(12) INTERNATIONAL APPLICATION PUBLISHED UNDER THE PATENT COOPERATION TREATY (PCT)

(19) World Intellectual Property Organization
International Bureau

(43) International Publication Date
19 January 2017 (19.01.2017)

WIPO | PCT

(10) International Publication Number
WO 2017/008683 A1

(51) International Patent Classification:
F01D 17/16 (2006.01)

(21) International Application Number:
PCT/CN2016/089146

(22) International Filing Date:
7 July 2016 (07.07.2016)

(25) Filing Language: English

(26) Publication Language: English

(30) Priority Data:
14/796,798 10 July 2015 (10.07.2015) US

(71) Applicant: **KANGYUE TECHNOLOGY CO., LTD** [CN/CN]; No.1 Luoqian Street, Shouguang Development Zone, Shouguang, Shandong 262700 (CN).

(72) Inventors: **ARNOLD, Steven Don**; 3351 Deluna Dr. Rancho, Palos Verdes, CA 90275 (US). **WANG, Hang**; Wenhuamingyuan 5-302, Wenmiao street, Shouguang, Shandong 262700 (CN). **LI, Yanzhao**; Room 401, Building 5, Kangyue Yuan, Shouguang Development Area, Shouguang, Shandong 262718 (CN). **XIN, Xiaofen**; Building 12, Chunleixiaoqu, Shouguang, Shandong 262700 (CN).

(74) Agent: **AFD CHINA INTELLECTUAL PROPERTY LAW OFFICE**; Suite B 1601A, 8 Xue Qing Rd.Haidian, Beijing 100192 (CN).

(81) Designated States *(unless otherwise indicated, for every kind of national protection available)*: AE, AG, AL, AM, AO, AT, AU, AZ, BA, BB, BG, BH, BN, BR, BW, BY, BZ, CA, CH, CL, CN, CO, CR, CU, CZ, DE, DK, DM, DO, DZ, EC, EE, EG, ES, FI, GB, GD, GE, GH, GM, GT, HN, HR, HU, ID, IL, IN, IR, IS, JP, KE, KG, KN, KP, KR, KZ, LA, LC, LK, LR, LS, LU, LY, MA, MD, ME, MG, MK, MN, MW, MX, MY, MZ, NA, NG, NI, NO, NZ, OM, PA, PE, PG, PH, PL, PT, QA, RO, RS, RU, RW, SA, SC, SD, SE, SG, SK, SL, SM, ST, SV, SY, TH, TJ, TM, TN, TR, TT, TZ, UA, UG, US, UZ, VC, VN, ZA, ZM, ZW.

(84) Designated States *(unless otherwise indicated, for every kind of regional protection available)*: ARIPO (BW, GH, GM, KE, LR, LS, MW, MZ, NA, RW, SD, SL, ST, SZ, TZ, UG, ZM, ZW), Eurasian (AM, AZ, BY, KG, KZ, RU, TJ, TM), European (AL, AT, BE, BG, CH, CY, CZ, DE, DK, EE, ES, FI, FR, GB, GR, HR, HU, IE, IS, IT, LT, LU, LV, MC, MK, MT, NL, NO, PL, PT, RO, RS, SE, SI, SK, SM, TR), OAPI (BF, BJ, CF, CG, CI, CM, GA, GN, GQ, GW, KM, ML, MR, NE, SN, TD, TG).

Declarations under Rule 4.17:
— *of inventorship (Rule 4.17(iv))*

Published:
— *with international search report (Art. 21(3))*

(54) **Title**: BALANCED VANES AND INTEGRATED ACTUATION SYSTEM FOR A VARIABLE GEOMETRY TURBOCHARGER

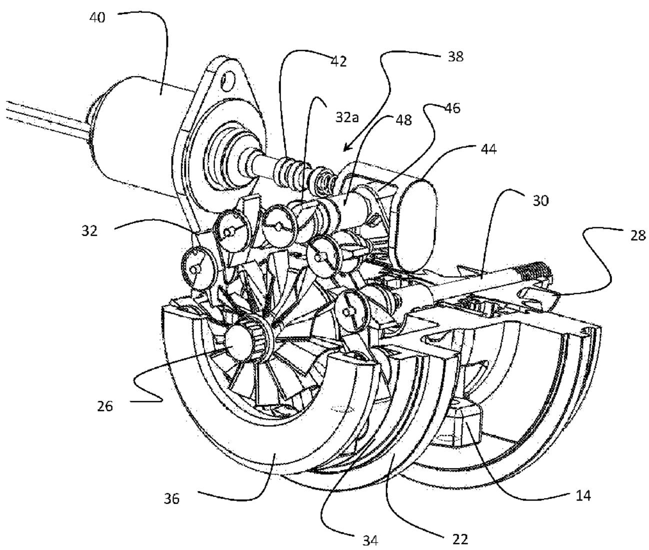

FIG. 2

(57) **Abstract**: Aa variable geometry turbine (VGT) for turbochargers incorporates a turbine nozzle having a plurality of vanes, each vane having an airfoil with an inner disc and an outer disc. A first nozzle plate incorporates pockets to receive the inner discs with the inner discs substantially flush with a nozzle surface on the first nozzle plate. A second nozzle plate has pockets to receive the outer discs with outer discs substantially flush with a second nozzle surface. An integral actuation system rotates the plurality of vanes for variation of the nozzle geometry.

WO 2017/008683 A1

BALANCED VANES AND INTEGRATED ACTUATION SYSTEM FOR A VARIABLE GEOMETRY TURBOCHARGER

BACKGROUND OF THE INVENTION

Field of the Invention

[Para 1] This invention relates generally to the field of charge air boosting of internal combustion engines and more particularly to a variable geometry turbine employing balanced variable vanes and a hydraulic servo driven by a proportional solenoid for vane closing with a reaction spring for vane opening.

Description of the Related Art

[Para 2] Variable Geometry Turbines (VGT) for turbochargers have become very popular in various automotive and truck market segments over the past 20 years. All major turbocharger manufacturers now have VGTs in production. The capability of the VGT to increase the controllability of the air flow to the engine has proven to be of great value to certain engine market segments. The VGT can be used to vary the air fuel ratio (AF), improve the transient response of the turbocharger, provide additional back pressure to the engine to drive external cooled EGR, and/or provide additional back pressure to augment decompression braking.

[Para 3] This increased capability does have certain drawbacks. For example increased turbocharger cost, decreased reliability, reduced durability, additional noise, more difficult packaging of the turbocharger in the engine compartment, and increased complexity in troubleshooting.

[Para 4] There have been attempts to mitigate some of these negative factors with a reduced cost design by using fewer and simpler parts which can be made using net-shape technologies, and integrated actuation. The integrated actuation improves the packaging by reducing the total volume, and reduces the number of failure modes related to non-integrated, bolt on actuation.

[Para 5] Electro-hydraulic actuation as used on embodiments disclosed in US Patent no. 6,269,642 issued on 08/07/2001, eliminates the need for external power to be brought to the turbocharger and instead uses the oil system supplied for lubrication and damping as a power source. An additional advantage to this is that it carries heat away rather than creating more heat as an electrical actuation system would.

[Para 6] However, a number of problems still exist with this type of design. To achieve a simple one piece net shape vane, the actuation system was incorporated in the flow passage. This increases the temperature, level of contaminants, and disrupts the aerodynamic flow, thus leading to reduced durability. The unique vane/unison ring conjugate action limited the rotation of the vanes to about 20 degrees.

[Para 7] A double acting hydraulic servo with four way spool valve and proportional solenoid has a number of issues as well. Due to the double acting nature, a spool valve has to have a number of nested annular passages which results in small flow areas. This limits the response of the system and makes it highly dependent on the oil viscosity. With cold starts in frigid climates, this system will take minutes to become functional with appropriate response times. Another issue with a double acting system is that it is more costly. Twice as many control interfaces are required and the precision required of the system is very high.

[Para 8] The vane mechanism is the heart of the variable geometry turbine, and is the most expensive and most difficult to produce cost effectively with high reliability. Initial designs in the industry had many small, very precise parts which were made from super-alloys and machined or ground to very fine tolerances. The majority of the designs used a multi-piece vane/arm/lever that had to be welded during assembly. These designs were very costly and also resulted in cantilevered loads, where the aerodynamic torque on the vane had to be resisted by a force out of the plane of the vane or the support bearings. These complex cantilevered loadings result in much higher loads, particularly at the ends of the “bearings” or supports. Since the vane mechanism has to move billions of times in its life, live in a high vibration environment, survive temperatures above 700C, in a oxidizing atmosphere, with no lubrication and no sticking or binding, additional loads trying to cock the vane in the bearing are very problematic.

[Para 9] It is therefore desirable to provide a variable geometry turbocharger which overcomes the shortcomings of the prior art to allow lower cost, higher durability and reliability, expanded range flexibility and improved response under all ambient conditions.

SUMMARY OF THE INVENTION

[Para 10] The embodiments of the present application describe a variable geometry turbine (VGT) for turbochargers incorporating a turbine nozzle having a plurality of vanes, each vane having an airfoil with an inner disc and an outer disc. A first nozzle plate incorporates pockets to receive the inner discs with the inner discs substantially flush with a nozzle surface on the first nozzle plate. A second nozzle plate has pockets to receive the outer discs with outer discs substantially flush with a second nozzle surface. An integral actuation system rotates the plurality of vanes for variation of the nozzle geometry.

[Para 11] In disclosed embodiments, the actuation system employs a single acting hydraulic servo driven by a proportional solenoid. The servo incorporates a spool positioned by a pin of the proportional solenoid. The spool has an oil supply metering edge and a drain metering edge positionable for connecting an oil source and an oil drain, respectively, with a passage. A piston is received in a chamber in a center housing with the chamber having a pressure cavity in communication with the passage. The piston translates toward a drain cavity in the chamber responsive to connection of the oil source to the passage and translates toward the pressure cavity responsive to connection of the oil drain to the passage. The piston positions a drive lever rotating the vane system in the turbine nozzle.

BRIEF DESCRIPTION OF THE DRAWINGS

[Para 12] These and other features and advantages of the present invention will be better understood by reference to the following detailed description when considered in connection with the accompanying drawings wherein:

[Para 13] FIG. 1 is a perspective view of a turbocharger in which the described embodiment may be employed;

[Para 14] FIG. 2 is a partially sectioned view of the turbocharger of FIG. 1 showing details of the embodiment;

[Para 15] FIG. 3A is a perspective rear view of a vane for use in the embodiment;

[Para 16] FIG. 3B is a perspective front view of the vane;

[Para 17] FIG. 4A is a perspective front view of a first nozzle plate for use in the embodiment;

[Para 18] FIG. 4B is a perspective rear view of the first nozzle plate;

[Para 19] FIG. 5 is a perspective front view of the first nozzle plate with vanes installed;

[Para 20] FIG. 6 is a perspective rear view of a second nozzle plate;

[Para 21] FIG. 7 is a sectioned perspective rear view of the first nozzle plate, vane and unison ring;

[Para 22] FIG. 8 is a perspective front view showing the assembled first nozzle plate, vanes and second nozzle plate;

[Para 23] FIG. 9 is a perspective front view of the vanes, unison ring and drive elements;

[Para 24] FIG. 10 is a partially sectioned view rear view of the turbocharger of FIG. 1;

[Para 25] FIG. 11 is a side sectioned view of the turbocharger showing an integrated hydraulic servo actuator with solenoid in the vanes open position;

[Para 26] FIGs. 12A and 12B are a detailed section views of the integrated actuator showing flow paths for pressurizing and draining oil respectively;

[Para 27] FIG. 13 is a perspective view of a hydraulic drive piston for the integrated actuator;

[Para 28] FIG. 14A is a side view of a drive cam for vane operation;

[Para 29] FIG. 14B is a perspective view of the drive cam;

[Para 30] FIG. 15 is a partial side sectioned view of the integrated hydraulic servo actuator in the vanes closed position; and,

[Para 31] FIG. 16 is a partial side sectioned view of the integrated hydraulic servo actuator in the vanes open position.

DETAILED DESCRIPTION OF THE INVENTION

[Para 32] The embodiments described herein provide a VGT nozzle for a turbocharger that employs simplified vanes having pressure balancing discs each with a pin extending through the vane between a first nozzle plate and a second nozzle plate for rotational support and spacing of the nozzle plates. Aan integrated solenoid driven hydraulic servo actuator is employed for rotation of the vanes. The actuator incorporates a spring balanced hydraulic piston internally engaging a cam for rotation of a vane drive axle to a master vane with a unison ring for mutual rotation of the remaining vanes.

[Para 33] Referring to the drawings, FIG. 1 show a turbocharger 10 having a turbine assembly 12, a center housing 14 and a compressor assembly 16. The turbine assembly 12 is contained within a turbine housing 18 and the compressor assembly is contained within a compressor housing 20 which are attached to the center housing 14. FIG. 2 shows the turbocharger 10 with the turbine housing and compressor housing removed for clarity and partially sectioned to display the internal components of the turbocharger. A turbine back plate 22 and compressor back plate 24 extend from the center housing 14 for attachment of the turbine housing 18 and compressor housing 20, respectively. A turbine wheel 26 is connected to a compressor wheel 28 with a shaft 30 supported by bearing assemblies in the center housing 14.

