this swirl will have a different effect on air flow through the compressor depending on whether the swirl is turning the airflow with or against the rotation direction of the compressor impeller. If the flow is swirling in the direction of the centrifugal impeller rotation then the amount of pressure rise across the impellor will decrease as the impellor will not be able to put as much work or energy into the flow. This would tend to decrease the amount of mass flow across the compressor and the boost pressure leaving it. If the inlet guide vanes were turned in the opposite direction, the resulting air flow swirl would turn the air against the impeller rotation. This would increase the amount of work or energy that the impellor will impart into the air flow increasing the pressure rise. If the centrifugal impellor was part of a turbocharger, this increase in pressure rise would result in slowing down the impeller and turbine. This could be a form of limited waste gating for limiting or reducing the turbocharger shaft speed.

[0022] In addition to the lower compression ratio causing lower combustion temperatures at lower loads, the ECI natural gas conversion systems changed the air flow configuration of the engine enough that the stock EMD turbocharger could overspeed at notch 8. In order to control overspeed ECI added a waste gate system to bypass some of the high temp exhaust gasses and reduce turbine speeds.

[0023] Another system implemented by ECI in its conversion system is improved aftercooling of the intake air to reduce detonation at high loads. At low loads this improved intake air cooling would exacerbate the low combustion temperature issues at lower throttle settings. The solution was to revert the aftercooling system back to the original system for notches 3 down to idle where heated engine coolant is used to warm up the intake air. This required adding an actuated coolant control valve and some plumbing to control whether heated or cooled water was flowing to the liquid cooled after-coolers.

[0024] What would be beneficial in these applications would be an airflow control system that reduced the excessively lean low load mixtures, increased boost and air flow at notch 6, and limited turbine speed in dual fuel engines at notch 8.

[0025] The third portion of the background is most closely related to a narrow angle sonic and dual stage gas inlet valve. In the case of the ECI conversions systems for 2 stroke locomotive engines, a system called low pressure direct injection (LPDI) is used where the natural gas is injected directly into the cylinder during the compression stroke. What this leads to is a mixing challenge where the air and fuel have limited time to mix as the piston rises up to top dead center right before ignition.

[0026] This mixing challenge is why SwRI on their single cylinder development EMD 710 engine decided to do premixing of the air and fuel even though it would not be practical on an 'in service' engine as too much unburned fuel would blow through the cylinder into the exhaust while scavenging.

[0027] The in cylinder mixing issue can make prechamber operation difficult if a rich pocket of air and gas gets pushed into the prechamber which already has excess fuel in it. In this instance the prechamber will misfire and there will be no combustion for that stroke. For this reason ECI installed 'jet caps' on the first iteration spark ignited prechamber (SIP) system on the Napa Valley Wine train. The jet cap is an additional cap fixed over the end of the main Gas inlet valve (GIV). The GIV had a poppet valve at the end that controlled the flow of fuel gas into the combustion chamber. With the 'jet cap' in place, after the gas flowed thru the GIV body and past the poppet valve, it then had to flow through a small orifice at the end of the 'jet cap'. This addressed several issues, all the gas was converged into one flow stream that now had higher velocity and was pointed away from the prechamber.

[0028] Another difference between the ECI kit and the system tested at SwRI is that the ECI system has to operate at very high Lambdas. Lambda is the ratio of the actual air/fuel ratio divided by the stoichiometric air/fuel ratio. Typical 4 stroke diesel engines operate at Lambdas around 1.9 at low load to 1.4 at full power. The SwRI single cylinder development engine didn't have to operate below 50% power. At low loads, an EMD 2 stroke locomotive operates at Lambda's above 3 and at idle the Lambda can exceed 4. At these very high lambdas it would require a larger prechamber that will produce fewer NOx emissions and have a lower thermal efficiency.

[0029] A solution to the very high Lambda value is to restrict inlet flow with a throttling system at low loads. This will allow operating the engine all the way from idle to full load with smaller volume prechambers that put out less NOx emissions and operate at higher thermal efficiency.

[0030] In a uniflow 2 stroke engine, scavenging is a process of blowing inlet air over the top of the piston at bottom dead center. This entering intake air pushes the spent combustion gasses out through the open exhaust ports at the top of the cylinder. The amount of in cylinder air motion and mixing as the piston rises in the compression stroke is proportional to how much velocity the inlet air carried in with it due to excess intake air box pressure. When the inlet is throttled to help reduce the low load air fuel Lambda, a large portion of this mixing energy is lost.

[0031] It is possible to reduce the inlet air box pressure to a low enough value that not enough inlet air enters to thoroughly scavenge the cylinder and some amount of exhaust gas will remain in the cylinder when the exhaust valves close. This effect can be desirable or have negative effects. This left over combustion gas is much hotter and less dense than the incoming air, so the resulting in cylinder air mass will now be lower and the average in cylinder temperature will be hotter at the beginning of the compression stroke. This has the double effect of both lowering the Lambda for easier combustion with less ignition energy using a smaller prechamber, and also faster and more efficient combustion because the compressed air fuel mixture is already much hotter at ignition.

