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(54) **ROTARY VALVE CONSTRUCTION**

(75) Inventors: **Anthony Bruce Wallis**, Gladesville (AU); **Andrew Donald Thomas**, North Epping (AU); **Klaus Juergen Roeske**, Lugarno (AU)

Correspondence Address:
BIRCH STEWART KOLASCH & BIRCH
PO BOX 747
FALLS CHURCH, VA 22040-0747 (US)

(73) Assignee: **BISHOP INNOVATION LIMITED**, North Ryde, New South Wales (AU)

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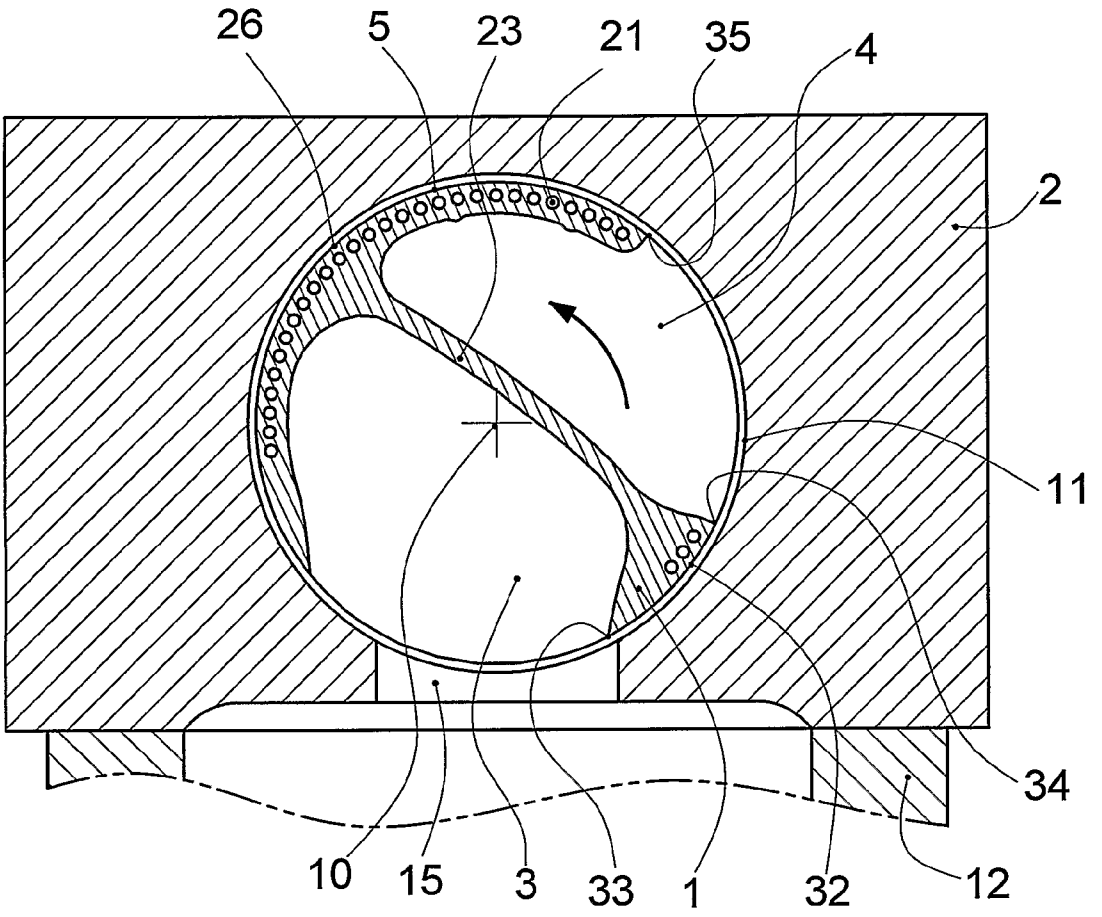
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(57) **ABSTRACT**

An axial flow rotary valve (**1**) for an internal combustion engine and method of manufacture thereof. The valve comprising a centre portion (**5**) having a cylindrical periphery, inlet and exhaust portions (**9, 16**) both smaller in diameter than the centre portion, and an inlet port (**3**) and an exhaust port (**4**) terminating respectively as an inlet opening (**6**) and an exhaust opening (**7**) in the periphery of the centre portion. The centre portion having at least one elongate passage (**21**) communicating between the inlet portion and the exhaust portion, the whole of the passage being disposed circumferentially outside the openings and radially outside at least one of the inlet and exhaust portions.