[Para 34] The VGT nozzle is provided by multiple vanes 32, which will be described in greater detail subsequently, circularly arrayed around the turbine wheel 26 between a first nozzle plate 34 and a second nozzle plate 36. While shown in the drawings for the exemplary embodiment as evenly spaced, the vanes can be equally spaced or, in alternative embodiments, unequally spaced to lower the tonal acoustic emissions. Rotational positioning of the vanes 32 is accomplished with an integral actuator 38 contained within the center housing 14. A solenoid 40, hydraulic metering spool 42 and piston 44, the operation of which will be described in detail subsequently, position a lever 46 which drives an actuating shaft 48 to rotate a master vane 32a.

[Para 35] The vane mechanism is the heart of the VGT, and in the prior art has been the most expensive and most difficult to produce cost effectively with high reliability. Prior art designs in the industry had many small, very precise parts which were made from super-alloys and machined or ground to very fine tolerances. The majority of the designs used a multi-piece vane/arm/lever that had to be welded during assembly. These designs were very costly and also resulted in cantilevered loads, where the aerodynamic torque on the vane had to be resisted by a force out of the plane of the vane or the support bearings. These complex cantilevered loadings result in much higher loads, particularly at the ends of the “bearings” or supports. Since the vane mechanism has to move billions of times in its life, in a high vibration environment with temperatures above 700C and an oxidizing atmosphere, with no lubrication and no sticking or binding, additional loads trying to cock the vane in the bearing are very problematic.

[Para 36] The vanes 32 in the present embodiment provide a novel geometry that allows the vanes and their actuating interface to be made as one piece using net shape technology and are manufacturable without expensive machining. The actuation force reacting the vane aerodynamic force is moved to be within the span of the support bearing(s) and flow disruption is minimized. The temperature at the wearing interface is reduced by moving the actuating interface mechanism out of the gas flow passage. Friction and wear is avoided by balancing the axial thrust force on the one piece vane to a net zero. The vanes 32 have at least 35 degrees of rotation to achieve a high turn-down ratio in the turbine nozzle.

[Para 37] As shown in FIGs. 3A and 3B, each vane 32 includes an airfoil 50 and a drive peg 52 having a key 54 to be received in the unison ring as will be described in greater detail subsequently. An inner disc 56 extends from the interface of the airfoil and drive stub to provide smooth airflow in the nozzle by providing an interface between the vane 32 and first nozzle plate 34. As seen in FIGs. 4A and 4B the first nozzle plate 34 has pockets 58 that receive the drive pegs 52 and are sized to closely receive the discs 56 flush with a first nozzle surface 59 as best seen in FIG. 5. The pockets 58 are relieved through an inner circumference wall 60 providing windows 62 through which the keys 54 extend. Bores 64 extend through the pocket bottoms to receive a bearing pin, as will be described in greater detail subsequently. To accommodate the master vane 32a, a master pocket 58a incorporates an enlarged bore 66 to receive the actuating shaft 48 as described with respect to FIG. 2.

[Para 38] For pressure balancing the vanes 32 an outer disc 68 is provided extending from the airfoil opposite the disc 56. The outer discs 68 for the vanes 38 are received in outer pockets 70 in the second nozzle plate 36 to be flush with a second nozzle surface 71 as seen in FIG. 6. To assure equal pressure distribution on the discs 56 and outer discs 68 a pressure equalization bore 72 extends through the discs and airfoil 50. The inner and outer discs employ reliefs or pockets 73a, 73b to provide a plenum volume to dampen pulses, to equalize the pressures and to make certain that the pressure equalization bore 72 is not obstructed. In addition, leakage around the vanes is undesirable and the reliefs 73a, 73b may be employed to prevent the plenum volume from connecting upstream of the vane and downstream of the vane.

[Para 39] Since the pockets 58 in the first nozzle plate are quite large, using this as a bearing surface for the vane to rotate on is not feasible due to the high friction created by the large moment arm. To provide a low friction bearing surface, as shown in FIG. 7, a pin 74 is pressed into a bore 76 in the first nozzle plate 34 and is closely received in an aperture 77 in the vane drive peg 52. The vane drive peg 52 rotates on the diametric surface of the pin 74.

[Para 40] Customarily, the second nozzle plate is positioned by attaching to the turbine housing or by stand-off pins protruding from the face of the first nozzle plate. A turbine housing is not axisymmetric so the thermal transients that the turbine housing is subjected to results in nozzle mounting faces that are not necessarily parallel or concentric. Variable geometry turbines in the prior art which rely on this type of mounting have to have additional vane side clearance to assure that the vanes do not bind during transient operation of the engine. This reduces the efficiency, especially when the vanes are exerting a significant turning or acceleration of the flow. Stand-off pins are also aerodynamic impediments to the flow and cause a flow disruption and flow distortion. They also induce an additional harmonic element to the flow field which can increase the noise of the turbocharger.

[Para 41] The present embodiment avoids these problems. The bearing pin 74 for each of the vanes has post 78 that extends through a central hole 80 concentric with aperture 77 in the vane 32 including the outer disc 68 (also seen in FIGs. 3A and 3B) and protrudes beyond the outer disc 68 of the vane as shown in FIG. 5. This provides an exact spacer for the second nozzle plate 34 to rest upon. The post 78 has a clearance with the hole through the vane so it does not create additional friction. Each vane (except the drive vane 32a, which drives the unison ring) has this spacer feature and the second nozzle plate 36 axial position is well controlled.

[Para 42] The second nozzle plate does not need to be firmly attached to the posts 78 and the posts are received in reliefs 82 in the pockets 70 as seen in FIG. 6. Upstream pressure from the turbine is fed to the backside of the nozzle plate causing a net thrust on the plate to move to be in contact with the spacer posts. The face of the second nozzle plate in contact with the hot gas flow will be at a lower pressure as the flow is accelerating through the nozzle and pressure is converted to velocity. Thus there is always a net pressure on the second nozzle plate pushing it into contact with the spacer posts 78. The assembled first and second nozzle plates 34, 36 with the vanes 32 are shown in FIG. 8.

[Para 43] Returning to FIG. 7, a unison ring 84 is concentrically received by the inner circumference wall 60. As previously described with respect to FIGs. 4A and 4B, the pockets 58 are relieved through the inner circumference wall 60 providing windows 62 through which the keys 54 extend. The unison ring 84 has circumferentially spaced notches 86 to receive the keys 54. Rotation of the master vane 32a by the actuating shaft 48 causes rotation of the unison ring with commensurate rotation of all of the vanes 32 through keys 54.

[Para 44] Control of the variable geometry system in prior art VGT turbochargers has typically required a separate actuation system designed to bolt to the turbocharger through a bracket and then connected by some means to a shaft extending through the turbine housing which facilitates the variable geometry movement. This is not ideal from many viewpoints: packaging, magnification of engine vibration, protection from heat of the turbine, calibration, leakage of hot gas through the actuating shaft, and tamper-proofing. Tamper-proofing is a challenge due to On-Board Diagnostic regulations as well as the cottage industry of the engine performance aftermarket, which sells kits to modify engine controls to circumvent emissions controls and extract more power from the engine.

[Para 45] The present embodiment incorporates the variable geometry mechanism as an integral system, incorporating the actuation components into the existing center housing. This reduces the vibration magnification caused by cantilevering an actuation system which has been bolted to the turbocharger through a bracket system. Moving the actuation to become an integrated part of the center housing reduces the vibration and also allows the actuation components to be cooled by the bearing lubricating oil. The actuating power required of the actuation system can then be provided by the existing oil system. This also aids tamper-proofing by eliminating connection adjustments as well as eliminating the ability to unbolt one actuation system and replace it with another design. The previously identified prior art design disclosed in US patent no. 6,269,642 employed a double-acting hydraulic servo which resulted in certain shortcomings which are overcome by the embodiment disclosed herein.

[Para 46] The present embodiment uses a single acting hydraulic servo driven by a proportional solenoid. The components of the overall actuation system are shown in FIGs. 9 and 10. The hydraulic servo employs the spool 42 driven by solenoid 40 through pin 104 to move the piston 44 and therefore the vanes 32 in one direction and an return spring102 works to move the vanes in the opposite direction. The piston 44 carried with the center housing 14, as previously described with respect to FIG. 2, provides actuation of the vanes 32 through the actuating shaft 48 and lever 46 as will be described subsequently The embodiment as a single acting system employs the return spring102 to open the vanes and the hydraulic servo to close the vanes. The maximum force required by the system is to open the vanes when they are fully closed. If the spring is used to open the vanes, then it is at maximum compression at the point where the vanes are closed (and thus maximum spring force). The maximum vane force which occurs with the vanes in the closed position is offset by the spring force, and the hydraulic servo only has to have enough force to overcome the maximum spring force, but assisted by the vane force.

[Para 47] In the system provided in the embodiment, there is no surrogate signal (i.e. cam) for determining position of the piston 44; instead, the position of the piston is measured directly by a feedback spring 106 creating a force proportional to position of the piston. In addition, the force of the feedback spring 106 augments the force of the return spring102.

[Para 48] The piston 44 can be round as a traditional piston or oblong, as shown, when space requirements dictate that a round piston of sufficient area is too large to fit within the dimensions of the center housing. The piston is carried in a chamber108 in the center housing 14 as shown in FIG. 11. Lever 46 is carried within the piston 44. The piston 44 moves the vanes by pushing on lever 46 connected to the actuating shaft 48 controlling the vanes.

[Para 49] The piston 44 has hydraulic pressure to push it one direction and the return spring102 and feedback spring 106 to push the opposite direction. Since the piston 44 must push the drive lever 46 both directions the drive lever is nested inside the piston and has a plane of symmetry through an axis of rotation of the principal vane 32a in the vane system.

The piston 44 provides a cavity 100 with a cutout 105 for the drive lever 46 and actuating shaft 48 as shown in detail in FIG. 13. Since the actuating shaft 48 has a center that is at a fixed position, and the drive lever 46 rotates around that, but the piston 44 must traverse laterally, cavity 100 must be present in the piston to allow that traverse to happen without interference. The inside of the piston must remain at drain cavity pressure and not retain oil. The main drain of the actuating system is through an aperture 109 in the bottom of the piston wall. Additionally there is a relief 110 in the piston that connects the drain cavity 111 of chamber 108 between the piston and the spool (that includes both springs), to the main drain 115. The piston 44 incorporates spring locators for the return spring102 and feedback spring 106. For the embodiment shown, an internal locator 107a is employed for the feedback spring and an external locator 107b is employed for the actuation spring.

[Para 50] The drive lever 46 shown in detail in FIGs. 14A and 14B has flat planes 112,114 (best seen in FIG. 14A) that mate with the walls 116, 118 of the piston cavity 100 when the vane system is at either end of its travel and becomes the system end stops. This gives a large surface area to resist wear when the vanes are at either of the two extremes of movement. When the piston is in an intermediate position, the two contact points in curved surfaces 120, 122 of the drive lever necessarily must have a width substanitally equal to the inside width of the piston cavity (minus a small clearance which prevents binding). The drive lever contact points must form a locus of constant width, which is achieved by creating the dual curves surfaces 120, 122 which have a common center point and form a circular radius. The diameter of this circle is the desired inside width of the cavity 100 in piston 44 minus the operating clearance.

[Para 51] FIGs. 12A and 12B show the hydraulic elements and flow paths to supply oil to the piston pressure cavity 113 (FIG. 12A flow path designated by arrow 200a to move the vanes closed) and to drain oil from the piston drain cavity 111 (FIG. 12B flow pateh designated by arrow 200b to move the vanes open).

[Para 52] The spool 42 has a supply metering edge 124 and a drain metering edge 126 that mate with an annular cut 128 in the center housing. Oil is connected to an annular recess 130 of the spool on the solenoid side of the spool by a port 131 that connects to the main oil gallery of the turbocharger. The annular recess 130 acts as the oil supply for the hydraulic servo. When the force of feedback spring 106 is less than the force provided by the proportional solenoid 40 through pin 104, the spool 42 translates toward the spring and opens the supply metering edge 124. Oil then flows through the passage 130 to the hydraulic side of the piston in pressure cavity 113 as shown by arrow 200a, and the piston 44 translates toward the drain cavity side of chamber 108 rotating drive lever 46 counterclockwise causing the vanes to be more closed. As the piston moves toward the drain cavity side of chamber 108, the feedback spring compression increases and will at some point match the solenoid force. When the two forces are in balance, then the spool becomes in balance and the metering edges are closed, oil stops flowing and actuation system has achieved the position commanded by current supplied to the proportional solenoid. Control of the VGT may then be accomplished through simple current control to the proportional solenoid by a master engine computer.