[0032] This is referred to as internal exhaust gas recirculation (EGR) where exhaust gas is purposely left behind to achieve these effects. In a uniflow 2 stroke, the downside of this is much less air velocity at intake port closing. This lowered in cylinder velocity and mixing energy reduces the amount of air and fuel mixing when the natural gas is injected at low loads.

[0033] A supersonic injector for gaseous fuel engines as described in U.S. Pat. No. 6,708,905 would be a solution that offers improved mixing and a bonus of lower temperature gas when injected. This particular device has two drawbacks. First it has many machined parts with complicated features that will be costly. Second, the design has a built in cavity where residual natural gas will be compressed into and remain unburned during the combustion event. Most of the compressed gas in this cavity will become methane exhaust emissions. This release of unburned methane is both a pollution emissions problem and an energy efficiency problem.

[0034] What is desired is an economical and practical way to achieve the benefits of a high velocity and focused sonic injection nozzle without the added cavity for residual unburned methane, better mixing in the combustion chamber of a natural gas engine with direct gas injection which would allow operating a uniflow 2 cycle engine to be throttled past the point that internal EGR effects are improving combustion.

[0035] The fourth portion of this background is most closely related to double pass prechamber cooling. Prechamber ignition systems are used to ignite air fuel mixtures that are too lean for a spark plug to ignite. The type of prechamber discussed here is a small prechamber at less than 5% of the combustion chamber clearance volume. Combustion inside of the prechamber will be easier to start and burn much more rapidly because the air fuel mixture is hotter and typically richer than the air fuel mixture in the main combustion chamber.

[0036] The cooling of a prechamber is one of the challenges, and the most challenging part of the prechamber to cool is the nozzle or tip area. This is because there is combustion happening on both sides of it. With insufficient cooling it has been documented that overheating prechambers will often melt the tip of the prechamber. Sometime before the tip actually melts, it will cause preignition which will limit how much power the engine can produce or cause the engine to run improperly during some conditions. An improved prechamber would be a design that has better cooling for the prechamber tip and nozzle area and is also more economical to make.

BRIEF SUMMARY OF THE INVENTION

[0037] The first portion of the summary is most closely related to a single element preturbine oxidation catalyst element for EMD turbocharged engines using twisted exhaust runners. With one revised exhaust manifold segment there is a way to use a single OC substrate without significantly affecting the exhaust flow of the cylinder closest to the turbocharger. This single substrate would replace the debris screen at the inlet to the turbo. This substrate would be installed in the last exhaust manifold segment before the turbo. Room for the substrate would be created by modifying the exhaust manifold runners for the last two cylinders.

[0038] The typical exhaust runners are a 9"×4" rectangular tube. The 9 inch dimension on the runner is along the axial flow path of the exhaust manifold segment. By having the rectangular tube transition from a 9"×4" shape to a 4"×9" shape as it meets the exhaust manifold segment, it will free up approximately 5" of axial length for a 5" long OC substrate.

[0039] This single substrate system will save the cost of making 3 extra exhaust manifold segments and 15 extra OC substrates.

[0040] By averaging the exhaust pulses all together in one flow, it minimizes the pulsing effect on the substrate and the substrate experiences relatively consistent and smooth exhaust flow.

[0041] By replacing the original debris screen it removes the effect of the pressure drop of the screen which helps to offset the pressure drop of the OC substrate on engine performance.

[0042] The holes in the OC substrate will be smaller than the holes in the original debris screen so it will be more effective at stopping smaller bits of debris from damaging the turbine blades and reducing the turbo performance.

[0043] The long OC substrate in front of the turbo converging duct acts as a flow straightener removing any swirl in the flow that may have been caused by the exhaust runner pulses entering the exhaust manifold close to the turbo inlet.

[0044] The second portion of the summary is most closely related to adjustable inlet guide vanes for improved emissions in EMD locomotives. With minor development and the addition of a modulated position actuator, the ECI inlet throttle system could be used to variably reduce the air flow in the EMD engine in very small increments. If the modulated position actuator had more than 90 degrees of travel it could also be used to increase boost at notch 6 and decrease turbine speed at notch 8.

[0045] The beneficial effects of restricted airflow at idle has been demonstrated with the inlet throttle restrictor fully closed in previous ECI natural gas conversions. In notches 2 and 3 where the engine loading is getting higher and detonation is not a threat it is beneficial to have higher intake air temperature so that the natural gas will combust easier. Current ECI dual fuel systems do not consume natural gas in notches 1 or 2, and at notch 3 natural gas substitution is limited to 65% because the combination of very lean mixtures and low compression make it difficult to ignite the natural gas. Because of the way 2 stroke engines scavenge, it would be possible with a variable inlet guide vane system to drop the intake air pressure and flow enough that all of the burned exhaust gases were not pushed out by the incoming intake air. This is referred to as internal exhaust gas recirculation (EGR) and has several benefits when used at low loads. Because the left behind exhaust gases are much hotter than the incoming intake air, the in cylinder temperatures of the mixed intake and EGR gases will be higher. Also because the EGR gases were hotter, they would be less dense. As the cylinder pressure when the intake ports and exhaust valves are closed will be almost the same, this hotter less dense mixture will have less mass. This will make the air fuel ratio less lean. The combination of a less lean mixture that is also hotter at the start of ignition will improve the combustibility of the natural gas and will increase the amount of gas at notch 3 that can replace diesel fuel, and will also allow the substituting of some natural gas for diesel fuel at notches 2 and 1, possibly even at idle. If the system is effective enough it could eliminate the need for the coolant diverter valve used to help preheat the intake air.