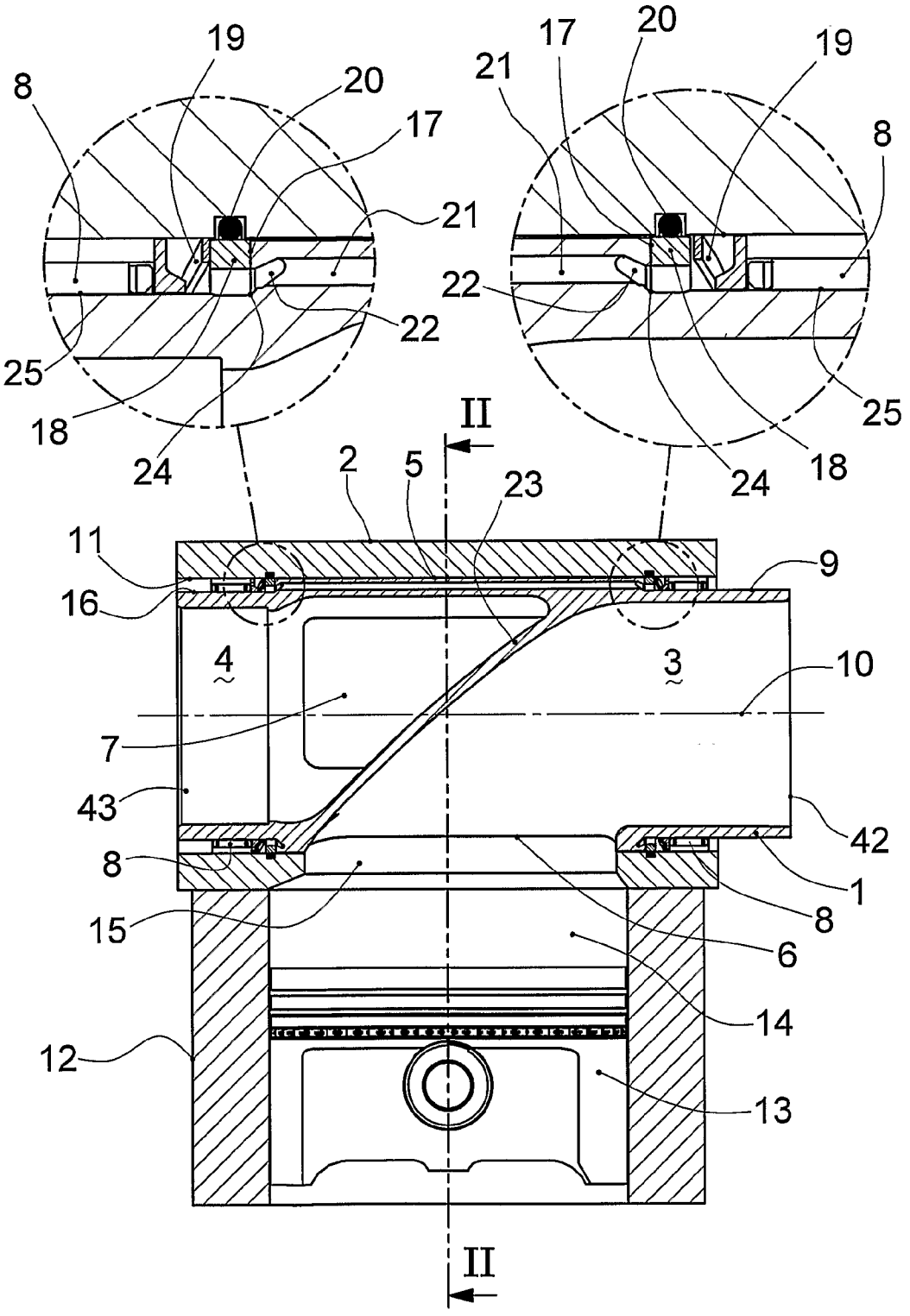

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FIG. 1

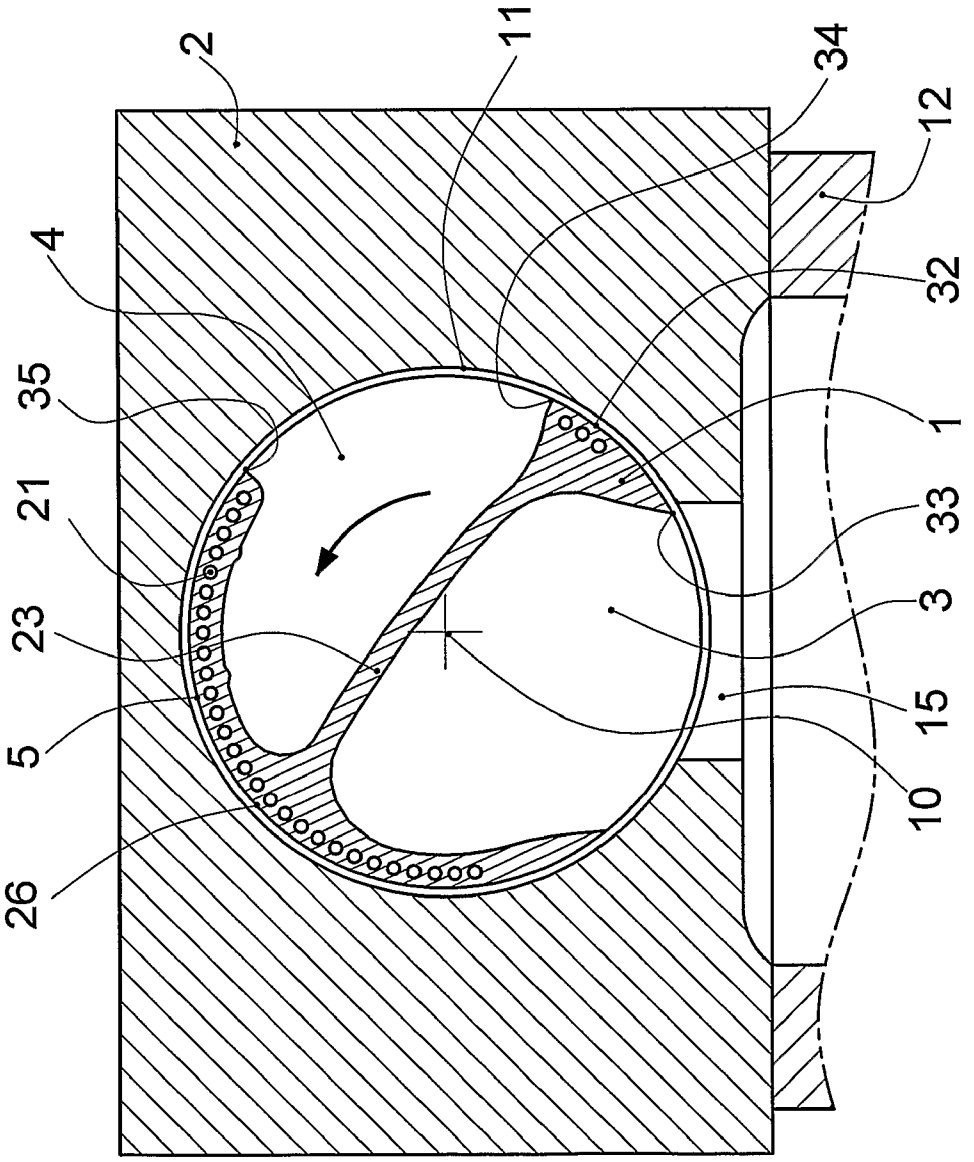


FIG. 2

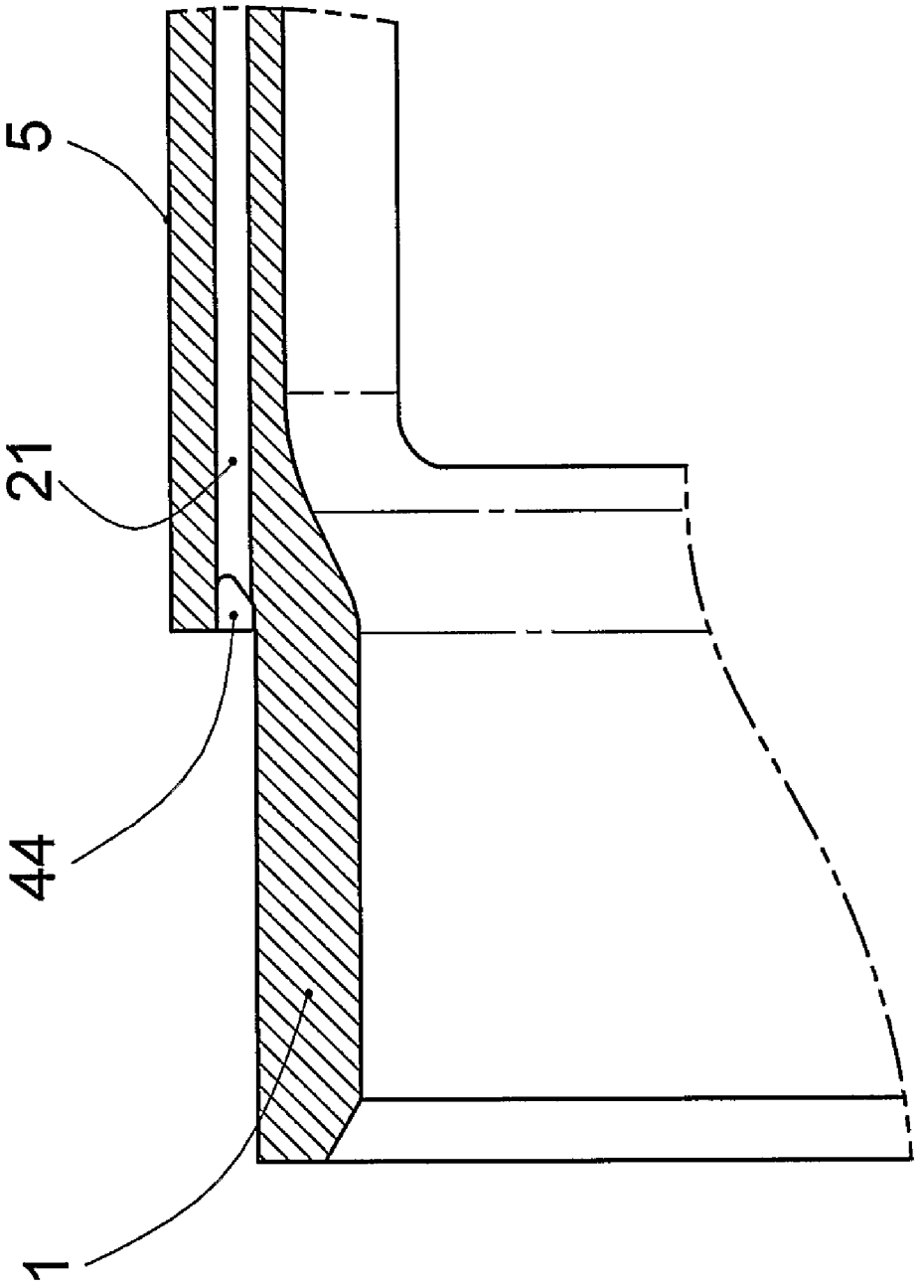