[Para 53] When the opposite situation happens, the feedback spring force is greater than the solenoid force, the spool moves toward the solenoid, opening up the drain metering edge 126. The oil passage to the hydraulic side of the piston in pressure cavity 113 is connected through the passage 130 to the drain cavity 111 between the spool and the piston as shown by arrow 200b and return spring 102 urges the piston 44 to translate toward the pressure cavity side of chamber 108. The piston and lever position for a fully open position of the vanes is shown in FIG. 16 while a fully closed position is shown in FIG. 15. The drain cavity 111 is also connected through passage 132 to the end of the spool 42 where it mates to the solenoid pin 104. This assures that there is no buildup of pressure on the end of the spool which could unbalance the system.

[Para 54] The actuating system employs master vane 32a which does not ride on a pin as do the other vanes. Instead, a hex shape hole in the peg of the vane is employed which allows the actuator shaft 48 (with a hex shaped section) to be pressed into the vane. Thus, the shaft drives the vane through the hex press fit, which then rotates the unison ring, which in turn drives the other vanes.

[Para 55] For assembly of the embodiment shown in the drawings, the drive lever 46 is first assembled loosely into the cavity 100 in the piston 44; then the piston inserted into the center housing chamber 108 wherein a boss 134 on the drive lever 46 (seen in FIG. 10) drops into a circular recess in the center housing. The piston is located laterally by a fixture, which orients the drive lever correctly. The actuating shaft 48 (with the primary vane 32a pressed on) can then be inserted through a bushing in the center housing and a hex shaped end can be pressed into a receiving hex shaped hole 136 (seen in FIGs. 14A and 14B) in the drive lever. The hex end of the shaft can be relieved with a slot(s) to reduce the stiffness of the hex and thus the press force requirements into the drive lever. The boss 134 on the drive lever positions the drive lever 46 in the precise position within the width of the chamber 108 in the piston 44. Chamber 108 is then closed with an end cap 138 seen in FIG. 11.

[Para 56] Having now described the invention in detail as required by the patent statutes, those skilled in the art will recognize modifications and substitutions to the specific embodiments disclosed herein. Such modifications are within the scope and intent of the present invention as defined in the following claims.

WHAT IS CLAIMED IS:

1. A variable geometry turbine (VGT) for turbochargers comprising:

a turbine nozzle having a plurality of vanes each vane having an airfoil with an inner disc and an outer disc;

a first nozzle plate having pockets to receive the inner discs, said inner discs substantially flush with a nozzle surface on the first nozzle plate;

a second nozzle plate having pockets to receive the outer discs, said outer discs substantially flush with a second nozzle surface; and,

an actuation system rotating the plurality of vanes.

2. The VGT as defined in claim 1 wherein each vane further comprises a drive peg extending from the inner disc, said drive peg having a key to engage a unison ring.

3. The VGT as defined in claim 1 further comprising:

a bearing pin for each of the plurality of vanes, said bearing pin extending from the first nozzle plate and received in a aperture in the drive peg.

4. The VGT as defined in claim 3 wherein each of the plurality of vanes has a hole extending through the vane concentric with the aperture in the drive peg and said bearing pin includes a post extending through the hole to protrude beyond the outer disc, said pockets in the second nozzle plate having reliefs to receive the posts whereby said second nozzle plate supported in the posts.

5. The VGT as defined in claim 2 wherein the first nozzle plate has an inner circumference wall and the keys extend through windows relieved through the wall in the pockets in the first nozzle plate.

6. The VGT as defined in claim 2 further comprising:

an actuating shaft;

a master vane receiving the actuating shaft in a hole in a drive peg; and,

a master pocket in the first nozzle plate incorporates an enlarged bore receiving the actuating shaft, a key extending from the drive peg of the master shaft engaging the unison ring.

7. The VGT as defined in claim 1 wherein each of the plurality of vanes further incorporates a pressure equalization bore extending through the inner and outer discs and airfoil.

8. The VGT as defined in claim 1 wherein the actuation system is integrally contained within a center housing.

9. The VGT as defined in claim 6 wherein the actuation system comprises:

a single acting hydraulic servo driven by a proportional solenoid.

10. The VGT as defined in claim 9 wherein the single acting hydraulic servo comprises:

a spool positioned by a pin of the proportional solenoid, said spool having an oil supply metering edge and a drain metering edge positionable for connecting an oil source and an oil drain, respectively, with a passage; and,

a piston received in a chamber in a center housing, said chamber having a pressure cavity in communication with the passage, said piston translating toward a drain cavity in the chamber responsive to connection of the oil source to the passage and translating toward the pressure cavity responsive to connection of the oil drain to the passage.

11. The VGT as defined in claim 10 further comprising a return spring urging the piston toward the pressure cavity.

12. The VGT as defined in claim 10 further comprising a feedback spring between the spool and the pin of the proportional solenoid.

13. The VGT as defined in claim 11 wherein the piston incorporates a cavity and the actuation system further comprises a drive lever carried within the cavity.

14. The VGT as defined in claim 13 wherein the actuating shaft is connected to the drive lever.

15. The VGT as defined in claim 14 wherein the drive lever incorporates flat planes that mate with walls in the cavity of the piston when the piston is at either end of its travel.

16. The VGT as defined in claim 15 wherein the drive lever incorporates contact points forming a locus of constant width, achieved by dual curved surfaces which have a common center point and form a circular radius.

17. An integral actuation system for a variable geometry turbine (VGT) comprising:

a single acting hydraulic servo driven by a proportional solenoid, said servo having:

a spool positioned by a pin of the proportional solenoid, said spool having an oil supply metering edge and a drain metering edge positionable for connecting an oil source and an oil drain, respectively, with a passage; and,

a piston received in a chamber in a center housing, said chamber having a pressure cavity in communication with the passage, said piston translating toward a drain cavity in the chamber responsive to connection of the oil source to the passage and translating toward the pressure cavity responsive to connection of the oil drain to the passage, said piston positioning a drive lever rotating a vane system in a turbine nozzle.

18. The integral actuation system as defined in claim 17 further comprising:

a return spring urging the piston toward the pressure cavity.

19. The integral acutation system as defined in claim 18 further comprising:

a feedback spring between the spool and the pin of the proportional solenoid.

20. A variable geometry turbine turbocharger comprising:

a turbine nozzle having a plurality of vanes each vane having an airfoil with an inner disc and an outer disc;

a first nozzle plate having pockets to receive the inner discs, said inner discs substantially flush with a nozzle surface on the first nozzle plate;

a second nozzle plate having pockets to receive the outer discs, said outer discs substantially flush with a second nozzle surface;

an actuating shaft;

a master vane receiving the actuating shaft in a hole in a drive peg; and,

a master pocket in the first nozzle plate incorporating an enlarged bore receiving the actuating shaft, a key extending from the drive peg of the master shaft engaging the unison ring;

a single acting hydraulic servo driven by a proportional solenoid, said servo having:

a spool positioned by a pin of the proportional solenoid, said spool having an oil supply metering edge and a drain metering edge positionable for connecting an oil source and an oil drain, respectively, with a passage; and,

a piston received in a chamber in a center housing, said chamber having a pressure cavity in communication with the passage, said piston translating toward a drain cavity in the chamber responsive to connection of the oil source to the passage and translating toward the pressure cavity responsive to connection of the oil drain to the passage, said piston positioning a drive lever rotating the actuating shaft;

a return spring urging the piston toward the pressure cavity; and,

a feedback spring between the spool and the pin of the proportional solenoid.