[0046] This variable inlet restriction would also be a benefit to a diesel fueled EMD engine that is using a Compact SCR system, as it can drive the exhaust temperature up at idle, and notches 1 through 3 where the Compact SCR system was not functioning or was less than 30% efficient.

[0047] These increases in combustion efficiencies at low loads due to the less lean mixtures and potential internal EGR will not only reduce emissions, they will increase thermal efficiency at these low loads.

[0048] The third portion of the summary is most closely related to a narrow angle sonic and dual stage gas inlet valve. What is proposed here is a gas inlet valve (GIV) that utilizes the valve head and valve seat at a narrow angle to accelerate and focus the gas flow.

[0049] This configuration has several advantages. First it merely requires a change in operating pressure and revised machining on two components to gain this effect.

[0050] Second, as the gas exits from an annulus instead of a hole, the gas exits as a cone formed from a sheet of gas with both an inner surface and outer surface. This surface is where the mixing happens and this design will have over twice the surface area for entraining the surrounding air.

[0051] Third, as the nozzle is formed by the movement of the poppet valve from the seat, the stoke can be adjusted to different sonic throat areas. Allowing longer valve opening times at higher pressures and lower flows.

[0052] This design completely eliminates the issue of residual unburned gaseous fuel remaining inside of a cavity in the GIV or Jet Cap after combustion.

[0053] These sonic GIV units can operate with any gaseous fuel including propane and hydrogen.

[0054] The fourth portion of this summary is most closely related to double pass prechamber cooling. Proposed is a two piece prechamber body and nozzle design that enhances the cooling of the prechamber by incorporating a double pass coolant system to insure that an adequate amount of cooling fluid makes it to the bottom of the prechamber, and then evenly flows around the periphery of the the prechamber to a point past the top of the inner prechamber combustion chamber wall.

[0055] The fifth portion of this summary is most closely related to an OPOC variable compression ratio mechanism. While the OPOC engine being developed by ECO Motors is not an EMD 2 Stoke engine as currently used in locomotives, it does offer interesting possibilities as a power plant for genset type locomotives or as a Head End Power generator engine for passenger locomotives. The value of the OPOC's low weight and volume compared to its power output are even higher for these application when there is an effort to operate the locomotive on an alternate fuel. A typical diesel engine design converted to natural gas will need to be scaled up 30% bigger in size to make the same power.

[0056] Because of the unique nature of the OPOC engine design, it is possible to incorporate an infinitely adjustable variable compression ratio (VCR) using an outer wrist pin with an offset inner wrist pin bore.

[0057] A sliding spline fit is used to control the rotation of the outer wrist pin, because this is a two stroke engine, the piston will always be under compression when operating so that all of the VCR component slop should be taken up. The only wear items would be the parts of the sliding spline, and they are replaceable without having to remove the piston.

BRIEF DESCRIPTION OF THE DRAWINGS

[0058] The drawing figures depict one or more implementations in accord with the present concepts, by way of example only, not by way of limitations. In the figures, like reference numerals refer to the same or similar elements.

[0059] FIG. **1** is a side view of a typical 16 cylinder EMD turbocharged engine as used in locomotives.

[0060] FIG. **2** is a side view of the same engine in FIG. **1** except that the exhaust system has been revised to accommodate a single substrate OC system.

[0061] FIG. **1** is an isometric view of a prior art GIV manufactured by Energy Conversions Inc (ECI)

[0062] FIG. **2** is a cross section view of the ECI GIV illustrating the conventional poppet valve and valve seat insert at full lift.

[0063] FIG. **3** is a side view of the prior art Miratech preturbine V-Cat system.

[0064] FIG. **4** is a side view of an EMD 16 cylinder illustrating the location of the Variable Inlet Guide Vane Unit.

[0065] FIG. **5** is an isometric view of the turbocharger and the Variable Inlet Guide Vane Unit with its blades closed.

[0066] FIG. **6** is an isometric view of a prior art Variable Inlet Guide Vane Unit with the blades partially open.

[0067] FIG. **7** is an isometric view of an ECI manufactured gas inlet valve (GIV).

[0068] FIG. **8** is cross section view of a conventional poppet valve in the open position of a prior art GIV.

[0069] FIG. **9** is a cross section view of the revised poppet valve and valve seat insert to achieve sonic gas injection flow.

[0070] FIG. **10**A is a cross section view illustrating a dual stage hydraulic valve assembly in the closed state.

[0071] FIG. **10**B is a cross section view illustrating a dual stage hydraulic valve assembly in the fully open state.