FIG. 3

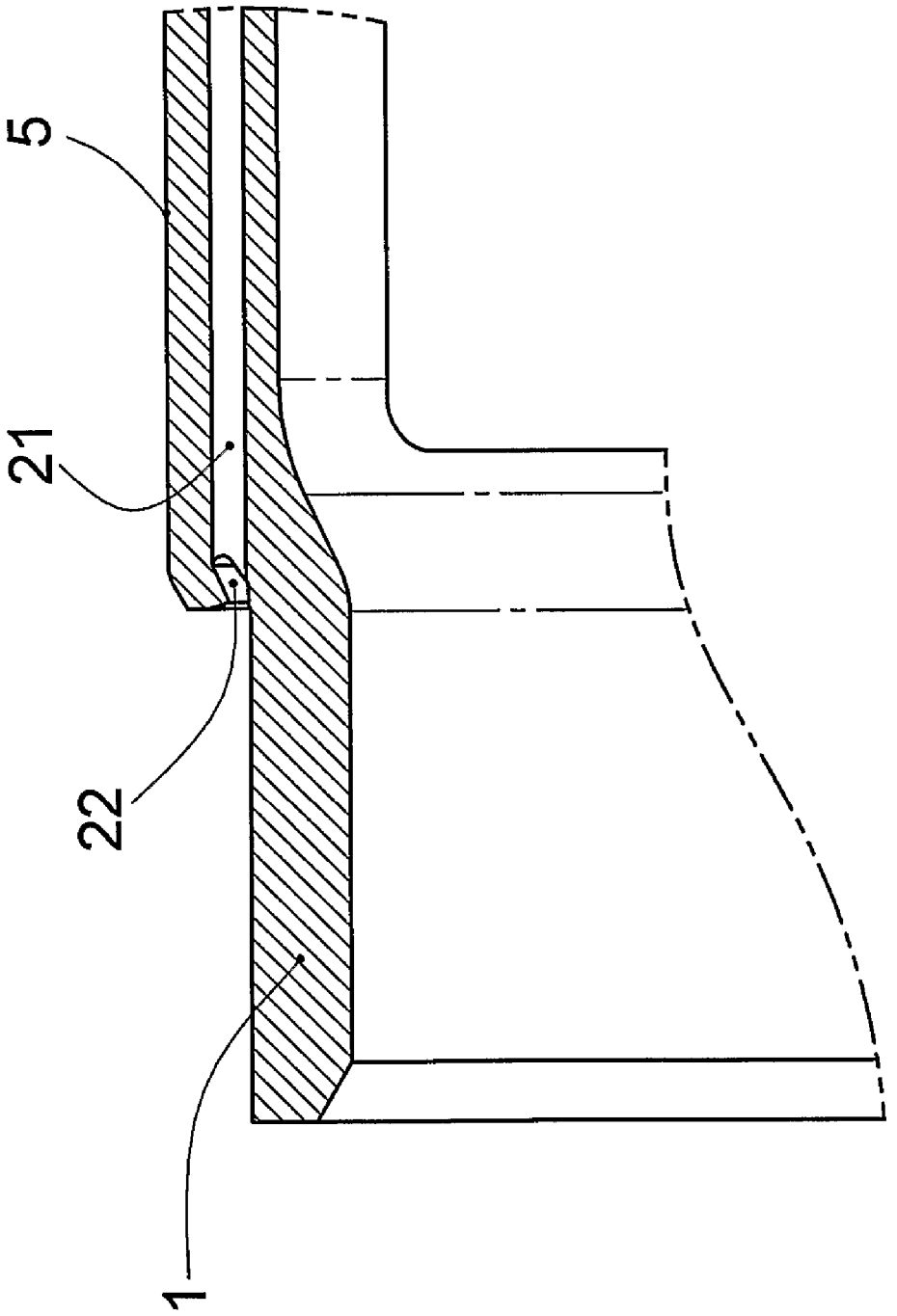


FIG. 4

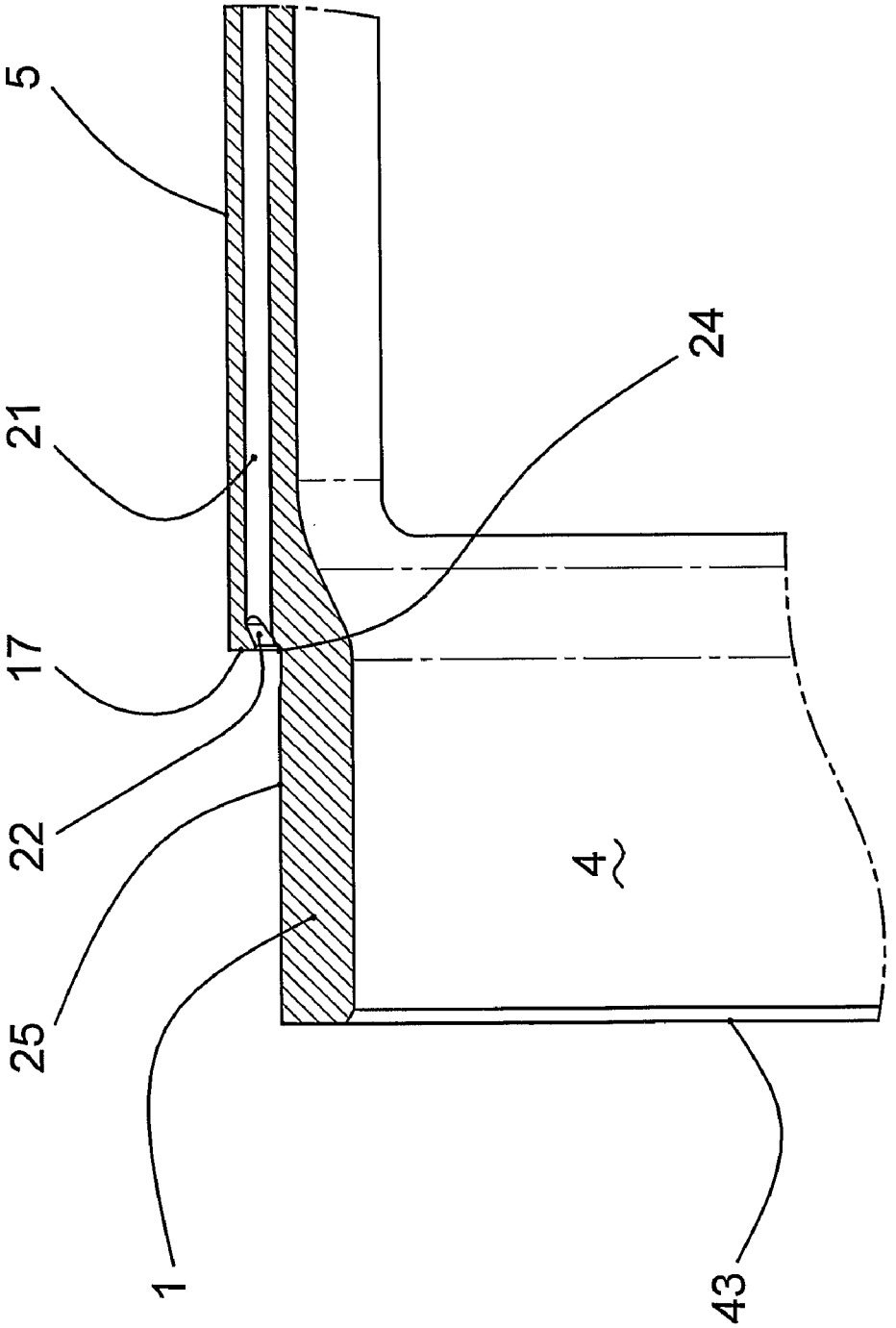


FIG. 5

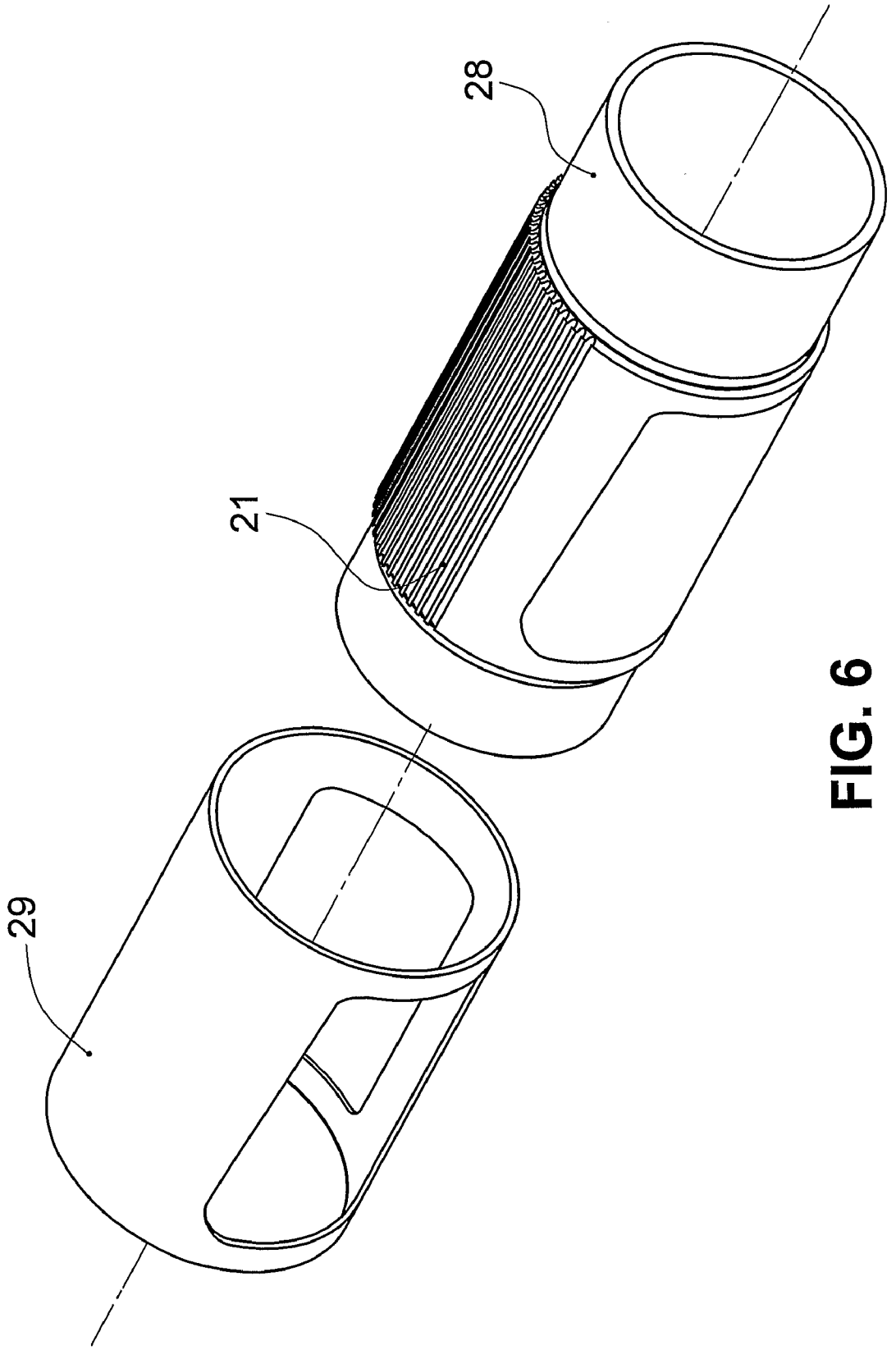


FIG. 6

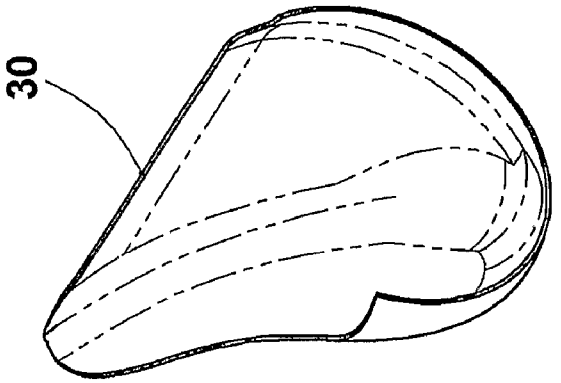