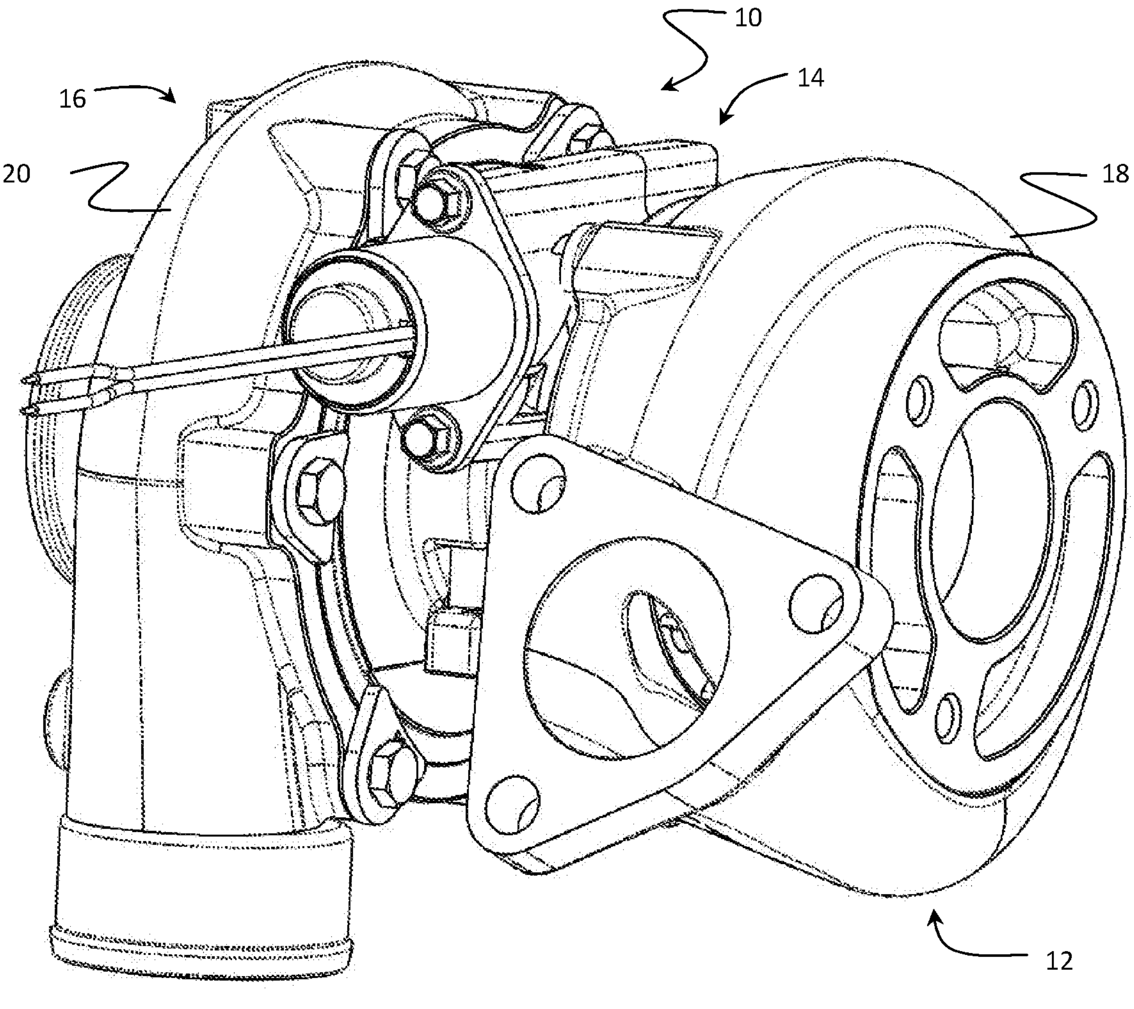

FIG. 1

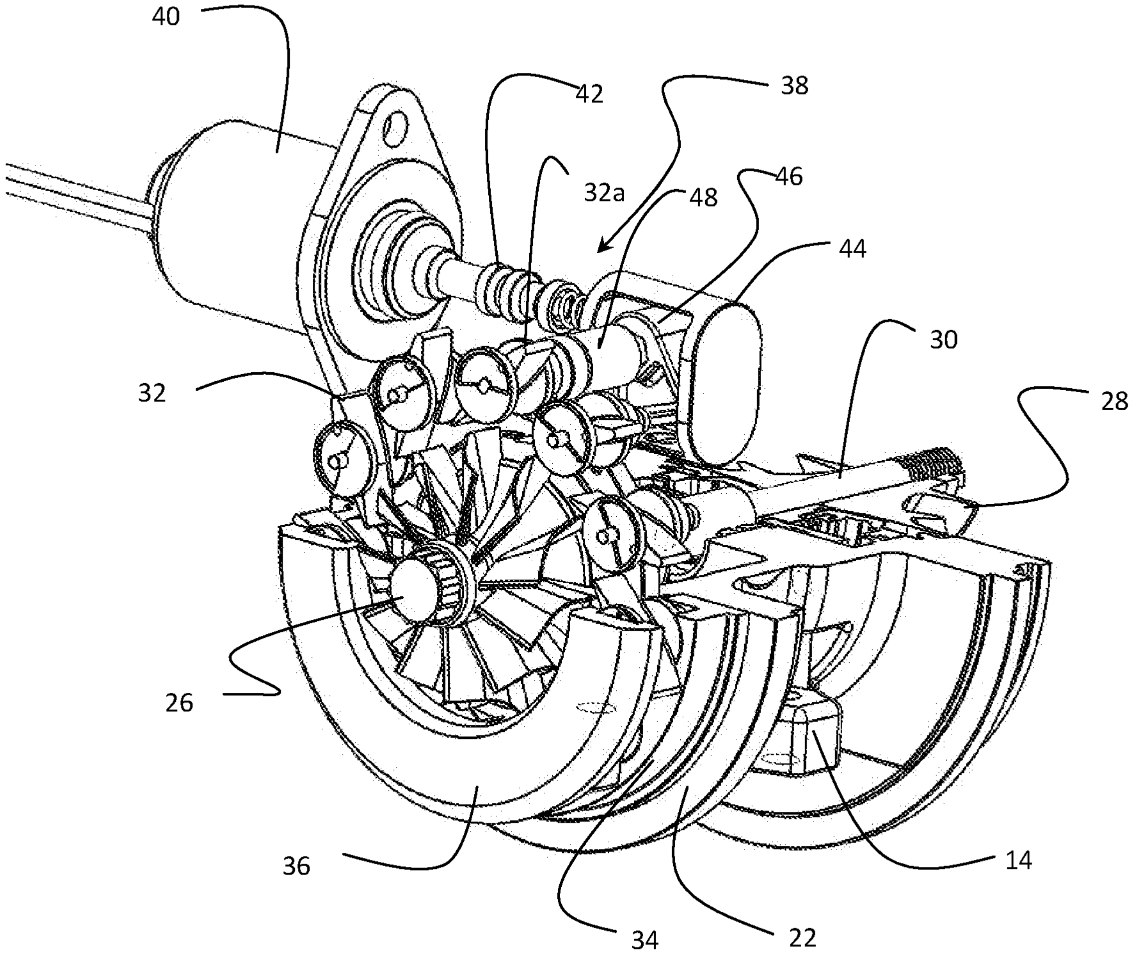

FIG. 2

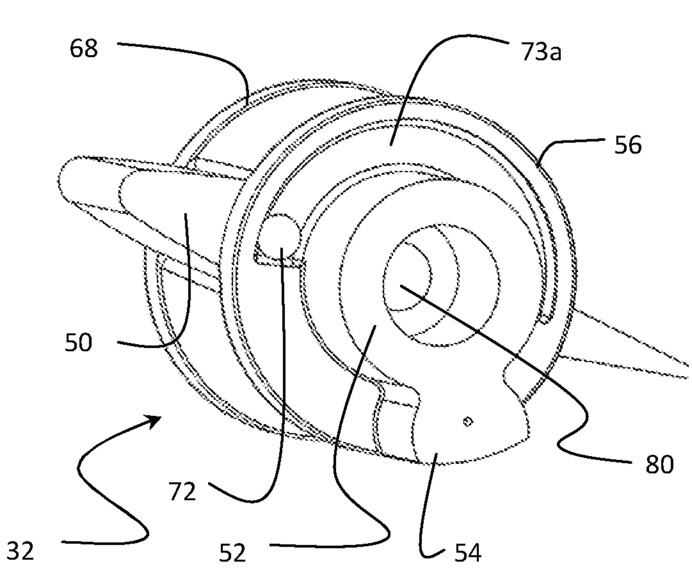

FIG. 3A

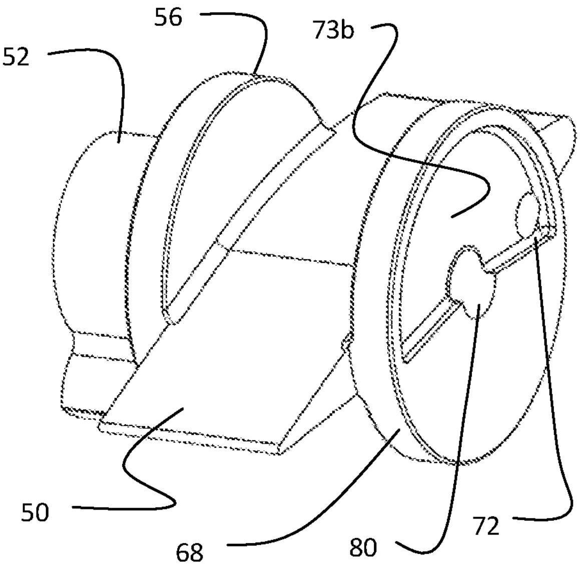

FIG. 3B