[0072] FIG. **10**C is a cross section view illustrating a dual stage hydraulic valve assembly in the partially open state.

[0073] FIG. **11** is a cross section view of an EMD cylinder head with a prechamber installed.

[0074] FIG. **12**A is a close up view of FIG. **11**.

[0075] FIG. **12**B is a Detail View of FIG. **12**A.

[0076] FIG. **13** is an exploded view of an OPOC engine Variable Compression Ratio system.

DETAILED DESCRIPTION OF THE INVENTION

[0077] To facilitate an understanding of the present disclosure, a number of terms and phrases are defined below:

[0078] Hydrocarbon (HC): Emissions resulting from incomplete combustion.

[0079] Main Charge: The air fuel mixture in the main combustion chamber space between the piston top and the cylinder head. If an opposed piston engine, this would be the space between the opposed piston faces.

[0080] Particulate Matter (PM): Particulate matter is a criteria pollution emitted from many sources. In this document we will commonly refer to it simply as PM. It could include both diesel soot PM that is considered toxic in California or the type of PM created by the consumption and combustion of lube oil from an engine. While still considered PM as a criteria emission, the PM from lube oil consumption is considered less toxic than diesel soot.

[0081] The first portion of the detailed description is most closely related to a single element preturbine oxidation catalyst element for EMD turbocharged engines using twisted exhaust runners. FIG. **1** is a side view of a typical 16 cylinder EMD turbocharged engine as used in locomotive and marine applications. In this prior art configuration Engine 2 has an exhaust system along the top of it. The exhaust system is composed of three exhaust collector segments **4** and one turbocharger adapter exhaust collector segment **8** that collect the exhaust gases from the 16 engine cylinders into one combined exhaust mass flow. Each one of these exhaust collector segments connects to 4 of the engines 16 cylinders, with exhaust gases flowing from an individual engine cylinder to an exhaust collector segment through an exhaust runner **6**. The standard exhaust runner **6** is a 4 inch by 9 inch rectangular tube. The longer 9 inch dimension is shown along FIG. **1** as going left to right and the four inch dimension is normal to FIG. **1**. The exhaust gasses flow in a direction from the bottom of the Figure through an exhaust runner **6** up into an exhaust collector segment. Each exhaust collector segment has two pairs of exhaust runners **6**, only one is visible as the second exhaust runner **6** of each pair is directly behind the first exhaust runner **6**. In some version of the EMD engines, the pairs of exhaust runners **6** are combined together into one larger runner. Sometimes this larger runner would have a shared wall in between, keeping the exhaust gases from the two cylinders separate until they mixed with the combined exhaust flow in the exhaust collector segments. In other cases this was missing or removed and the exhaust gases from the pair of engine cylinders would mix in the combined exhaust runner **6** volume before mixing with the combined exhaust flow in the exhaust collector segments. This would appear the same in FIG. **1** and functionally does not affect this description of the prior art.

[0082] The three exhaust collector segments **4** and one turbocharger adapter exhaust collector segment **8** are connected to each other by flexible bellows **5** at three places. The now combined exhaust gasses flow from the turbocharger adapter exhaust collector segment **8** thru debris screen housing **10** and small flexible bellows **12** into the turbocharger inlet **14**. As the combined exhaust mass flows through the debris screen housing **10**, it must pass through debris screen **16**. Debris screen **16** is a metal plate installed in debris screen housing **10** with a large number of small holes that will allow the exhaust gases to flow through it, but will block any small solid parts from traveling with the exhaust gases into the turbocharger and damaging the turbine blades. This debris screen **16** does cause a small pressure drop in the exhaust system which reduces engine performance and efficiency, but it prevents damage to the turbocharger assembly in the case of a component failure elsewhere in the engine. This is a valuable trade off as the turbocharger is one of the most expensive parts of the engine.

[0083] FIG. **2** is a side view of the same engine in FIG. **1** with a revised exhaust system to include a single substrate OC system. Engine 2' has similar components in its exhaust system upstream of turbocharger adapter exhaust collector segment **8'** and downstream of debris screen housing **10'**. The primary difference is the deletion of the debris screen **16** and the addition of the OC substrate **18** into modified turbocharger adapter exhaust collector segment **8'**.

[0084] Turbocharger adapter exhaust collector segment **8'** has been modified to allow the OC substrate **16** to slide into it. The primary modification to make this possible is the reshaping of exhaust runner **6'**. Where exhaust runner **6** had a consistent 9 inch by 4 inch rectangular shape along its path, the cross section of exhaust runner **6'** will change along its length. It will start with the same 9 inch by 4 inch shape at the engine cylinder port, but as it travels towards the turbocharger adapter exhaust collector segment **8'** its shape will transform as depicted in FIG. **2**. The goal is to have a similar cross section area along the exhaust flow path of exhaust runner **6'**, but have the length and width dimension transition from 9 inches by 4 inches to something close to 4 inches by 9 inches. This will allow the creation of a cylindrical pocket that allows OC substrate **16** to slide in. The pocket does not need to be cylindrical, but the changing cross section of the exhaust runners **6'** is what allows a single OC substrate **16** to fit between the exhaust runner **6'** and the small flexible bellows **12**.