FIG. 8

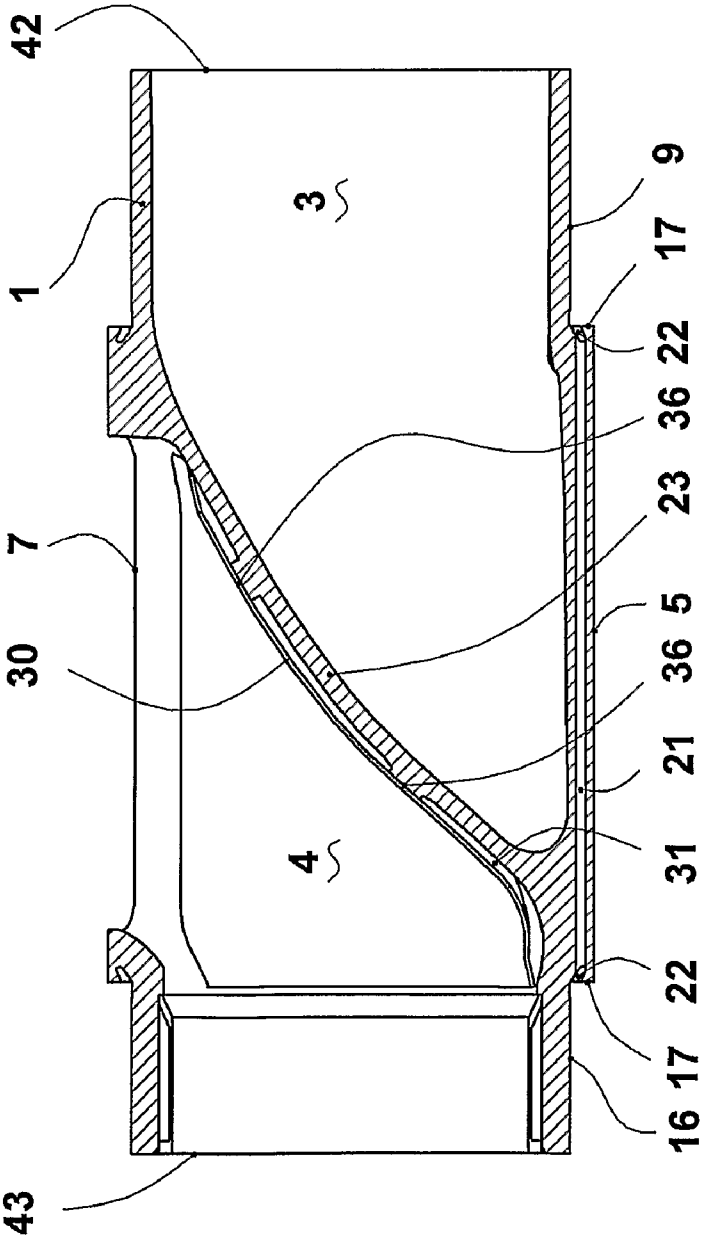


FIG. 7

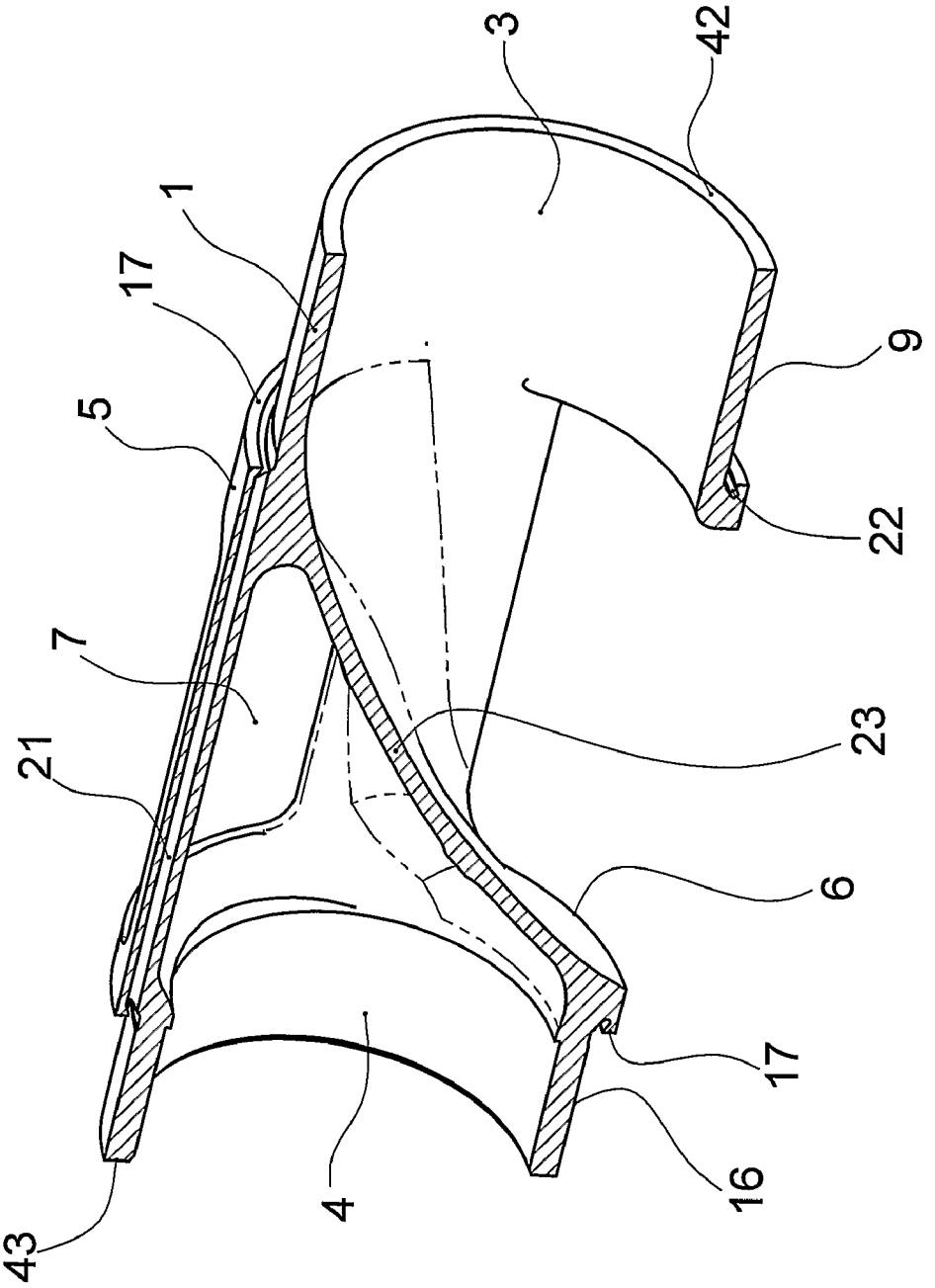


FIG. 9

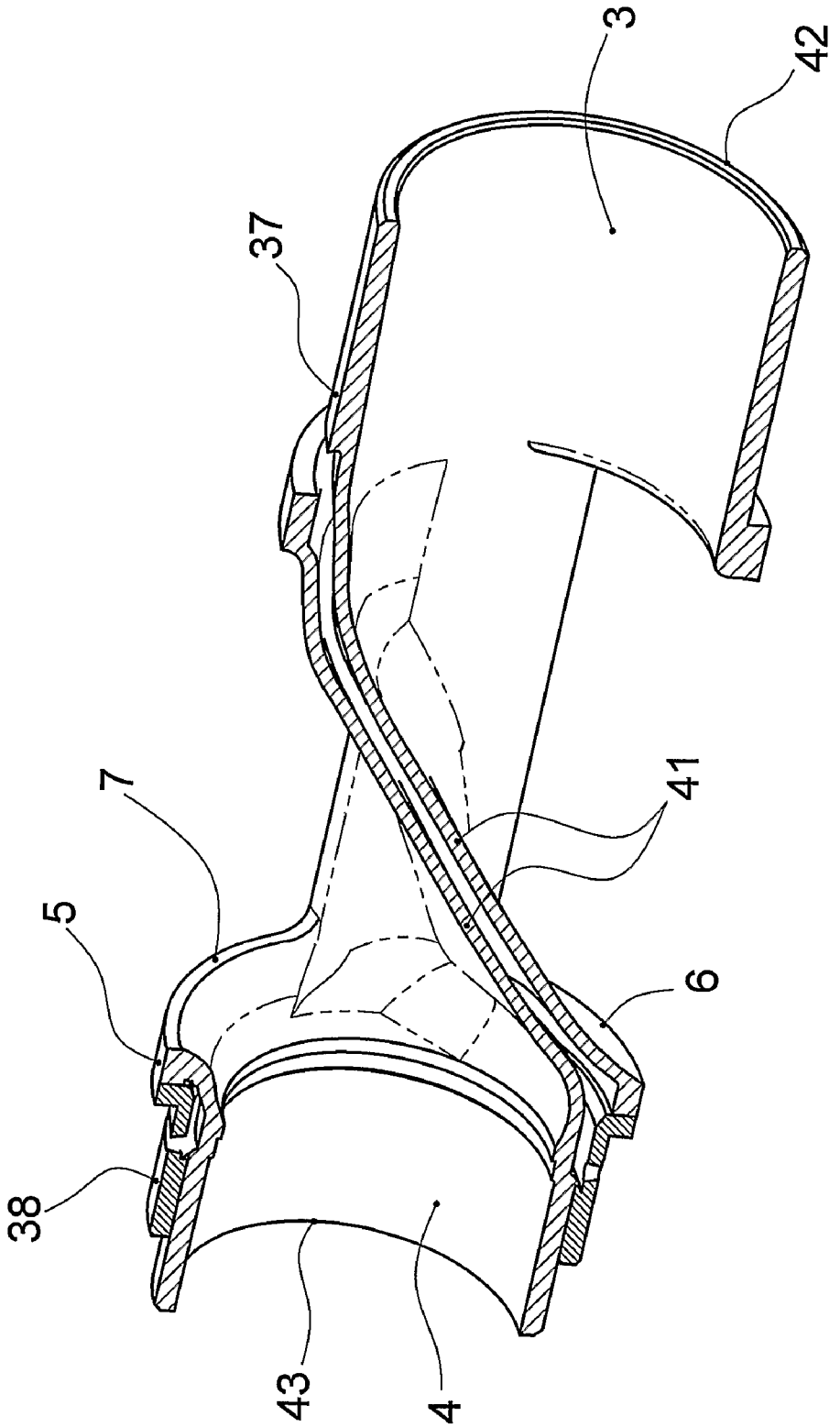


FIG. 10
Prior Art.