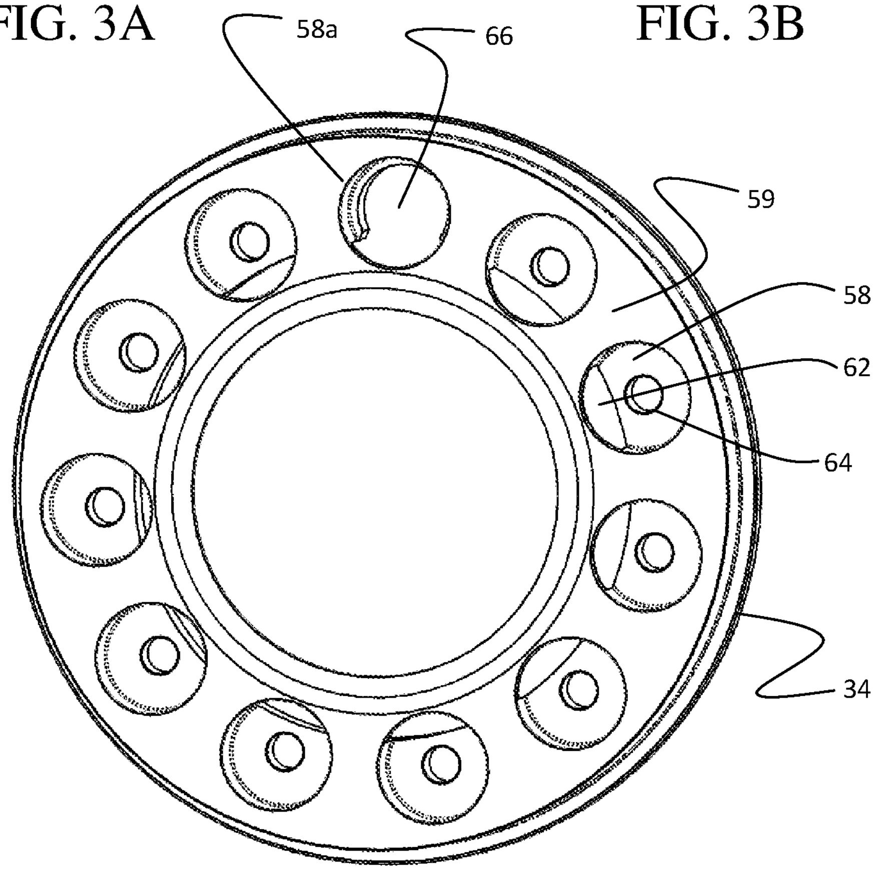

FIG. 4A

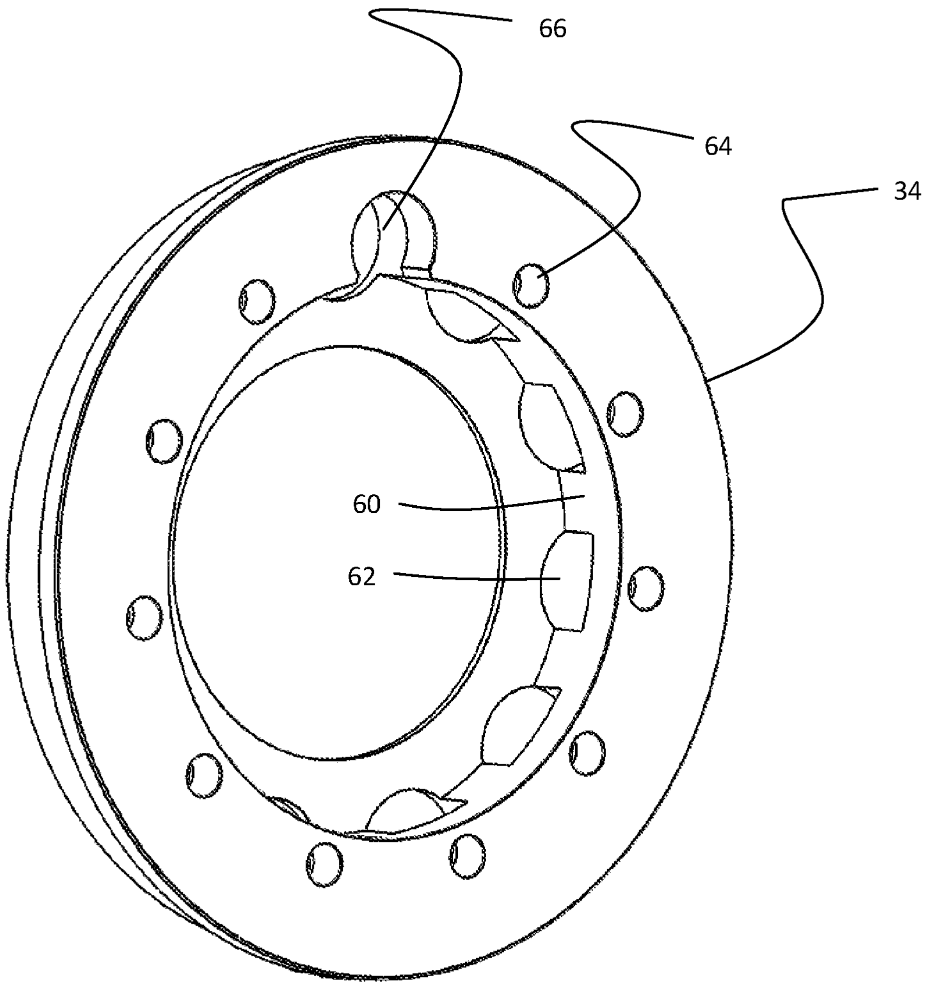

FIG. 4B

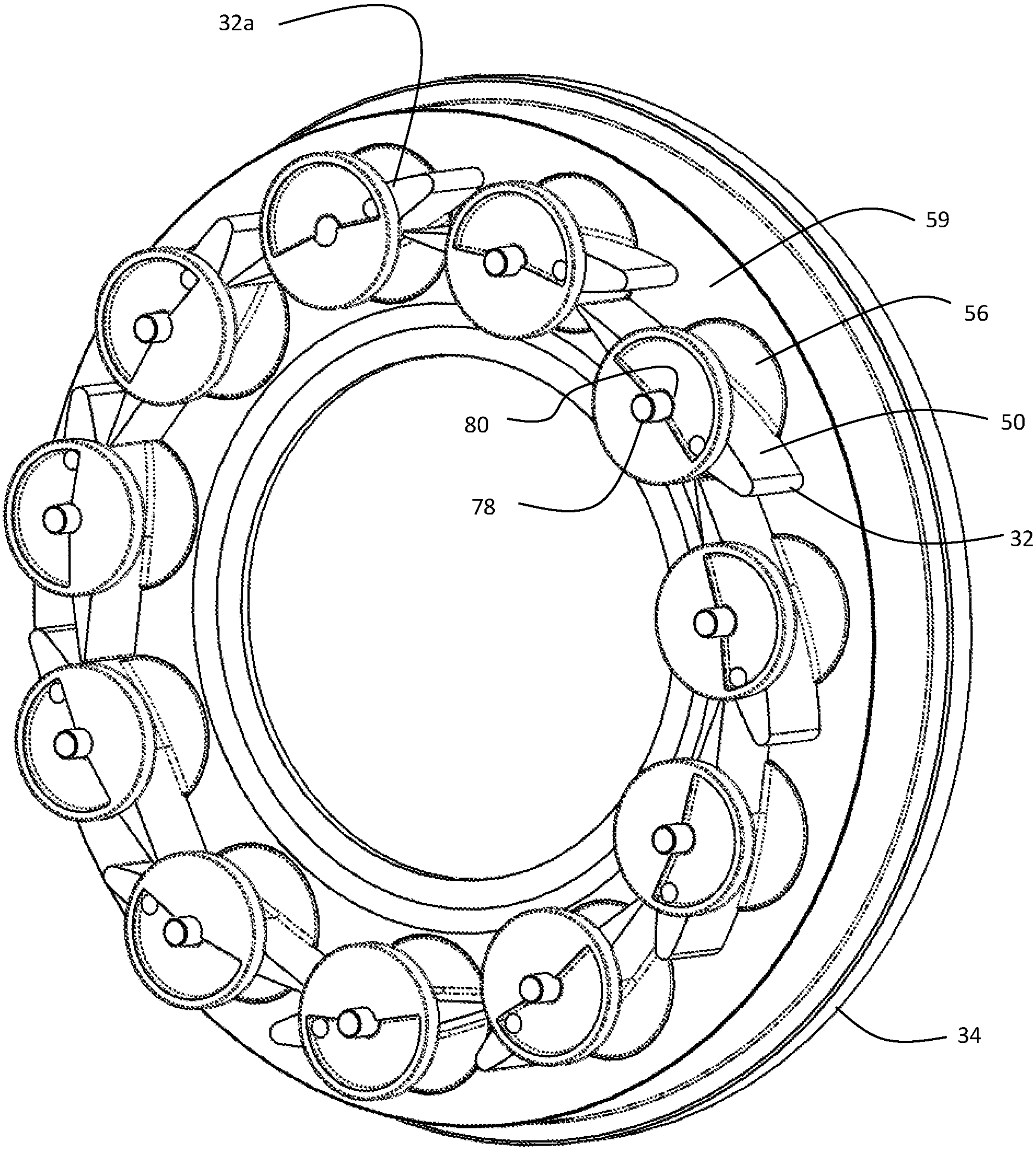

FIG. 5

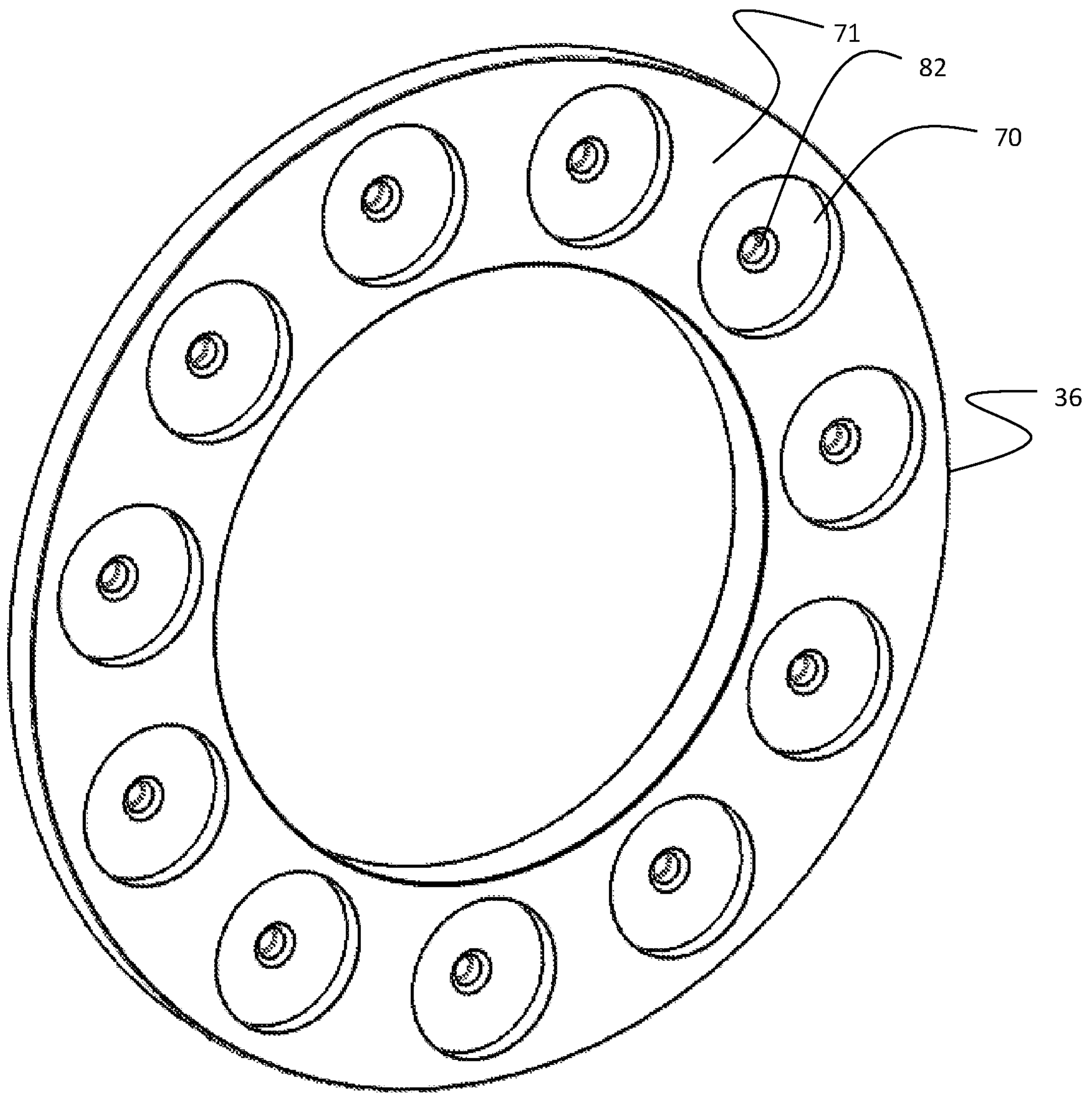

FIG. 6