[0085] OC substrate **16** is likely to be a round metallic substrate approximately 18" in diameter and 5" thick. These sizes and substrate material composition will vary depending on system design. OC substrate **18** may slide all the way into either turbocharger adapter exhaust collector segment **8'** or into debris screen housing **10'**, but is most likely to protrude partially into each. Other shapes of substrate and pockets to fit it in may not be cylindrical, but may be oval or rectangular.

[0086] In this embodiment it is designed that the OC substrate **18** slides into a pocket created in turbocharger adapter exhaust collector segment **8'** and is retained in that pocket by debris screen housing **10'**. In another embodiment, turbocharger adapter exhaust collector segment **8'** and debris screen housing **10'** may be combined into one assembly with OC substrate **18** sliding into this assembly from direction normal or close to normal to the axis of exhaust gas flow. This would require some kind of cover plate to be used to cover the pocket opening similar to the cover plates used in the Miratech V-Cat design.

[0087] FIG. **3** is a side view of the prior art V-Cat system 80 patented and manufactured by Miratech. The V-Cat system 80 comprises four exhaust collector segments **82** which replace the three exhaust collector segments **4** and one turbocharger adapter exhaust collector segment **8** from FIG. **8**. In each exhaust collector segment **82** are four individual OC substrates to service the exhaust gases of four engine cylinders. A pair of OC substrates is captured on each side of an exhaust collector segment **82** by a cover **84**. Each exhaust collector segment **82** has four exhaust runners **86** similar to the exhaust runners **6** in FIG. **8** and FIG. **9**.

[0088] It is a cover similar to cover **84** that could be used to retain a single OC substrate **18** into a combined turbocharger adapter exhaust collector segment **8'** and debris screen housing **10'**.

[0089] The second portion of the detailed description is most closely related to adjustable inlet guide vanes for improved emissions in EMD locomotives. FIG. **4** is a side view of a 16 cylinder EMD engine 2. Turbocharger 15 is mounted to engine 2. Variable inlet guide vane unit **6** is mounted to the compressor inlet of turbocharger **4**.

[0090] Even with as much value and performance that the variable inlet guide vane units adds, FIG. **4** illustrates what a small and easy to package system the variable inlet guide vane unit is.

[0091] No parts on the engine need to be replaced, only the intake pipe bringing in outside air to the turbocharger compressor inlet.

[0092] On the other hand this unit may allow the simplification and cost reduction of the ECI conversion kit by eliminating the waste gate assembly the aftercooler diverter valve and its extra plumbing.

[0093] FIG. **5** is an isometric view of the engine turbocharger **24** and the variable inlet guide vane unit **26**.

[0094] In this view the guide vanes **28** are in the fully closed position, this leaves a small flow area **30** formed by the blade tips. In the prior art version of this device the valve was either fully open or fully closed, manipulation of this state was done by actuator **32**.

[0095] New embodiments of this system will have actuator **32** upgraded to have variable positions. In one embodiment a 90 degree variable position actuator may be used and the fully closed position will not have the guide vanes **28** rotated so far that they touch. This now allows the vanes when rotated 90 to have traveled past neutral and be positioned at an angle to cause increased boost at notch 6 or act as a waste gate limiting turbine rpm at notch 8.

[0096] A further embodiment will have an actuator like the Delphi Smart Remote Actuator that has 120 degrees of travel. With this variable actuator, the guide vanes **28** can be rotated fully closed and still have the range to rotate 30 degrees past neutral well into the range where notch 6 boost is increased.

[0097] FIG. **6** is an isometric view of a prior art inlet guide vane unit **26'** with the guide vanes **28'** partially open.

[0098] The third portion of the detailed description is most closely related to a narrow angle sonic and dual stage gas inlet valve. FIG. **7** is an isometric view of a standard ECI GIV assembly **40**. It illustrates the relationship between the GIV body **41** the valve seat insert **42** and the poppet valve **43**. In this view the poppet valve is in the fully extended position. This particular valve assembly is designed to inject natural gas into and EMD 2 Stroke natural gas engine on the compression stroke. It is possible to use this direct injection valve design and any embodiment of the current invention in any reciprocating engine using any gaseous fuel.

[0099] FIG. **8** is a cross section view of the prior art GIV assy **40** from FIG. **7**. FIG. **8** illustrates the poppet valve **43** and valve seat insert **42** when the poppet valve is fully extended. This valve is typical in construction to the exhaust and intake valves in most recipricating piston engines. The valve seat area **44** is around 0.065" wide and the valve seat angle is 60 degrees from the valve axis. The intent of this valve system is specifically to allow the most air flow to pass through it with the minimal amount of pressure drop during the time is has available to be open. There is minimal consideration as to what the characteristics the exiting air flow has and the pressure drop across these valves is typically under 2:1 for conventional engine intake and exhaust valves and up to 4:1 for the GIV units used on turbocharged EMD engines with a natural gas feed pressure of 80 psi.