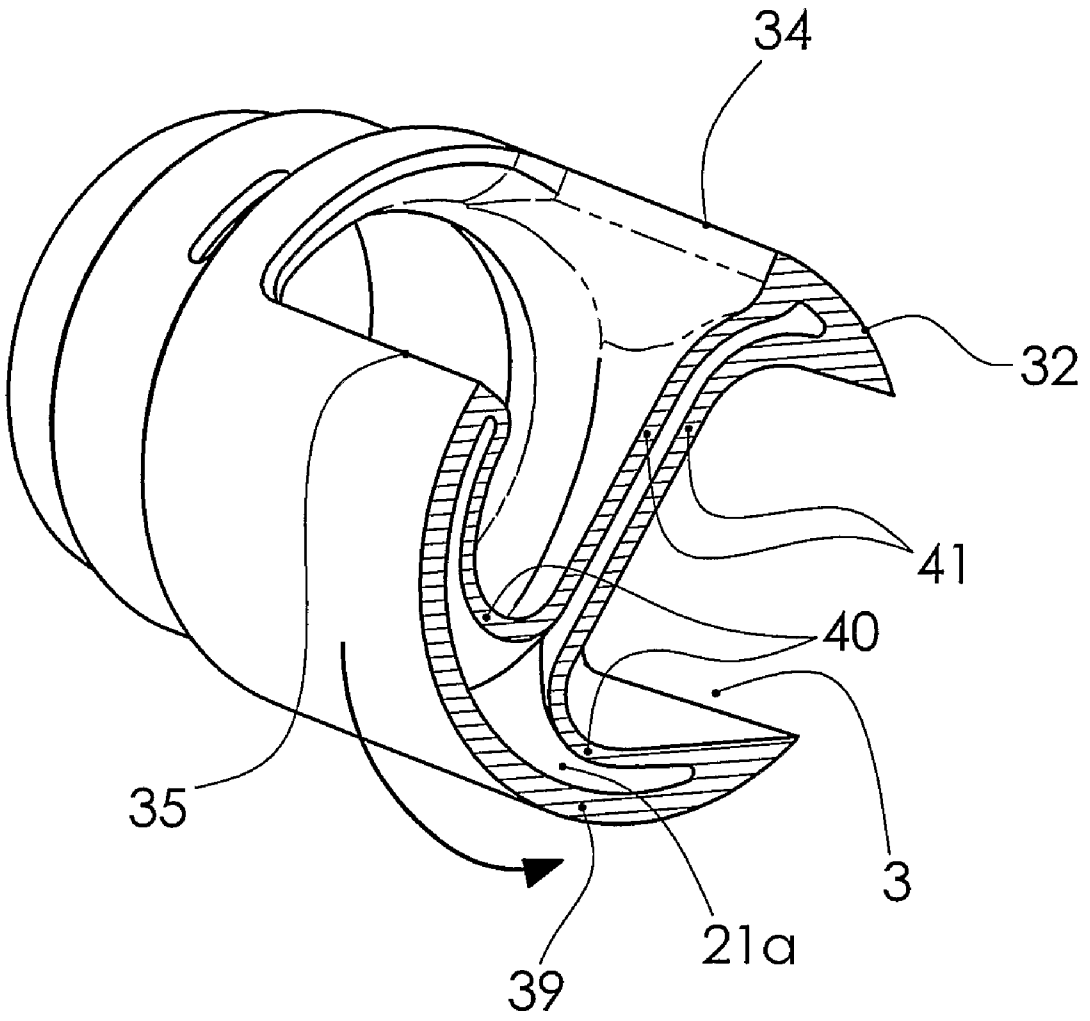


FIG. 11
Prior Art.

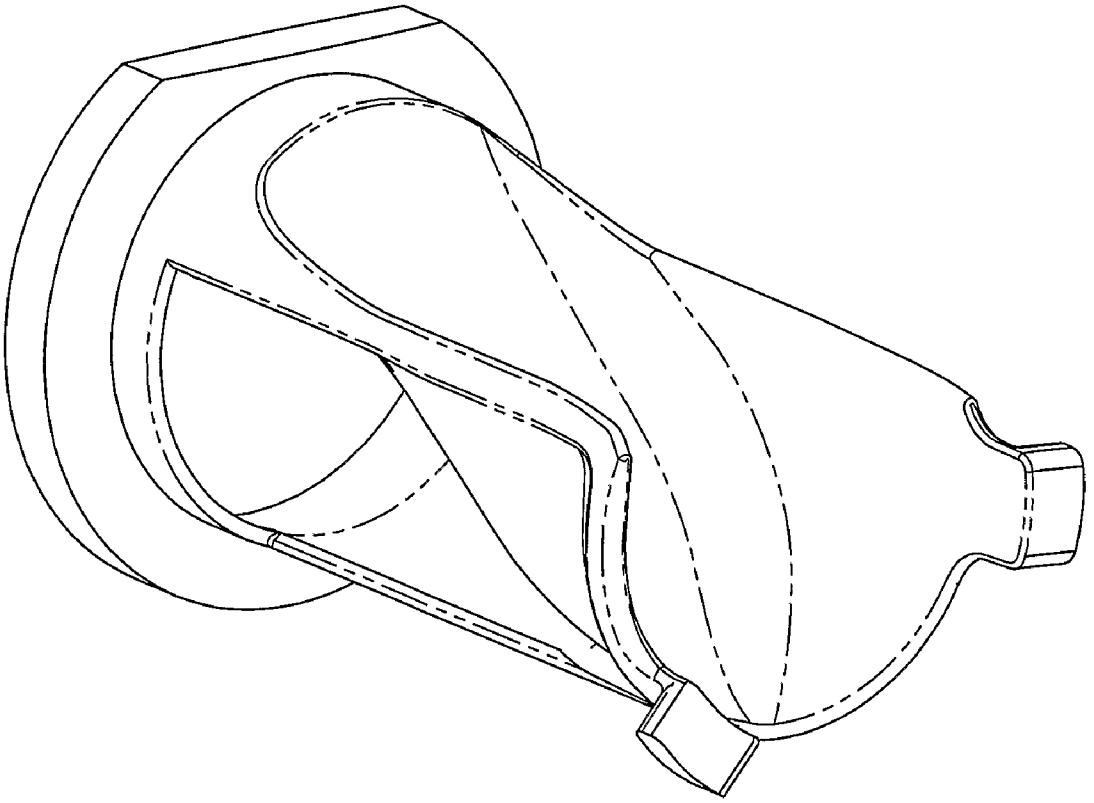


FIG. 12
Prior Art.

ROTARY VALVE CONSTRUCTION

TECHNICAL FIELD

[0001] The present invention relates to a valve for a rotary valve internal combustion engine and manufacture thereof. In particular it relates to axial flow rotary valves with internal cooling and incorporating both an inlet port and an exhaust port in the same valve.

BACKGROUND

[0002] The present invention relates to axial flow rotary valves incorporating an inlet and an exhaust port in the same valve. These ports terminate as openings in the periphery of the valve. During rotation these openings periodically align with a similar window in the cylinder head allowing the passage of gas from the valve to the cylinder and vice versa.

[0003] Rotary valves which run with a small predetermined clearance to the cylinder head bore were developed to address the problem of thermal and mechanical deflection of the valve, which had plagued rotary valve development for the most part of the 20th century. Rotary valves were traditionally sealed by the use of a close fitting stationary sleeve around the outer diameter of the valve. These sleeves function to gas seal the combustion chamber, and in the case of valves incorporating both inlet and exhaust ports to seal between the ports. These arrangements failed because they were unable to cope with the inevitable thermal and mechanical distortion of the valve which invariably resulted in seizure of the valve in the sleeve. Although many attempts were made to design sleeves that overcome this problem, none were commercialised.

[0004] In order to address these problems, the valve was designed to run with a small clearance to the cylinder head bore and the combustion chamber was sealed by means of a floating array of seals arranged around the window in the cylinder head. One of the first of these arrangements is found in U.S. Pat. No. 4,852,532 (Bishop). In this arrangement the valve incorporated both an inlet and exhaust port in the same valve. Flow of gas between the inlet and exhaust port was controlled by making the radial clearance between the valve and the cylinder head bore small. The combination of this small clearance and small pressures in the inlet and exhaust ports limit the flow of gases between the ports.

[0005] The existence of a small clearance between the outer diameter of the valve and the cylinder head bore enabled a small amount of thermal and mechanical distortion to occur without the valve making contact with the bore. This change was a major step forward in rotary valve design. However, in order to be able to capitalise on this advance valves needed to be developed that had minimal thermal and physical distortion. The smaller the distortion, the smaller is the required radial clearance and the better the sealing between the inlet and exhaust ports.

[0006] Thermal distortion generally results in a "permanent" distorted valve shape that rotates with the valve. If a thermally induced high point on the valve makes contact with the cylinder head bore the contact point will rotate with the valve generating considerable heat in the valve at the point of contact. This type of contact typically results in seizure of the valve in the cylinder head bore.

[0007] Axial flow rotary valves incorporating both inlet and exhaust ports in the same valve are the most demanding category of rotary valve from a thermal distortion perspective. At one end of the valve is the hot exhaust port at the other end is the cool inlet port and the periphery is part subject to combustion, part subject to exhaust flow and part subject to inlet flow. Further, the presence of two ports in the same valve dramatically reduces the space available for valve cooling passages.

[0008] Mechanical distortion occurs as a result of cylinder pressure acting on the periphery of the valve that is blocking the window during the compression and power strokes. The distortion is approximately proportional to the cylinder pressure. In the event that some operating conditions result in higher than normal cylinder pressure which deflects the valve into contact with the cylinder head bore, this contact will only be momentary as the peak cylinder pressure occupies only a very small portion of the cycle. In most cases the valve will be able to rub momentarily against the bore without seizure.

[0009] Thus unlike the case of mechanical distortion where there is latitude for the valve to make occasional contact with the cylinder bore, there is no latitude in the case of thermal distortion.

[0010] Cooling passages are an essential requirement for all rotary valves with the possible exception of those with very low power outputs. However, some prior art patent documents only show the rotary valve constructional and cooling details schematically, as these details were not relevant to the disclosure. A typical example of this is U.S. Pat. No. 5,509,386 (Wallis et al).

[0011] A rotary valve constructed similarly to that of U.S. Pat. No. 5,509,386 (Wallis et al) is shown in FIGS. **10** and **11** of the present specification. Important features of this valve are the complex cooling passages completely isolating the inlet and exhaust ports from each other and extending into the area adjacent the exhaust trailing edge **34** (refer FIG. **11**). FIG. **12** is an isometric view of the core necessary to manufacture the cooling passage in this valve by a casting process. This core is exceedingly complex with thin wall sections and can only be manufactured using ceramic, which is very fragile, expensive and difficult to remove from the casting. This valve is typical of previous prior art valves in that it is manufactured as a steel casting with cored internal cooling passages. In practise these valves are extremely difficult to cast and require use of the investment casting process which further increases the complexity and cost.