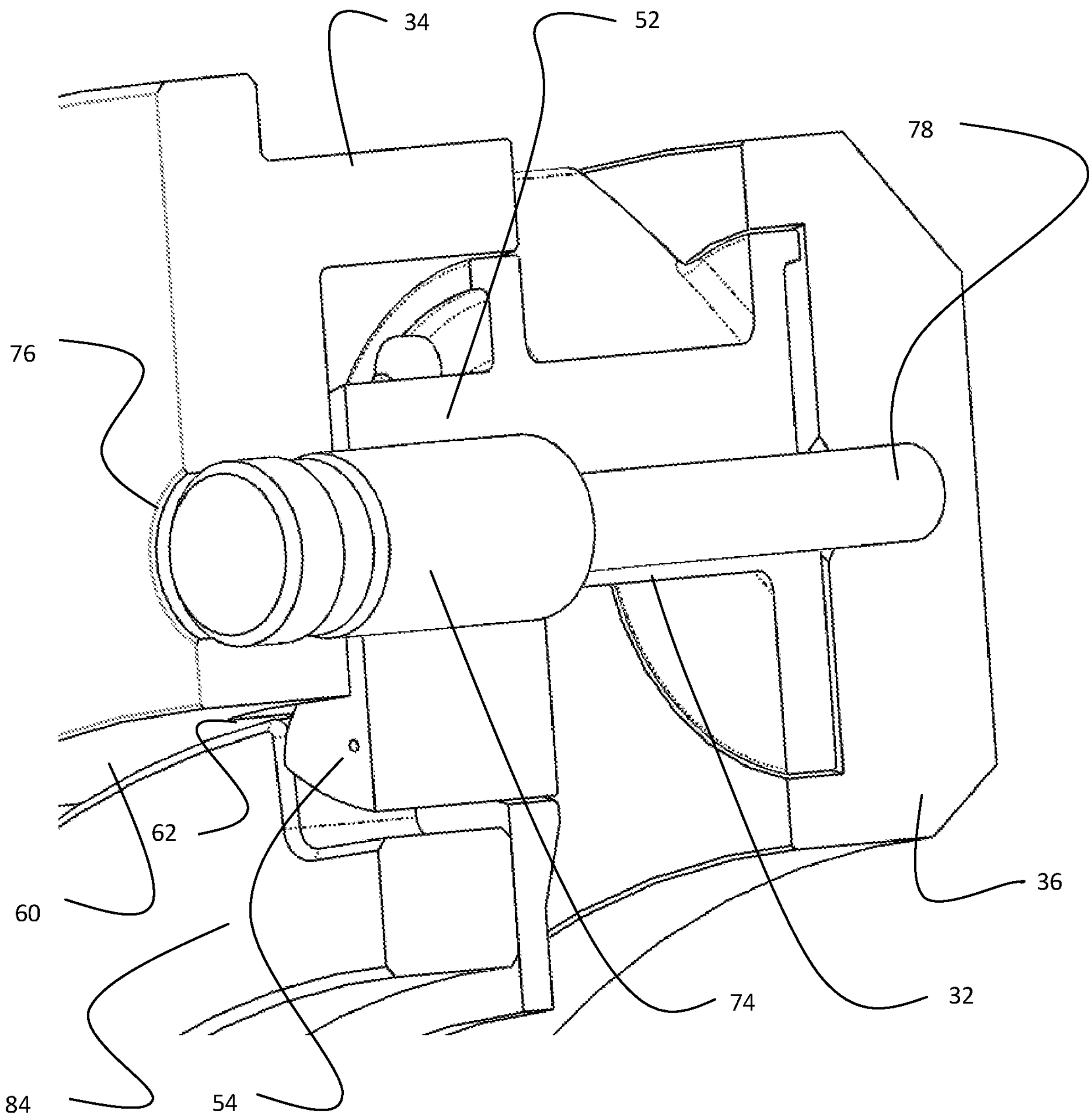

FIG. 7

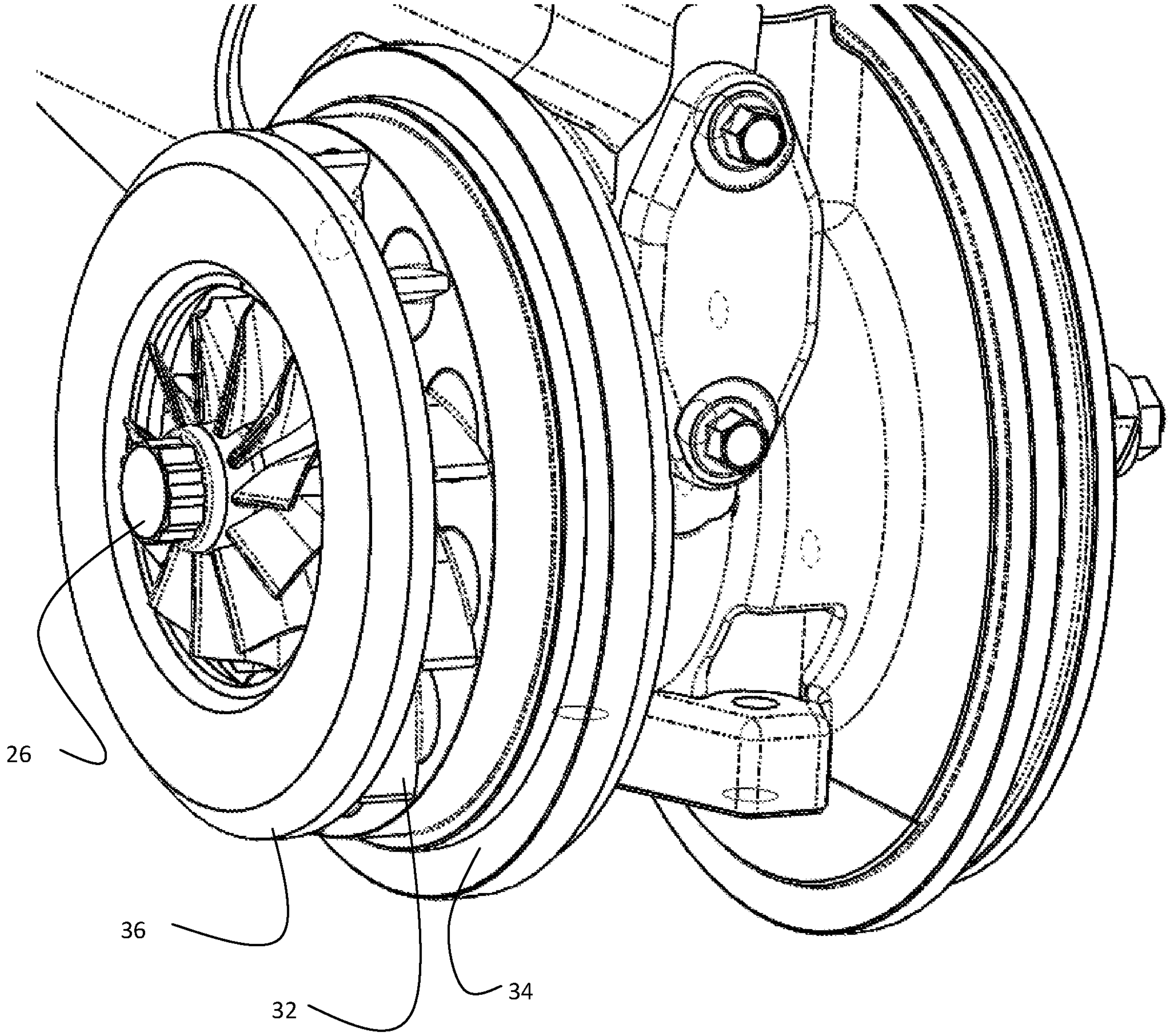

FIG. 8

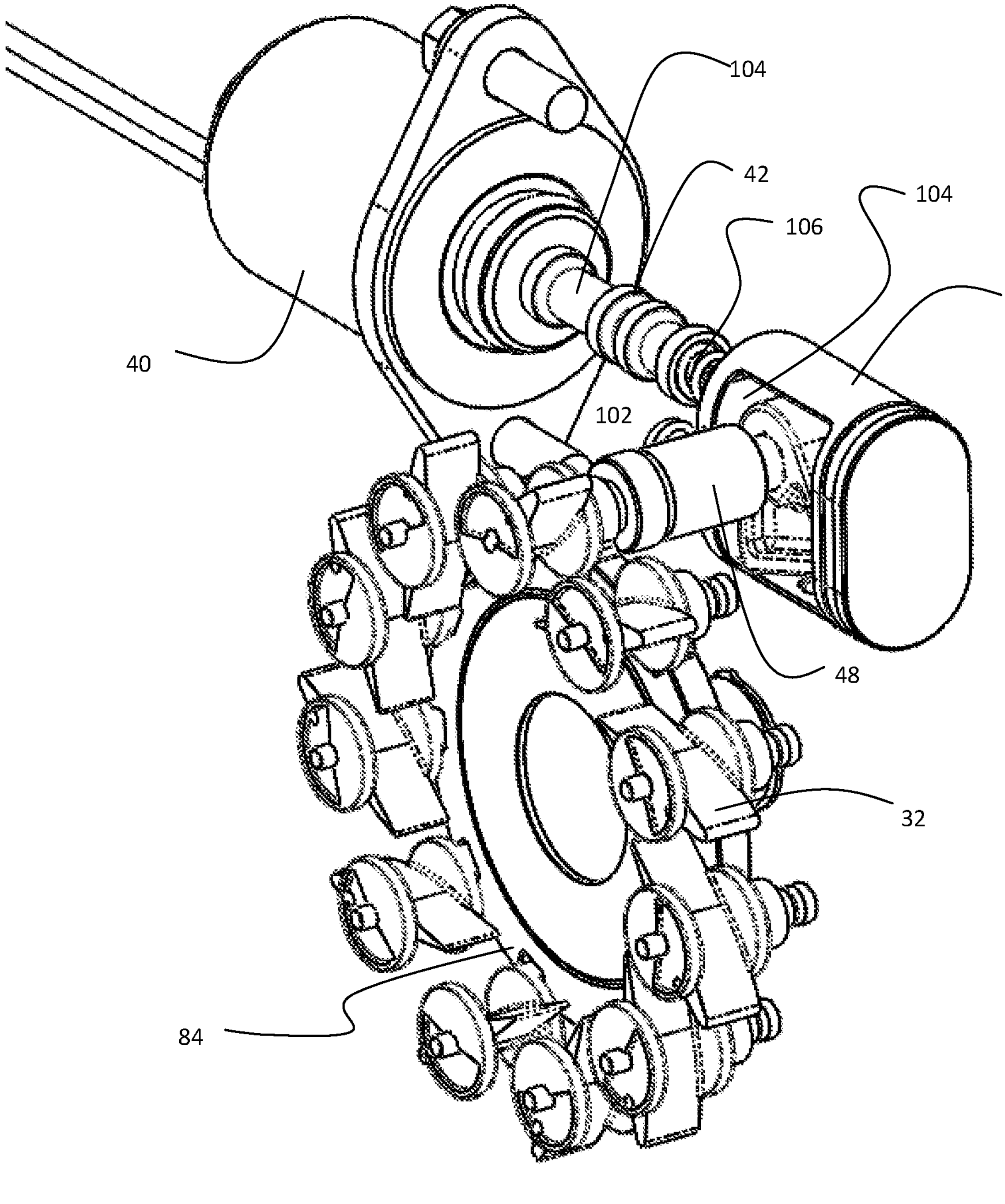

FIG. 9

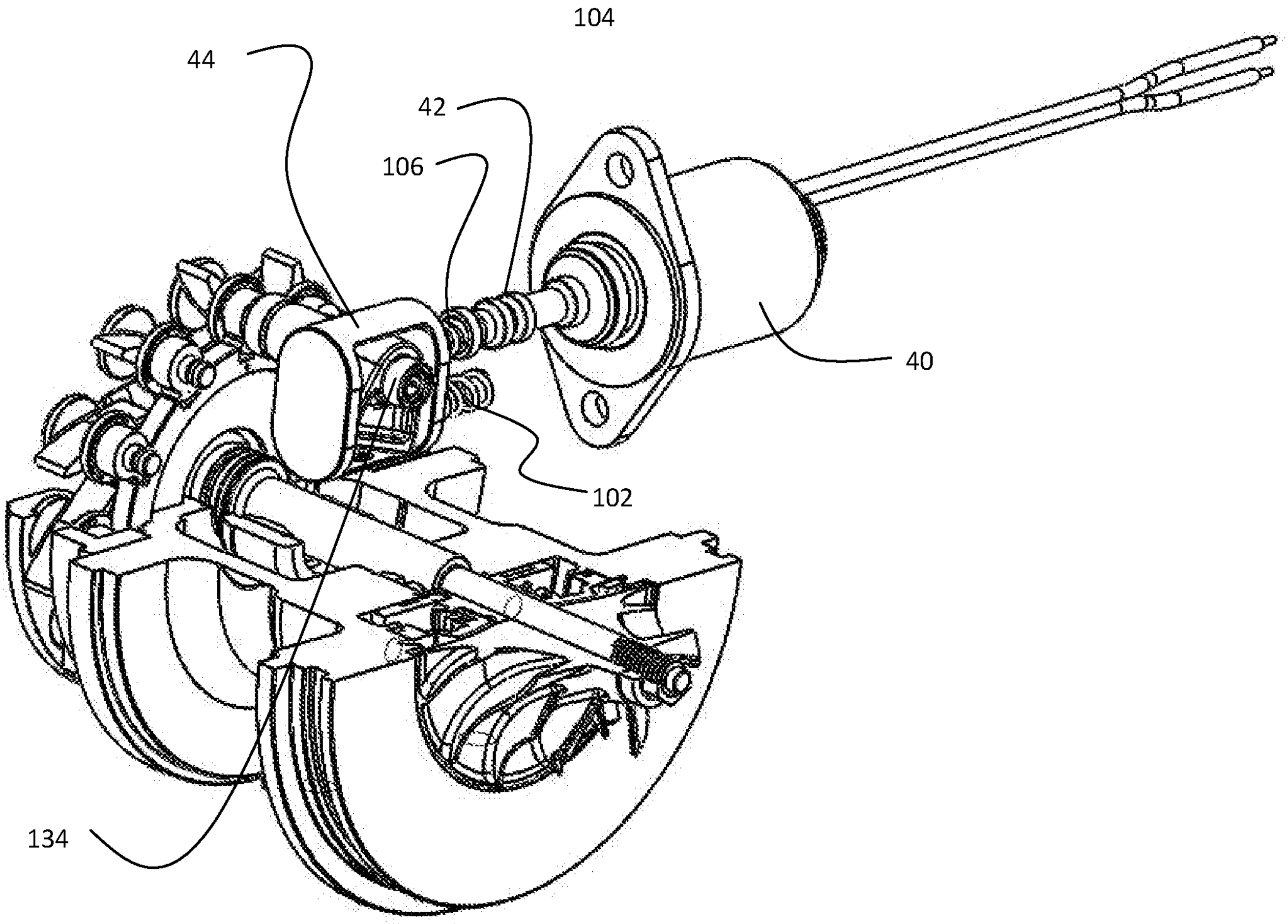

FIG. 10