[0100] FIG. **9** is a cross section of the new poppet valve **43'** and valve seat insert **42'** design. Just the modification of these two parts converts ECI's standard GIV into a version that creates a sonic cone of injected gaseous fuel. The view on the left shows the valve in the closed position. Significantly different from FIG. **8** is that the flow cone angle is 50 degrees instead of 120. The valve seat angle is actually 20 degrees from the axis of poppet valve **43'** instead of 60 degrees in the prior art design. The cone angle could be more or less and 50 degrees. The narrower this angle is angle aims the injected gas stream further into the cylinder before it impinges on the cylinder wall for improved mixing. Also the valve head contact width has increased from 0.063" to 0.185". This longer contact area is a result of using the valve head and the valve insert seat to create the expanding nozzle shape. It would be possible to decrease the seat width by back cutting the upper part of the valve seat insert or the upper part of the valve head, but that would require the flow path to be longer to get the desired expansion ratio.

[0101] The right view illustrates the valve in the open position. This valve has a full stroke of 0.103" to achieve the same flow as the FIG. **2** valve did at 0.230". This is due to the higher operating pressure of over 250 psi. At the 0.103" valve lift the normal distance is 0.052 inches between the poppet valve head and the valve seat surface. That gap remains constant throughout the nozzle, but the area increases because the radius of the cone increases as it moves towards the exit. Both the 60 degree cone angle and 0.103" lift are arbitrary; this design could be executed with many different angles and lifts.

[0102] FIG. **10**A is a cross section view of a hydraulic actuator for the GIV Assy **40** with two discrete open positions. This view illustrates the GIV Assy **40** in the closed position. In this view the plunger **51** is inside of the plunger body **50**, and it is the plunger **51** that the hydraulic fluid pushes down on to open up the poppet valve **43**. These two parts are consistent with the standard prior art version of GIV **40**. What is added in this embodiment is the plunger follower **52**, the plunger stop body **55** and the movable plunger stop **54**.

[0103] FIG. **10**B the GIV Assy **40** is in the full open position. The plunger **51** was forced down by the hydraulic fluid until it contacted the movable plunger stop **54**. The movable plunger stop **54** is resting on the top surface of the plunger stop body **55**. When the plunger **51** started to move in the downward direction, it contacted, pushed down on and moved the plunger follower **52**. The plunger follower **52** was in contact with the top of the poppet valve **43** and pushed it down also. All three parts continued to move downward until the plunger motion **51** was stopped as it contacted the movable plunger stop **54**.

[0104] FIG. **10**C illustrates the GIV Assy **40** in the partially open position. To stop the poppet valve **43** in this position, pressurized hydraulic fluid is fed into the Plunger Stop hydraulic Port **53**. This pressurizes the Plunger Stop Hydraulic Cavity **57** and this pressure forces the Movable Plunger Stop **54** to move up until it contacts the bottom of the Plunger Body **50**. With the movable Plunger Stop **54** in this position, the Plunger **51** now travels a shorter distance before contacting the Movable Plunger Stop **54** which will now limit the poppet valve **43** opening to a reduced stroke in the Partially Open Position. The Movable Plunger Stop **54** is able to keep the Plunger **51** from moving it down because it has more surface area exposed to the hydraulic fluid pressure in the Plunger Stop Hydraulic Cavity **57**.

[0105] This system could be designed to have more than one movable stop by multiplying certain features in this design.

[0106] The standard way to operate an ECI low pressure direct injection EMD conversion is to have the valves stay open for set amount of time for each piston stroke. This time period is set by the amount of time available at high RPM to inject gas after the intake ports are closed. After this time period is set, engine load is controlled by adjusting the gas supply pressure to the injectors. As the load and RPM decreases and less fuel is required, the supply pressure is decreased. It would be possible to maintain a constant pressure and then reduce the injection time as fuel demand decreased, but that may decrease the amount of air and fuel mixing because the high velocity fuel gas was injected for a shorter period of time.

[0107] On a fuel system using standard poppet valves that achieve sonic flow at the valve periphery this would be a measurable effect.

[0108] This is the primary advantage of the GIV with multiple valve stroke settings. It reduces the total amount of injector feed pressure. Instead of reducing the pressure for all 8 throttle notches in a locomotive. The pressure could be reduced incrementally for Notches 7 and 6, and then Notch 5 will have the GIV assy **40** operate at reduced poppet valve **43** lift and a slightly longer valve open time because the RPM is now lower. From this point both the valve open time and gas supply pressure will be reduced incrementally down to the minimum flow needed at idle. The goal is to have the GIV fuel gas feed pressure remain high enough that good mixing is maintained, but balance that with manipulation of the valve open time to maximize the amount of time the high velocity injected gas is mixing with the air in the combustion chamber.

[0109] As an example, Instead of having a constant 80 milliseconds of injection time starting at a pressure of 300 and dropping to 100 at notch 1. Now the highest 3 throttle notches will have an 80 ms injection time and pressure will drop to 250 in notch 6. At throttle notch 5 the injection time is raised to 115 ms, the poppet valve **43** lift is 40% of full open and the injector feed pressure is raised back to 300. By notch 3 the injection time has be lowered back to 80 ms and pressure feed pressure has only been reduce down to 275. By throttle notch 1, the pressure has been further reduced to 220. By ending at a 220 psi supply pressure instead of 100 psi, the exit velocity of the gas leaving the GIV should still be. If it had dropped down to 100 psi, it would likely have become subsonic in the GIV.