[0012] The constraints introduced by the necessity to adequately support the ceramic core during the casting process means the valve shown in FIGS. **10** and **11** must be manufactured in two pieces comprising cast valve **37** and collar **38**. In order to **30** improve the structural integrity of the valve, collar **38** is electron beam welded to cast valve **37** (refer to FIG. **10**).

[0013] U.S. Pat. No. 4,852,532 (Bishop) shows some details of the cooling passage of another prior art axial flow rotary valve. In this arrangement, as is common with this category of prior art valve, there are no cooling passages adjacent the exhaust trailing edge. During a substantial portion of the exhaust process, hot exhaust gas flows over the exhaust trailing edge. As a result this edge becomes extremely hot and suffers considerable local thermal distortion which damages the axial seals used to seal the com-

bustion gases in the cylinder. It is an essential requirement for all rotary valves that the exhaust trailing edge is cooled. Localised cooling of this area cannot be provided by means of a separate core as the cross-section of the core is very small and the area it has to span relatively long. This particular problem is addressed in the prior art valve shown in FIGS. **10** and **11** of the present specification by extending the core that separates the inlet and exhaust port, into the area adjacent the exhaust trailing edge **34**.

[**0014**] U.S. Pat. No. 2,158,386 (Sykes) shows cooling details of an axial flow rotary valve incorporating both inlet and exhaust ports in the same valve. Cooling is provided in a manner similar to that of the prior art rotary valve shown in FIGS. **10** and **11** of the present specification including cooling of exhaust trailing edge. The cooling passages in the valve shown in U.S. Pat. No. 2,158,386 (Sykes) are however much larger than those of the valve shown in FIGS. **10** and **11**, and therefore make the process of manufacturing the casting somewhat easier. However the problem with this valve is the very small ports that result from this approach. Both the inlet and exhaust ports have a diameter that is smaller than 50% of the valve outside diameter. By way of comparison, the valve shown in FIGS. **10** and **11** has an inlet port with a diameter that is 80% of the valve outer diameter and an exhaust port with a diameter that is 70% of the valve outside diameter. As the flow areas will vary as the square of the port diameter, the valve shown in U.S. Pat. No. 2,158,386 (Sykes) will have an inlet flow area of less than 40% that of a valve as shown in FIGS. **10** and **11** having the same outside diameter. The breathing capacity of such a valve would be so low as to render it inefficient compared to modern poppet valve technology.

[**0015**] This illustrates the problem facing manufacturers of axial flow rotary valves incorporating both inlet and exhaust ports in the same valve. The greater the cross-sectional areas allocated to the cooling passages the easier the casting, but the smaller port size and the lower the flow capacity of the valve. For rotary valves to be competitive with modern poppet valves they require large port sizes which minimises the allowable cross-section for the cooling passages, and makes the casting extremely difficult to manufacture and expensive. In addition these valves often require additional components welded onto the valve casting to give the valve adequate structural integrity.

[**0016**] This issue could potentially be addressed by increasing the outside diameter of the valve. However, this is not a realistic option as in rotary valve technology the diameter of the valve is the critical design issue as it determines the position of the spark plug and the shape of the combustion chamber. The greater the valve diameter the further the sparkplug must be located from the centre of the cylinder. From a combustion perspective the optimum spark plug position is the centre of the cylinder.

[**0017**] Prior art rotary valves of the type shown in FIGS. **10** and **11** of the present specification tend to be heavy. This in part arises because the thickness of the walls is determined by casting considerations which dictate the walls must be thicker than is required from a functional or structural point of view. This problem may be partially addressed by machining the internal port surfaces of the casting. However, this merely adds additional manufacturing operations and cost.

[**0018**] Finally, prior art rotary valves of the type shown in FIGS. **10** and **11** tend to be structurally inefficient with low stiffness to weight ratio. Prior art rotary valves such as that shown in U.S. Pat. No. 5,509,386 (Wallis et al) show axially disposed passages even if they were, as discussed previously, only schematic. The axial passage of this valve has a large circumferential length and there is connection between the inner and outer walls **40**, **39** (FIG. **11**) at the circumferential ends of this axial passage only. This is the source of its structural inefficiency.

[**0019**] The rotary valve in FIG. **11** has an axial passage **21***a* extending circumferentially from adjacent the exhaust port leading edge **35** to the inlet port **3**. The resulting structure formed by the outer wall **39**, axial passage **21***a* and inner walls **40** of the valve is inherently flexible as the inner and outer walls are not tied together can therefore move independently of each other. This arrangement results in a very poor stiffness to weight ratio.

[**0020**] This low stiffness problem can be best illustrated when one considers how movement of port dividing walls **41** towards outer wall **39** is reacted in the prior art valve shown in FIG. **11**. Movement of port dividing walls **41** are reacted by inner walls **40** which act as cantilever beams anchored remotely at the points where inner walls **40** meet outer walls **39**. Consequently, dividing walls **41** and inner walls **40** act like a bellows pivoting about their points of attachment to inner walls **40** and outer wall **39** respectively.

[**0021**] The thinner the outer and/or inner wall the greater the problem created by this circumferentially long axial passage. A convenient method of quantifying this issue is aspect ratio. For the purposes of this application the aspect ratio of an axially extending passage is defined as the maximum circumferential length of the passage divided by the minimum radial height between the radial extremities of the axial passage and the adjacent surface of either the inner or outer wall, which ever is the closer.

[**0022**] In the rotary valve shown in FIG. **11** the radial wall thickness of outer wall **39** is 3.5 mm, the radial wall thickness of inner wall **40** is 2 mm and the circumferential length of axial passage **21***a* is 70 mm giving axial passage **21***a* an aspect ratio of thirty five to one (35:1).

[**0023**] There are several issues that must be considered when designing a rotary valve. Firstly, the outside diameter of the valve must be a minimum consistent with the required minimum flow area in both the inlet and exhaust ports. Secondly, the valve must be sufficiently stiff such that under maximum combustion load the deflection is less than the predetermined small clearance. Thirdly, the valve cooling and the valve construction must be such to ensure that the thermal distortion is less than the predetermined small clearance. Finally, the valve cooling and construction must be such that valve distortion due to mechanical load does not add to the thermal distortion. Otherwise, the predetermined small clearance would have to be larger than necessary.

[**0024**] The present invention seeks to overcome one or more of the disadvantages associated with the abovementioned prior art rotary valves that have internal cooling passages cast therein.

SUMMARY OF INVENTION

[**0025**] In a first aspect, the present invention consists of an axial flow rotary valve for an internal combustion engine,

said valve being adapted to rotate about an axis within a bore in the cylinder head of said engine, one end of said valve being an inlet end and the other end being an exhaust end, said valve comprising a centre portion having a cylindrical periphery, an inlet portion smaller in diameter than said centre portion, extending between said centre portion and said inlet end, an exhaust portion smaller in diameter than said centre portion, extending between said centre portion and said exhaust end, an inlet port extending from said inlet end and terminating as an inlet opening in the periphery of said centre portion, and an exhaust port extending from said exhaust end and terminating as an exhaust opening in the periphery of said centre portion, characterised in that said centre portion has at least one elongate passage communicating between said inlet portion and said exhaust portion, said passage being disposed circumferentially outside said openings and radially outside at least one of said inlet and exhaust portions.

[0026] Preferably, said passage has an aspect ratio smaller than 10:1.

[0027] Preferably, at least one end of said centre portion has an annular valve seat extending radially inwards from said cylindrical periphery and at least a portion of said passage is radially outside the inside diameter of said annular valve seat.

[0028] Preferably, said centre portion has at least one annular cavity, axially outside said inlet and exhaust openings, having an annular opening radially inside of said annular valve seat, and an end of said passage terminates at said cavity.

[0029] Preferably, said at least one elongate passage comprises a plurality of circumferentially spaced apart elongate passages.

[0030] Preferably, at least one of said passages is adjacent the trailing edge of said exhaust opening.

[0031] Preferably, each of said passages has a substantially circular section and extends in a substantially axial direction.

[0032] Preferably, said inlet and exhaust ports are separated by a common wall having no internal passages.

[0033] Preferably, said elongate passage is a coolant passage.

[0034] In one preferred embodiment, said valve is manufactured from a single solid piece of material.

[0035] In another preferred embodiment, said valve is manufactured from an inner body and at least one outer body, said outer body at least partially surrounding said inner body and attached thereto such that said passage is formed at the interface between said bodies.

[0036] In a second aspect, the present invention consists of a method of manufacturing a rotary valve in accordance with the first aspect of the present invention, from an unfinished valve having an oversize centre portion stepping radially inwards at both ends, comprising the steps of:

[0037] a. machining said oversize centre portion to form said elongate passage and an annular groove in the end face of at least one end of said oversize centre portion;

[0038] b. deforming said end of said oversize centre portion radially inwards such that said annular groove becomes said annular cavity; and

[0039] c. machining said end face to form said annular valve seat and machining the outside diameter of said oversize centre portion to form said centre portion.

[0040] In a third aspect, the present invention consists of a rotary valve assembly for an internal combustion engine, comprising a rotary valve in accordance with the first aspect of the invention, a cylinder head having a bore in which said valve is supported by bearing means and rotates with a small predetermined clearance between said bore and said centre portion of said valve, said inlet and exhaust openings of said valve periodically communicating with a window in said bore, and first and second sealing rings flexibly sealed to said bore and biased axially inwards against first and second valve seats respectively formed by opposite ends of said centre portion.

[0041] Preferably, said rotary valve assembly further comprising a heat insulation barrier in the exhaust port of said valve covering at least a portion of the common wall separating said inlet and exhaust ports of said valve.

[0042] Preferably, there is an air gap between said insulation barrier and said common wall.