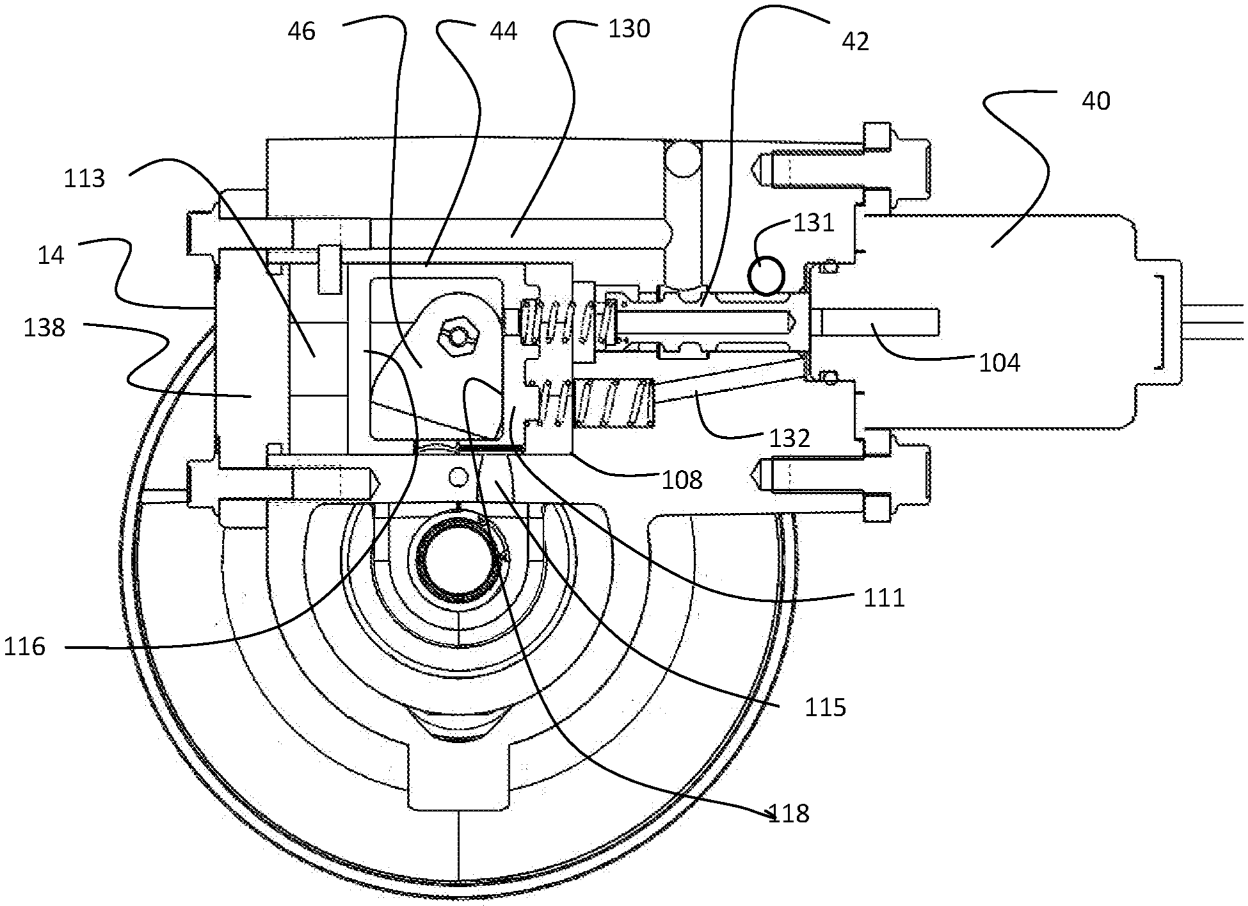

FIG. 11

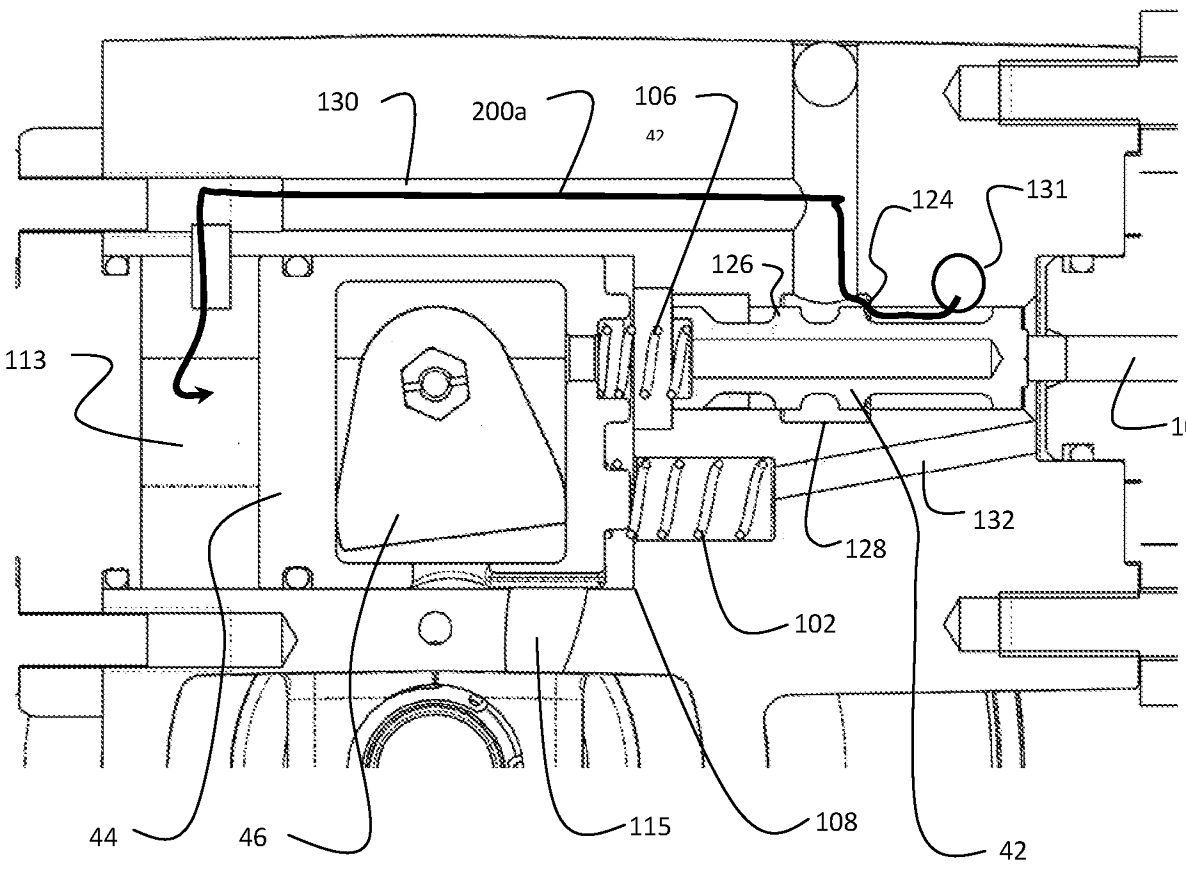

FIG. 12A

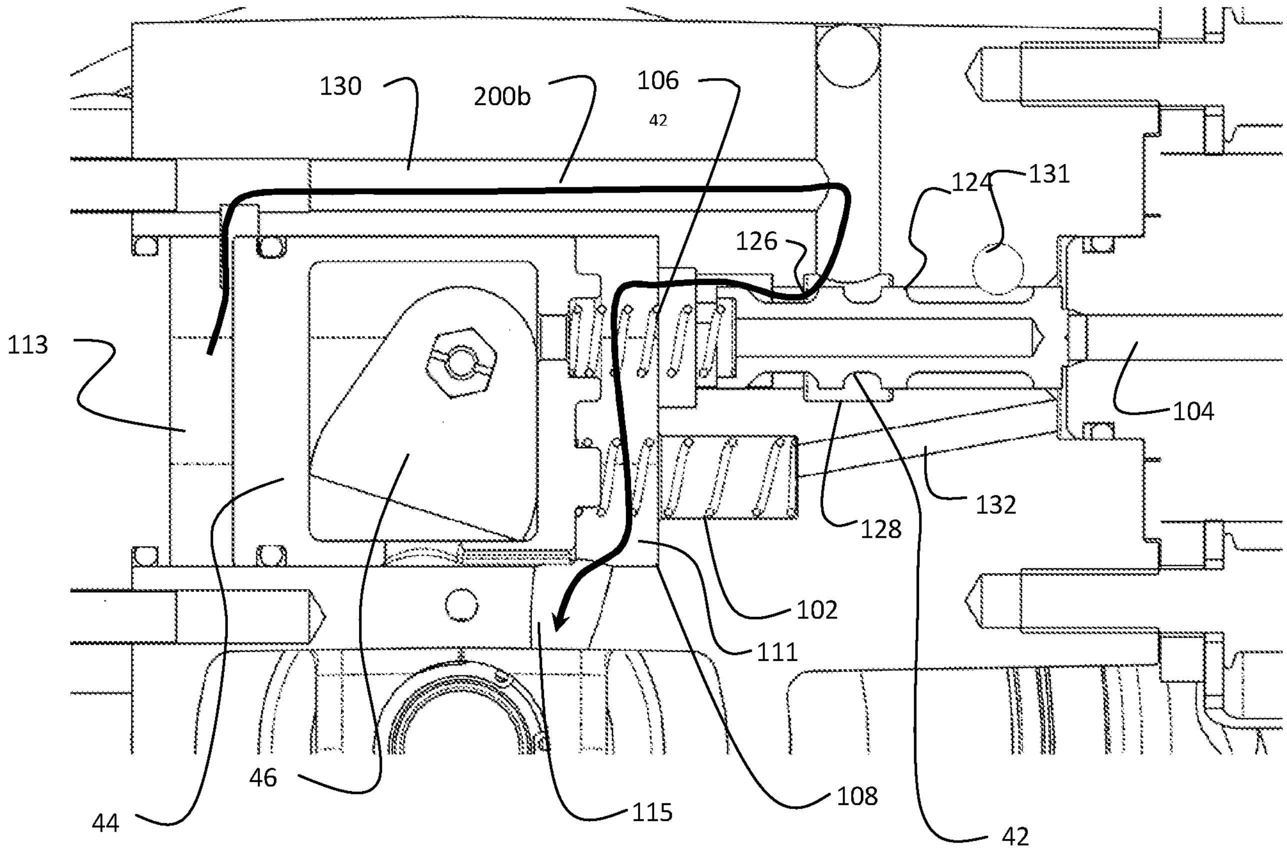
130
200b
106
42
124
131
126
113
104
132
128
102
111
108
44
46
115
42