[0110] An interesting further use of this concept would be in large ship engines. Both 2 stroke and 4 stroke engines that are diesel pilot ignited would benefit from added swirl in the combustion chamber. Any number of these GIV's could be placed offset from the engine cylinder axis and tilted at an angle to induce a swirl to the air in the combustion chamber. If more than one supersonic GIV is used, they should have a similar angle in reference to the engine cylinder axis so that they induce swirl in the same direction. This swirl of air around the engine cylinder axis in the combustion chamber improves the combustion of the diesel pilot helping to lower PM or NOx emissions. This is because the swirl improves the air utilization during mixing controled combustion as the surface of the diesel fuel jet is in contact with more air molecules than it would be if the air was stationary.

[0111] Another interesting possibility will be the incorporation of sonic flow GIV's with an opposed piston engine. If only one sonic GIV was used per cylinder there would be the risk of the gas flow impinging on the opposite cylinder wall. This may or may not have detrimental effects such as a colder spot at the cylinder wall with possible lubrication or thermal stress issues. If cylinder wall impingement is to be avoided or for improved mixing, two of these sonic GIV's could be placed directly opposite of each other across the combustion chamber, in this case the two cone shape flows would collide in the middle of the chamber causing a great amount of turbulence and entraining significantly more intake air in the cylinder before the cold gases reach the cylinder walls.

[0112] The fourth portion of this detailed description is most closely related to double pass prechamber cooling. FIG. **11** is a cross section of a cylinder head **58** illustrating the placement of the prechamber **59** in relation to the cylinder head **58** and the piston **70'**. The o-ring **61** at the top creates a cavity between the prechamber **59** outer surface and the cylinder head **58** pocket wall. This cavity is sealed at the bottom where the lower tapered section of the prechamber **59** is forced against the bottom of the cylinder head **58** pocket. This seal at the bottom is designed to resist blow by of combustion gasses when the engine is operating so it will not have an issue keeping the prechamber coolant out of the engine cylinder.

[0113] At the top of the prechamber **59** is the cooling fluid inlet **60**. Pressurized cooling fluid is injected here and an internal passage brings the cooling fluid to an exit port on the outer surface of the prechamber below the o-ring **61**. The cooling fluid can be many different fluids including water, but in this preferred embodiment it would be engine oil to eliminate the need for return plumbing to a separate cooling fluid reservoir.

[0114] In this embodiment, the cooling fluid is injected into a feed groove **67** around the prechamber **59**. This feed groove **67** acts as a manifold and helps distribute the cooling fluid around the entire circumference of the prechamber **59** body before it starts to flow through the narrow cavity between the prechamber **59** body and cylinder head **58** wall.

[0115] In this prechamber **59** embodiment is a diesel injector. This prechamber configuration uses a micropilot of diesel fuel to start ignition. This invention would work in a similar fashion with a spark plug ignited prechamber with or without additional fuel being added to the prechamber **59**.

[0116] Another embodiment not depicted could replace the single feed groove **67** around the prechamber **58** body with a spiral groove. The upper portion of the prechamber **58** body has a thicker wall section and in this area of the prechamber body a spiral groove could be cut into the outer surface of the prechamber. Possibly 10 to 15 turns, it would appear similar to an acme square thread except the eternal thread feature would be thin compared to the size of the passage. This spiral passage would slow the cooling fluid down allowing it more time to absorb heat from the prechamber body. The spiral groove feature could also give the cooling fluid more than twice the surface area to transfer heat.

[0117] FIG. **12**A is a close up cross section of the lower half of the prechamber **59** body. FIG. **12**B is a detail view of FIG. **12**A. Clearly visible is the second body nozzle **68** that slides over the prechamber **59** body from the bottom. The nozzle **68** is designed to contact the prechamber **59** body at two points with press fit pilots. There is a press fit pilot at the top of the nozzle **68**; this pilot is in a low stress area and only seals against the cooling fluid going from the coolant first pass straight to the coolant collection groove. This is also the area that the nozzle **68** and the prechamber **59** body could be optionally welded together.

[0118] If the prechamber **59** body upper half was equipped with an optional spiral coolant groove it would end before the optional weld area.

[0119] The second contact point between the nozzle **68** and the prechamber **59** body is the press fit at the bottom of the nozzle **68**. This press fit is important as it seals the prechamber combustion area from the coolant cavity around the prechamber **59**. The thermal expansion stress from the prechamber **59** body heating up and the forces of combustion both enhance the sealing capacity.

[0120] With or without the optional spiral cooling groove, the coolant first pass **64** starts at the point where the cooling fluid is first injected at feed groove **67** on the exterior of the prechamber **59** and continues down the length of the outer surface of both the prechamber **59** body and nozzle **68**. As the cooling fluid moves along the coolant first pass **64**, it will be simultaneously absorbing heat from the prechamber **59** and nozzle **68** and transferring that excess heat to the cylinder head **58** surface.