BRIEF DESCRIPTION OF DRAWINGS

[0043] FIG. **1** is a sectional view of an axial flow rotary valve internal combustion engine in accordance with a first preferred embodiment of the present invention.

[0044] FIG. **2** is a partial cross sectional view along II-II of the engine depicted in FIG. **1** where the clearance between the valve and the cylinder head bore has been exaggerated for clarity.

[0045] FIG. **3** is a partial cross sectional view through an axial flow rotary valve in the early stages of manufacture according to the present invention.

[0046] FIG. **4** is a partial cross sectional view as in FIG. **3** but with the valve at a later stage of manufacture.

[0047] FIG. **5** is a partial cross sectional view as in FIG. **3** and FIG. **4** but with the valve at the end of manufacture.

[0048] FIG. **6** is an isometric view of a rotary valve according to a second preferred embodiment of the present invention, manufactured from two pieces.

[0049] FIG. **7** is a cross sectional view through a rotary valve showing details of a heat insulation method for the exhaust port according to the present invention.

[0050] FIG. **8** is an isometric view of the sheet metal baffle used to insulate the common wall and the exhaust port of the valve shown in FIG. **7**.

[0051] FIG. **9** is an isometric axial sectional view through the centre of the rotary valve shown in FIG. **1**.

[0052] FIG. **10** is an isometric axial sectional view through the centre of a prior art rotary valve.

[0053] FIG. **11** is an isometric transverse sectional view through the centre of the prior art rotary valve shown in FIG. **10**.

[0054] FIG. **12** is an isometric view of the core required to manufacture the cooling passages in the prior art valve shown in FIGS. **10** and **11**.

BEST MODE OF CARRYING OUT THE INVENTION

[0055] FIG. **1** depicts a first embodiment of a rotary valve assembly according to the present invention comprising a valve **1** and a cylinder head **2**. Valve **1** has an inlet port **3** and an exhaust port **4**. Valve **1** has a centre portion **5** of constant diameter with an inlet portion **9** of reduced diameter on one side and exhaust portion **16** of reduced diameter on the other side. Inlet portion **9** extends between centre portion **5** and the inlet end **42** of valve **1**. Exhaust portion **16** extends between centre portion **5** and the exhaust end **43** of valve **1**. Inlet port **3** extends from inlet end **42** of valve **1** and terminates at inlet opening **6** in the periphery of centre portion **5**. Exhaust port **4** extends from exhaust end **43** and terminates at exhaust opening **7** in the periphery of centre portion **5**. Inlet port **3** and exhaust port **4** are separated by a common wall **23**. Valve **1** is supported by bearings **8** to rotate about axis **10** in cylinder head **2**. Bearings **8** support the periphery of inlet portion **9** and exhaust portion **16**. Bearings **8** allow valve **1** to rotate about axis **10** whilst maintaining a small running clearance between the periphery of centre portion **5** and bore **11** of cylinder head **2**.

[0056] Cylinder head **2** is mounted on top of cylinder block **12**. Piston **13** reciprocates in cylinder **14**. As valve **1** rotates in cylinder head **2**, inlet opening **6** and exhaust opening **7** periodically communicate with window **15** in cylinder head **2**, allowing the passage of fluids between valve **1** and cylinder **14**.

[0057] Valve cooling and lubricating oil is prevented from entering the zone between the periphery of centre portion **5** and the bore **11** by two face seal arrangements comprising valve sealing rings **18**, O-rings **20**, annular valve seats **17** and face seal springs **19**. Centre portion **5** extends axially a small distance past the axial extremities of window **15**. Valve **1** steps radially inward either side of centre portion **5** forming radial faces that form valve seats **17** against which valve sealing rings **18** are axially inwardly preloaded by face seal springs **19**. Valve sealing rings **18** are slidingly sealed against cylinder head bore **11** by means of O-rings **20**.

[0058] A plurality of small axially extending elongate axial passages **21** allow cooling oil to pass through valve **1** close to the surface of centre portion **5**. Passages **21** extend axially beyond the axial extremities of inlet opening **6** and exhaust opening **7**, and communicate between inlet portion **9** and exhaust portion **16**. Cooling oil enters and exits the centre portion **5** through the annular gaps between the inside diameter of valve seats **17** and the radially adjacent valve surfaces **24**. Immediately axially inboard of valve seats **17** are annular cavities **22** providing access to axial passages **21**. Axial passages **21** terminate at annular cavities **22**, and each annular cavity **22** has an annular opening radially inside of valve seats **17**. Annular cavities **22** are axially outside of inlet opening **6** and exhaust opening **7**. FIG. **9** shows an isometric sectional view of valve **1**.

[0059] Axial passages **21** have a small aspect ratio. In the event axial passages **21** are formed by drilled holes, typically the hole diameter is 2 mm and the radial wall thickness between the radial extremity of the hole and the adjacent surface is 1.5 mm giving axial passages **21** an aspect ratio of 1.33 (ie. 2 mm/1.5 mm). This ensures that the inner and outer walls are always effectively tied together and act as a very efficient beam. Unlike the prior art valves any movement of the common wall **23** is directly reacted by both the inner and outer walls.

[0060] Referring to FIG. **2**, axial passages **21** are approximately equally circumferentially spaced apart and are located at a constant radius from axis **10** close to the surface of cylindrical portion **5**. Axial passages **21** are disposed circumferentially outside inlet opening **6** and exhaust opening **7**, including several of axial passages **21** being located in the bridge **32** of valve **1**. Bridge **32** is formed by the portion of the periphery of centre portion **5** that spans between the inlet opening leading edge **33** and the exhaust opening trailing edge **34**. Axial passages **21** are wholly disposed radially outside the extremities of both the periphery of inlet portion **9** and the periphery of exhaust portion **16**. However, in other not shown embodiments, the axial passages may be radially outside the extremities of only one of the inlet or the exhaust portions. It is necessary for the axial passages to be radially outside at least one of the inlet or exhaust portions so that access is available to machine the axial passages.

[0061] Axial passages **21** are small in diameter and are typically less than 2 mm diameter. As there is no known means of casting such passages they must be machined into valve **1**. Axial passages **21** must be close to the surface of cylindrical portion **5** to minimize the heat path between the surface of the valve subject to heat input from the combustion chamber and axial passages **21**.

[0062] The two walls separated by a cooling passage found in conventional arrangements are replaced by a single wall with axial passages **21** running though the centre of this wall. Conventional arrangements using production style processes would typically require a radial thickness in the order of 11 mm (two 3 mm walls separated by a 5 mm thick core) compared to 4 to 5 mm in the case of the present embodiment. This potentially reduces the required valve outer diameter by 12 mm or more.

[0063] Axial passages **21** can easily be provided adjacent exhaust trailing edge **34**. In prior art arrangements this was only possible by extending the cooling passage between the inlet and exhaust port. A single common wall **23** separates the inlet port **3** from the exhaust port **4**, replacing the typical prior art arrangement of two walls separated by an internal cooling passage with a consequent saving of material and weight.

[0064] The requirement to position axial passages **21** close to the surface of centre portion **5** is in conflict with the requirement to form valve seats **17** of adequate radial depth at both ends of centre portion **5**. Preferably, the radial extremities of axial passages **21** are located radially within 1.5 mm of the periphery of centre portion **5**. Typically, valve seats **17** require a radial depth greater than 3 mm to function satisfactorily as face seals. As such, at least a portion of each axial passage **21** is radially outside the inside diameter of valve seats **17**, as can be seen in FIG. **1**. Consequently any axial passages **21** machined directly into valve **1** using conventional manufacturing techniques would penetrate valve seats **17** against which valve sealing ring **18** is preloaded thus preventing proper sealing. In accordance

with the present invention, FIGS. **3** to **5** illustrate a method of manufacturing valve seats **17** and axial passages **21** that overcomes this difficulty.

[0065] FIG. **3** shows a partial section through unfinished valve **1**. At this stage of manufacture, centre portion **5** has not yet been finished machined and its outside diameter is larger than its final diameter. An annular groove **44** is machined in each end face of centre portion **5**. Axial passages **21**, terminating at grooves **44**, may be machined through centre portion **5** before or after machining grooves **44**. At this stage of manufacture, axial passages **21** are some distance from the oversized outside diameter of centre portion **5** of valve **1** which still has a machining allowance remaining on its diameter. After machining grooves **44** and axial passages **21**, the outside diameter of unfinished valve **1** is deformed inwardly at both ends of centre portion **5** such that grooves **44** become annular cavities **22** adjacent the ends of the axial passages **21**, as shown in FIG. **4**. The ends of centre portion **5** may be deformed by rolling, swaging or any other suitable process. The outside diameter of centre portion **5** is then machined to final size and the ends of centre portion **5** are machined to form valve seats **17**, as shown in FIG. **5**.

[0066] Using this manufacturing process, axial passages **21** can be machined into valve **1** radially outboard of the inner diameter of valve seats **17** whilst providing sufficient radial thickness for valve seats **17** to ensure correct functioning of the face seal.

[0067] The radial gap between the inside diameter of valve seat **17** and its radially adjacent surface **24** forms a coolant opening to cavities **22**. This gap must be sufficient to ensure the required oil flow to the axial passages **21** can pass to and from cavities **22** without excessive pressure drop. The diameter of the radial adjacent surface **24** determines the size of inlet port **3** and exhaust port **4**, and therefore the breathing capacity of valve **1**. The maximum allowable port diameter adjacent this surface **24** is the diameter of the radial adjacent surface **24**, minus twice the minimum allowable wall thickness.

[0068] Bearing surface **25** is the surface against which the rolling elements of bearings **8** roll and is typically designed to be no smaller in diameter than that of the radial adjacent surface **24**, as this would unnecessarily reduce the size of inlet port **3** and exhaust port **4**. The diameter of bearing surface **25** cannot be larger than the inner diameter of the valve sealing ring **18** as it must be able to be assembled over this diameter.