FIG. 12B

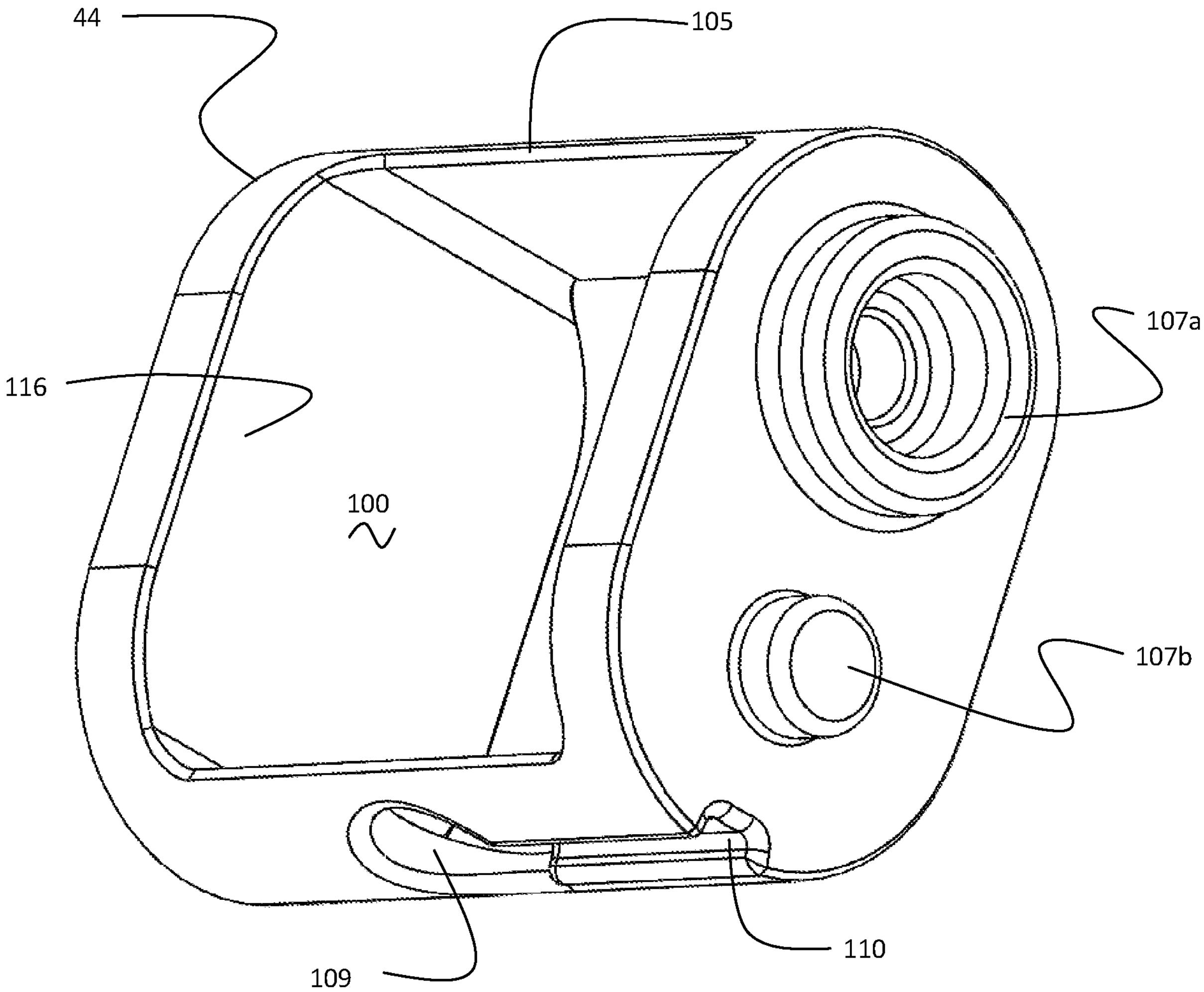

FIG. 13

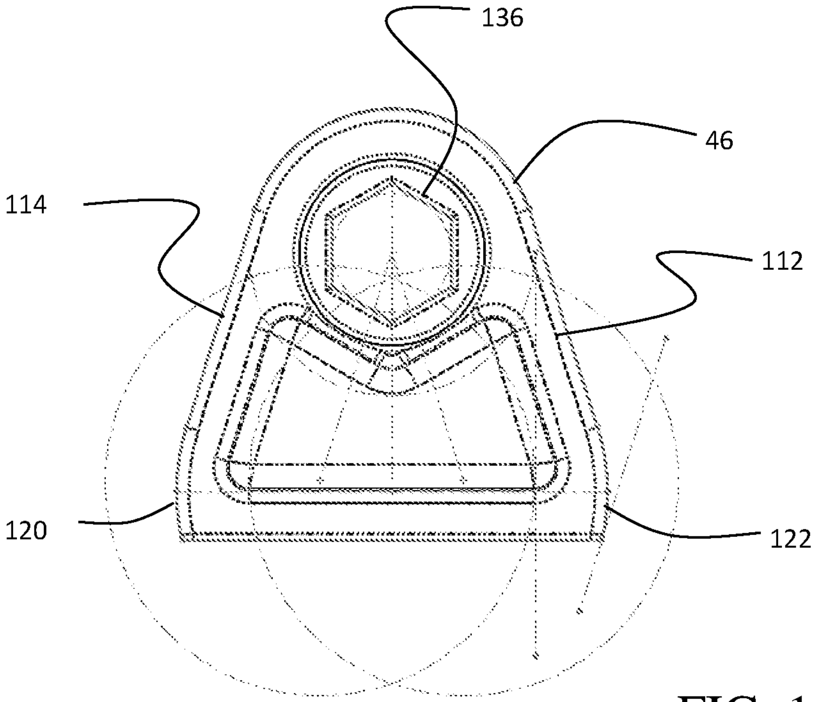

FIG. 14A

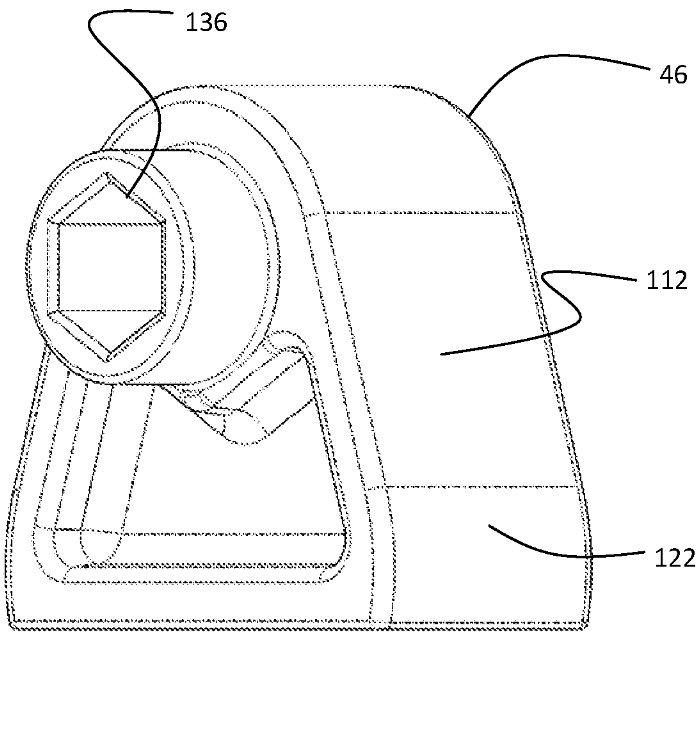

FIG. 14B

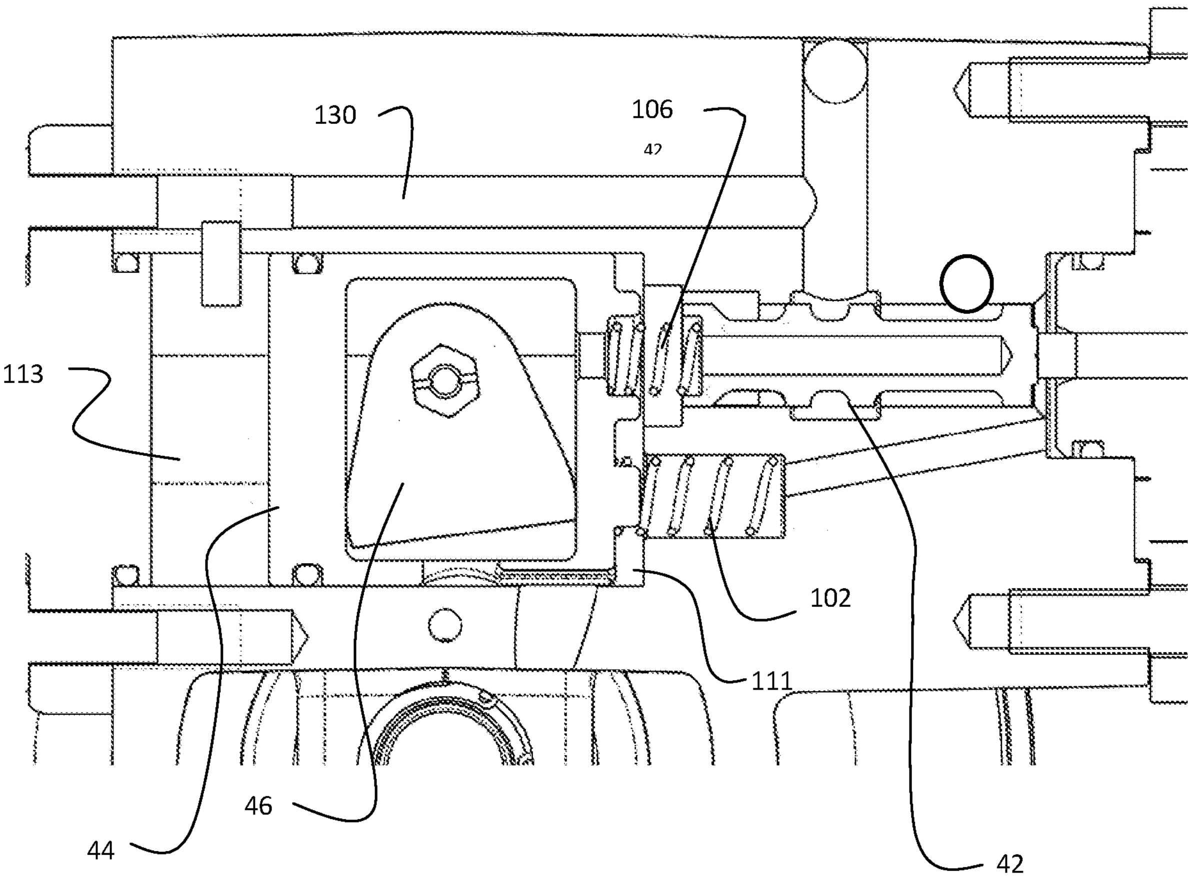

FIG. 15

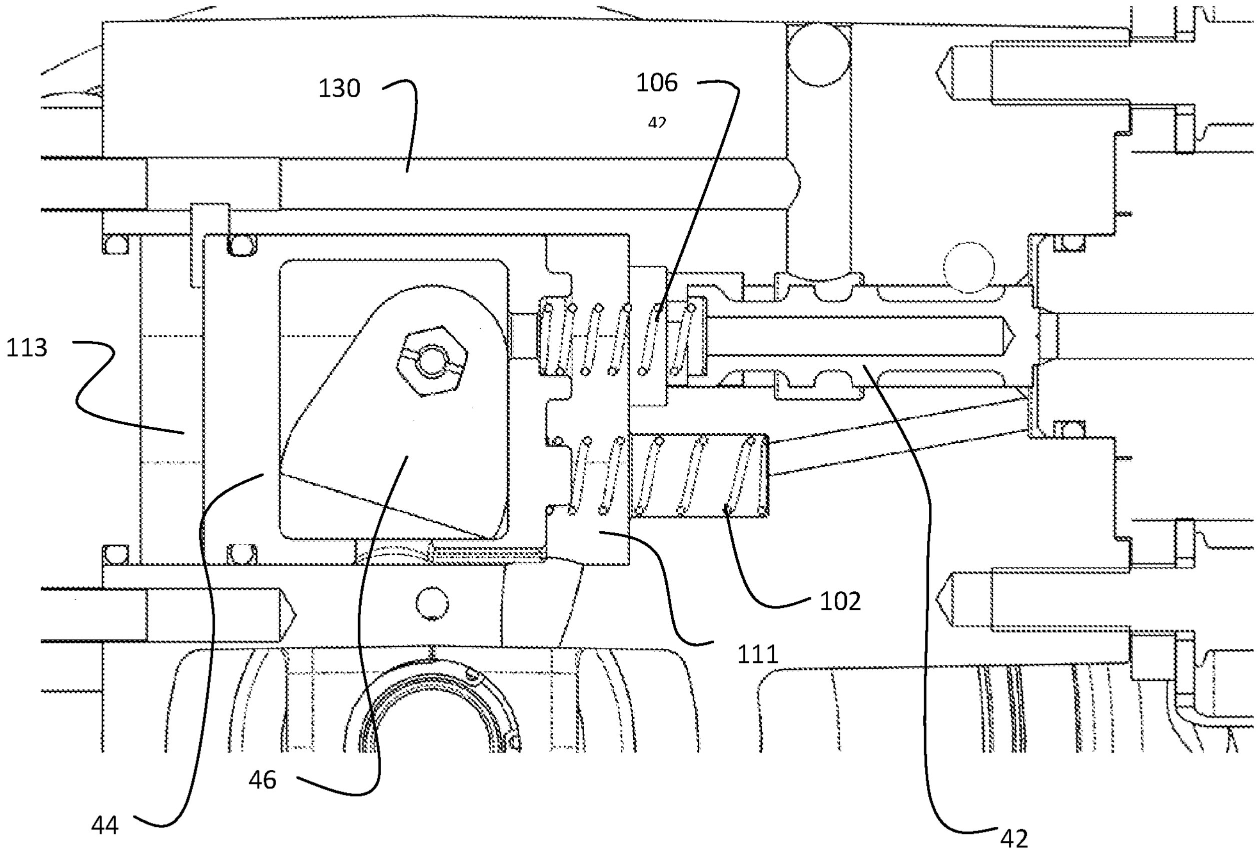

FIG. 16

INTERNATIONAL SEARCH REPORT

International application No.
PCT/CN2016/089146

A. CLASSIFICATION OF SUBJECT MATTER

F01D 17/16(2006.01)i

According to International Patent Classification (IPC) or to both national classification and IPC

B. FIELDS SEARCHED

Minimum documentation searched (classification system followed by classification symbols)

F01D

Documentation searched other than minimum documentation to the extent that such documents are included in the fields searched

Electronic data base consulted during the international search (name of data base and, where practicable, search terms used)

EPODOC,WPI,CNPAT,CNKI:turbocharger, geometry, VGT, airfoil?, variable, vane?, turbine, peg, piston, pocket

C. DOCUMENTS CONSIDERED TO BE RELEVANT

Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
X	CN 102296992 A (HONEYWELL INT INC) 28 December 2011 (2011-12-28) see description, paragraphs [0035]-[0036] and figure 6	1, 7, 8
X	CN 1451076 A (ALLIED SIGNEAL INC) 22 October 2003 (2003-10-22) see description, page 9, line 27 to page 10, line 8 and figures 9a-9e	17-19
A	CN 102297016 A (WUXI KAIDI BOOSTER DELER CO LTD) 28 December 2011 (2011-12-28) see the whole document	1-20
A	CN 103732881 A (CATERPILLAR INC) 16 April 2014 (2014-04-16) see the whole document	1-20
A	CN 101173614 A (HONEYWELL INT INC) 07 May 2008 (2008-05-07) see the whole document	1-20
A	US 2011171009 A1 (HONEYWELL INT INC) 14 July 2011 (2011-07-14) see the whole document	1-20
A	US 2008031728 A1 (SAUSSE LORRAIN ET AL.) 07 February 2008 (2008-02-07) see the whole document	1-20

☑ Further documents are listed in the continuation of Box C. ☑ See patent family annex.

* Special categories of cited documents:
- "A" document defining the general state of the art which is not considered to be of particular relevance
- "E" earlier application or patent but published on or after the international filing date
- "L" document which may throw doubts on priority claim(s) or which is cited to establish the publication date of another citation or other special reason (as specified)
- "O" document referring to an oral disclosure, use, exhibition or other means
- "P" document published prior to the international filing date but later than the priority date claimed
- "T" later document published after the international filing date or priority date and not in conflict with the application but cited to understand the principle or theory underlying the invention
- "X" document of particular relevance; the claimed invention cannot be considered novel or cannot be considered to involve an inventive step when the document is taken alone
- "Y" document of particular relevance; the claimed invention cannot be considered to involve an inventive step when the document is combined with one or more other such documents, such combination being obvious to a person skilled in the art
- "&" document member of the same patent family

Date of the actual completion of the international search	Date of mailing of the international search report
30 September 2016	**21 October 2016**

Name and mailing address of the ISA/CN	Authorized officer
STATE INTELLECTUAL PROPERTY OFFICE OF THE P.R.CHINA **6, Xitucheng Rd., Jimen Bridge, Haidian District, Beijing 100088** **China**	**WU,Fei**
Facsimile No. **(86-10)62019451**	Telephone No. **(86-10)62085294**

C. DOCUMENTS CONSIDERED TO BE RELEVANT

Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
A	EP 2397652 A2 (HONEYWELL INT INC) 21 December 2011 (2011-12-21) see the whole document	1-20

Box No. III Observations where unity of invention is lacking (Continuation of item 3 of first sheet)

This International Searching Authority found multiple inventions in this international application, as follows:

[1] Group 1 (Claims 1-16): a VGT for turbochargers with an airfoil;

[2] Group 2 (Claims 17-19): an integral actuation system for a VGT.

[3] The two inventions do not have the same or corresponding technical feature. Further, they do not have the same or corresponding special technical feature. Therefore the inventions are not so linked as to form a single general inventive concept, as required by PCT Rule 13.1, 13.2 and 13.3.

1. ☐ As all required additional search fees were timely paid by the applicant, this international search report covers all searchable claims.

2. ☑ As all searchable claims could be searched without effort justifying additional fees, this Authority did not invite payment of additional fees.

3. ☐ As only some of the required additional search fees were timely paid by the applicant, this international search report covers only those claims for which fees were paid, specifically claims Nos.:

4. ☐ No required additional search fees were timely paid by the applicant. Consequently, this international search report is restricted to the invention first mentioned in the claims; it is covered by claims Nos.:

Remark on Protest

☐ The additional search fees were accompanied by the applicant's protest and, where applicable, the payment of a protest fee.

☐ The additional search fees were accompanied by the applicant's protest but the applicable protest fee was not paid within the time limit specified in the invitation.

☐ No protest accompanied the payment of additional search fees.

INTERNATIONAL SEARCH REPORT
Information on patent family members

International application No.
PCT/CN2016/089146

Patent document cited in search report			Publication date (day/month/year)	Patent family member(s)			Publication date (day/month/year)
CN	102296992	A	28 December 2011	CN	102296992	B	23 September 2015
				EP	2397652	A2	21 December 2011
				EP	2397652	A3	17 December 2014
				US	2011312246	A1	22 December 2011
				US	8172508	B2	08 May 2012
CN	1451076	A	22 October 2003	CA	2349917	C	02 December 2008
				WO	0169045	A1	20 September 2001
				AT	313007	T	15 December 2005
				EP	1264078	A1	11 December 2002
				AU	3879600	A	24 September 2001
				DE	1264078	T1	26 June 2003
				DE	60024879	T2	20 July 2006
				EP	1264078	B1	14 December 2005
				AU	758433	B2	20 March 2003
				JP	4460814	B2	12 May 2010
				DE	60024879	D1	19 January 2006
				CA	2349917	A1	13 September 2001
				CN	1313711	C	02 May 2007
				JP	2003527522	A	16 September 2003
CN	102297016	A	28 December 2011	WO	2013023441	A1	21 February 2013
				DE	112012001868	T5	06 February 2014
				CA	2833790	A1	21 February 2013
				JP	2014521881	A	28 August 2014
				US	2014086725	A1	27 March 2014
				CN	102297016	B	12 December 2012
CN	103732881	A	16 April 2014	DE	112012002727	T5	13 March 2014
				US	8857178	B2	14 October 2014
				US	2013000300	A1	03 January 2013
				WO	2013003576	A1	03 January 2013
CN	101173614	A	07 May 2008	EP	1887189	A2	13 February 2008
				US	2008031728	A1	07 February 2008
US	2011171009	A1	14 July 2011	CN	102121401	A	13 July 2011
				US	8668443	B2	11 March 2014
				EP	2351911	A2	03 August 2011
				EP	2351911	A3	06 March 2013
				CN	102121401	B	04 May 2016
US	2008031728	A1	07 February 2008	EP	1887189	A2	13 February 2008
				CN	101173614	A	07 May 2008
EP	2397652	A2	21 December 2011	CN	102296992	B	23 September 2015
				CN	102296992	A	28 December 2011
				EP	2397652	A3	17 December 2014
				US	2011312246	A1	22 December 2011
				US	8172508	B2	08 May 2012