[0121] Just before the contact point where the nozzle **68** seals to the cylinder head **58**, there is a ring of radial coolant passages **66**. These radial cooling passages **66** are at the end of the coolant first pass **64** and the start of the coolant second pass **65**. These radial coolant passages **66** are equally spaced small holes around the nozzle **68** and the pressure drop that the cooling fluid experiences as it transitions these radial coolant passages **66** equalizes the flow around the perimeter of the nozzle **68**. This encourages the flow before and after the radial coolant passages **66** to be more evenly distributed even if the thickness of the first and second coolant passes may vary slightly due to machining tolerances of the prechamber **59** or the head **58**.

[0122] Once the cooling fluid enters the coolant second pass **65**, it will flow upwards around the outside of the prechamber **59** and the inside surface of the nozzle **68**. This cooling fluid ends up collecting in coolant return groove **62** and exiting prechamber **59** through coolant exit port **63**. This cavity for coolant second pass **65** should be thinner than that of coolant first pass **64** so that the cooling fluid travels faster and picks up less heat. The goal is to absorb only the amount of heat required out of the prechamber **59** body, but not so much that it can over heat the cooling fluid. When the coolant fluid is oil, overheating will result in the oil coking in this area and the corresponding overheating and failure of the prechamber due to lack of cooling fluid. A slower velocity along the outside of the nozzle **68** in the coolant first pass **64** will allow the cooling fluid to absorb more heat from the nozzle **68** and transfer it to the cylinder head **59** wall.

[0123] There are three general goals of prechamber cooling; keeping the spark plug from overheating, keeping the nozzle **68** from getting hot enough to cause pre-ignition, while keeping the prechamber **59** inner combustion chamber walls hot enough to insure easy and rapid combustion internally.

[0124] The coolant first pass around the top of the prechamber **59** is the area that will control spark plug temperature. The optional spiral cooling groove could enhance that cooling if needed. Nozzle **68** will get cooling from both coolant passes and will transfer some heat to the cylinder head **58** at its contact point. The heat transfer between contacting metal surfaces can be an order of magnitude less than the heat transfer through conduction of the base metal. Although the nozzle **68** to cylinder head contact **58** point is a cooling path, it is likely that significantly more heat from the nozzle is conducted up through the nozzle and absorbed by the cooling fluid that passes by two surfaces on the nozzle. The prechamber body **59** wall around the prechamber combustion chamber is left as thick as possible to reduce the heat conduction rate and it is only cooled by a single pass of the cooling fluid.

[0125] The fifth portion of this detailed description is most closely related to an OPOC variable. This variable compression ratio system would operate on the outer pistons in the OPOC design.

[0126] FIG. **12** is an exploded view of the VCR system. The outer wrist pin **71** slides into the piston **70**. There is an offset hole in the outer wrist pin **71** that the inner wrist pin (not shown) would be captured by. It is by rotating this outer wrist pin **71** around the inner wrist pin that the compression ratio is varied. The outer wrist pin **71** has a set of teeth machined into it and these teeth match the teeth cut into the rack gear **72**. The rack gear is free to slide axially along a bored hole in the piston **70**. As the rack gear **72** moves relative to the piston **70** it rotates the outer wrist pin **71**, adjusting the compression ratio. The rack gear **72** has a female threads cut into it and the rack gear threaded insert **73** has a matching male thread on its OD that interfaces with the rack gear **72** internal thread. The rack gear threaded insert **73** is axially restrained in the piston **70** between a boss inside and the threaded insert retainer **74** that bolts to the back of the piston. It is the rack gear threaded insert **73** that positions the rack gear **72** axially in the piston **70** to set the compression ratio. The VCR actuator **75** is attached to the engine end cover and is fixed in place relative to the reciprocating motion of piston **70**. It has a male splined shaft **76** that interfaces with the female internal splines inside the rack gear threaded insert **73**. As the piston reciprocates inside its cylinder, the rack gear threaded insert **73** slides back and forth over the VCR actuator male splined shaft **76**. It is the VCR actuator that sets the compression ratio in each cylinder. In this embodiment there is an actuator for each cylinder in the engine. It would be possible to belt drive multiple spline rod assemblies with one actuator.

[0127] In this design both the VCR actuator **75** male splined shaft **76** and the rack gear threaded insert **73** can be replaced as service items without disassembling the engine.

[0128] It should be noted that various changes and modifications to the presently preferred embodiments described herein will be apparent to those skilled in the art. Such changes and modifications may be made without departing from the spirit and scope of the present invention and without diminishing its attendant advantages.

I claim:

1. A single substrate oxidation catalyst system comprising:

a turbocharger adapter exhaust collector segment having a first end and a second end;

a debris screen housing in fluid communication with the second end of the turbocharger adapter exhaust collector segment; and

an oxidation catalyst substrate located in the in the second end of the turbocharger adapter exhaust collector segment, wherein the oxidation catalyst substrate slides into and out of position in the second end of the turbocharger adapter exhaust collector segment.

* * * * *