[0069] For a given valve diameter, a given required radial depth of the valve seat **17** and a given cooling oil flow this arrangement produces rotary valves with the greatest possible flow areas in both inlet port **3** and exhaust port **4**.

[0070] The number of axial passages **21** and their distribution depends on design details of the engine and its application. Their number and distribution are determined by two considerations. Firstly, they should be positioned to maximise the heat removal from the hottest areas on the valves periphery. Secondly, they should be positioned to minimise the thermal distortion of the valve. The optimum arrangement is generally established experimentally.

[0071] Certain areas of the periphery of centre portion **5** are subjected to greater heat loads than others. For example the portion of the periphery exposed to combustion sees a greater heat load than the portion of the periphery exposed to the combustion chamber during compression. Bridge **32** adjacent to exhaust trailing edge **34** is also subject to high heat load whilst the area of bridge **32** adjacent to leading inlet leading edge **33** has very little heat load. In general axial passages **21** are positioned adjacent those areas on the surface of the valve subject to high heat load. In some applications where weight is particularly important, as many axial passages as possible may be provided to minimise the valve weight. Some of these axial passages **21** may subsequently be blocked to isolate the cooling oil flows to those areas discussed above.

[0072] Minimising the number of axial passages **21** that have cooling oil flowing through them maximises the velocity of the cooling oil flow through the axial passages **21** for a given oil flow through the valve. This higher velocity improves the ability of the oil to cool the valve surface. Distributing the axial passages **21** that carry cooling oil as described above ensures heat is more efficiently removed from those surfaces with high heat input, for a given cooling oil flow.

[0073] In general the heat input to surface **26** (refer to FIG. **2**) of valve **1**, which is exposed to compression and combustion, is greater than the heat input to bridge **32**. Consequently the surface temperature of surface **26** is greater, and all other things being equal, the valve will tend to bow in a convex shape towards surface **26**. All things are not however equal. Common wall **23** between inlet port **3** and exhaust port **4** is directly exposed to the exhaust gas on the side of exhaust port **4**. Common wall **23** has no internal cooling and consequently has the capacity to become very hot. This hot common wall **23** expands and thermally distorts valve **1** in a manner that is difficult to predict.

[0074] Exhaust port **4** has some surfaces directly exposed to axial passages **21**. These axial passages **21** are subject to heat input from both the outer surface of the valve and the inner surface of exhaust port **4** and are subject to a disproportionate heat load.

[0075] In some arrangements where the heat load from the exhaust is high or the predetermined small clearance is small or a combination of both these, it is necessary to limit the heat input to common wall **23** and to axial passages **21** from exhaust port **4**. This can be achieved by various methods. Walls of the exhaust port **4** can be sprayed with an insulating layer. Alternatively, a heat barrier in the form of a sheet metal baffle **30** (refer to FIG. **8**) can be inserted into exhaust port **4**, which provides an air gap **31** (refer to FIG. **7**) between the hot exhaust gases and the walls of exhaust port **4**. Small raised pads **36** may be provided on the surface of exhaust port **4** to prevent baffle **30** touching the walls of exhaust port **4** except at these pads **36**.

[0076] By careful radial positioning of axial passages **21** carrying coolant close to the outside diameter of centre portion **5**, and by insulating the internal surfaces of exhaust port **4** from the exhaust gas, valve distortion can be maintained at a level which can be accommodated by a predetermined small clearance, whose size does not adversely affect the engine's performance, even under the most arduous conditions.

[0077] From a mechanical stiffness aspect it has been found that this design has a high stiffness to weight ratio.

The low aspect ratio of axial passages **21** ensures that common wall **23** effectively ties the outer surfaces on opposite sides of valve **1** together much more effectively than if the outer surfaces of valve **1** were separated from the common wall by a cooling passage.

[**0078**] Axial cooling passages **21** are limited to an area immediately radially adjacent the outside diameter of valve **1**. This arrangement has the great advantage that it can be manufactured from a casting into which the axial cooling passages **21** are subsequently machined. Such a casting is very easy to cast due to the absence of complex internal cooling passages. This arrangement has the added benefit that such valves may be machined from a solid billet of material. This is particularly important in the manufacture of prototype valves where even simple casting require long lead times.

[**0079**] By placing axial passages **21** immediately radially outboard of the peripheries of inlet portion **9** and exhaust portion **16**, immediately radially inboard of periphery of centre portion **5**, and embedding them in the centre of the valve's outside wall, the diameter and weight of the valve **1** is minimised for any given port flow area and its stiffness maximised. By deforming the outer diameter of valve **1** during manufacture an adequate valve seat **17** may be obtained without having to increase the valve outside diameter. Finally by appropriate placement of the axial passages carrying coolant and insulation of the exhaust port a valve with minimal distortion is obtained.

[**0080**] However, axial passages **21** are relatively expensive to machine due to their small diameter and long length. This issue is addressed by the second embodiment of the present invention shown in FIG. **6** whereby valve **1** is manufactured from several pieces welded together. Axial passages **21** are formed in the outer diameter of valve inner body **28**. In this case axial passages **21** are no longer necessarily round in cross-section or axially extending. Axial passages **21** may be replaced by passages that extend between the exhaust portion and the inlet portion but follow a diagonal, curvilinear or other path. Axial passages **21** are no longer confined by manufacturing considerations to the very low aspect ratios of those valves with machined holes. However, they should maintain a low aspect ratio to ensure structural efficiency.

[**0081**] Valve outer body **29** is shrunk over valve inner body **28** and welded to valve inner body **28**. At a minimum, valve outer body **29** must be welded to valve inner body **28** adjacent to inlet opening **6** and exhaust opening **7** in centre portion **5**, to ensure that cooling oil does not leak into inlet port **3** or exhaust port **4**. Additional welding of valve outer body **29** to the valve inner body **28** in the zone between the axial cooling passages will stiffen up the completed valve. In this arrangement, valve inner body **28** may for example be produced as a casting and valve outer body **29** may for example be manufactured from extruded tube.

[**0082**] If the minimum valve diameter is to be achieved the outer diameter of the valve will need to be deformed and machined in a similar fashion to the first embodiment described earlier with reference to FIGS. **4** and **5**, after valve outer body **29** has been attached to valve inner body **28**.

[**0083**] In both of the abovementioned embodiments it should be understood that "small aspect ratio" for axial passages **21**, is defined as one that is less than ten to one (10:1).

[**0084**] The term "comprising" as used herein is used in the inclusive sense of "including" or "having" and not in the exclusive sense of "consisting only of".

1. An axial flow rotary valve for an internal combustion engine, said valve being adapted to rotate about an axis within a bore in the cylinder head of said engine, one end of said valve being an inlet end and the other end being an exhaust end, said valve comprising a centre portion having a cylindrical periphery, an inlet portion smaller in diameter than said centre portion, extending between said centre portion and said inlet end, an exhaust portion smaller in diameter than said centre portion, extending between said centre portion and said exhaust end, an inlet port extending from said inlet end and terminating as an inlet opening in the periphery of said centre portion, and an exhaust port extending from said exhaust end and terminating as an exhaust opening in the periphery of said centre portion, characterised in that said centre portion has at least one elongate passage communicating between said inlet portion and said exhaust portion, said passage being disposed circumferentially outside said openings and radially outside at least one of said inlet and exhaust portions.

2. A rotary valve as claimed in claim 1 wherein said passage has an aspect ratio smaller than 10:1.

3. A rotary valve as claimed in claim 1 wherein at least one end of said centre portion has an annular valve seat extending radially inwards from said cylindrical periphery and at least a portion of said passage is radially outside the inside diameter of said annular valve seat.

4. A rotary valve as claimed in claim 3 wherein said centre portion has at least one annular cavity, axially outside said inlet and exhaust openings, having an annular opening radially inside of said annular valve seat, and an end of said passage terminates at said cavity.

5. A rotary valve as claimed in claim 1 wherein said at least one elongate passage comprises a plurality of circumferentially spaced apart elongate passages.

6. A rotary valve as claimed in claim 5 wherein at least one of said passages is adjacent the trailing edge of said exhaust opening.

7. A rotary valve as claimed in claim 5 wherein each of said passages has a substantially circular section and extends in a substantially axial direction.

8. A rotary valve as claimed in claim 1 wherein said inlet and exhaust ports are separated by a common wall having no internal passages.

9. A rotary valve as claimed in claim 1 wherein said elongate passage is a coolant passage.

10. A rotary valve as claimed in claim 1 wherein said valve is manufactured from a single solid piece of material.

11. A rotary valve as claimed in claim 1 wherein said valve is manufactured from an inner body and at least one outer body, said outer body at least partially surrounding said inner body and attached thereto such that said passage is formed at the interface between said bodies.

12. A method of manufacturing a rotary valve as claimed in claim 4, from an unfinished valve having an oversize centre portion stepping radially inwards at both ends, comprising the steps of:

a. machining said oversize centre portion to form said elongate passage and an annular groove in the end face of at least one end of said oversize centre portion;

b. deforming said end of said oversize centre portion radially inwards such that said annular groove becomes said annular cavity; and

c. machining said end face to form said annular valve seat and machining the outside diameter of said oversize centre portion to form said centre portion.

13. A rotary valve assembly for an internal combustion engine, comprising a rotary valve as claimed in claim 1, a cylinder head having a bore in which said valve is supported by bearing means and rotates with a small predetermined clearance between said bore and said centre portion of said valve, said inlet and exhaust openings of said valve periodically communicating with a window in said bore, and first and second sealing rings flexibly sealed to said bore and biased axially inwards against first and second valve seats respectively formed by opposite ends of said centre portion.

14. A rotary valve assembly as claimed in claim 13 further comprising a heat insulation barrier in the exhaust port of said valve covering at least a portion of the common wall separating said inlet and exhaust ports of said valve.

15. A rotary valve assembly as claimed in claim 13 wherein there is an air gap between said insulation barrier and said common wall.

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