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(54) **SCREW COMPRESSOR WITH A SHUNT-ENHANCED COMPRESSION AND PULSATION TRAP (SECAPT)**

(57) A shunt-enhanced compression and pulsation trap (SECAPT) for a screw compressor assists internal compression (IC), reduces gas pulsation and NVH, and improves off-design efficiency, without using a slide valve and/or a serial pulsation dampener. The SECAPT includes an inner casing (e.g., an integral part of the compressor chamber) and an outer casing (e.g., surrounding part of the inner casing near the compressor discharge port) forming at least one diffusing chamber with a nozzle

and a feedback region that provides a feedback flow loop between the compressor chamber and the compressor discharge port. The SECAPT automatically compensates cavity pressure to meet different outlet pressures (hence eliminating under-compression and/or over-compression when the discharge port opens), partially recovers potential energy associated with the under-compression (UC), and traps and attenuates gas pulsations and noise before the discharge port opens.

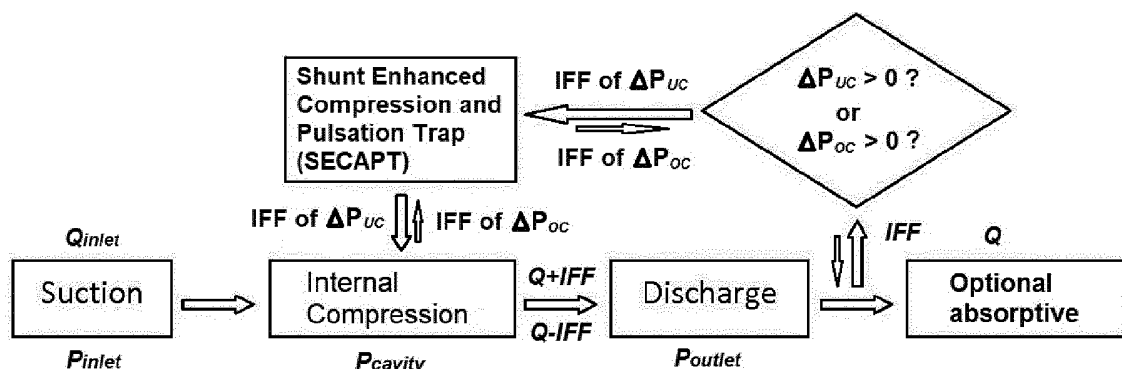


FIG.5

Description

TECHNICAL FIELD

[0001] The present invention relates generally to the field of rotary gas compressors, and more particularly relates to rotary screw compressors having twin meshing helical-shaped multi-lobe rotors.

BACKGROUND

[0002] A rotary screw compressor uses two helical screws, known as rotors, to compress the gas. In a dry running rotary screw compressor, a pair of timing gears ensures that the male and female rotors each maintain precise positions and clearances. In an oil-flooded rotary screw compressor, injected lubricating oil film fills the space between the rotors, both providing a hydraulic seal and transferring mechanical energy between the driving and driven rotor. Gas enters at the suction port of the compressor and gets trapped between moving threads and compressor casing forming a series of moving cavities as the screws rotate. Then the volumes of the moving cavities decrease and the gas is compressed. The gas exits at the end of the screw compressor through a discharge port normally connected to a discharge dampener to finish the cycle. It is essentially a positive displacement mechanism but using rotary screws instead of reciprocating motion so that displacement speed can be much higher. The result is a more continuous stream of flow with a more compact size when comparing with the traditional reciprocating types.

[0003] However, it has long been observed that screw compressors inherently generate gas pulsations with pocket passing frequency at discharge, and the pulsation amplitudes are especially significant when operating under high pressure and/or at off-design conditions of either an under-compression (UC) or an over-compression (OC). An under-compression, as shown in Figure 1c, happens when the gas pressure at the compressor outlet (discharge port) is greater than the gas pressure inside the compressor cavity just before the discharge port opening. This results in an "explosive" backflow of the gas from the outlet into the cavity as illustrated in Figure 1a. On the other hand, an over-compression, as shown in Figure 1d, takes place when the pressure at the compressor outlet is smaller than the pressure inside the compressor cavity just before the discharge port opening, causing an "explosive" forward flow of the gas from the cavity into the outlet illustrated in Figure 1b. All fixed pressure ratio positive displacement compressors suffer from the under-compression and/or over-compression due to the impossibility of matching one fixed pressure ratio to varying system back pressures. Typical applications with variable pressure ratios include various refrigeration and heat pump systems, and vacuum pump. For example, when ambient temperature rises or falls, the pressure ratios used in the refrigeration and heat pump systems

have to change accordingly. Often, the range of the pressure ratio variation is significant and the effects of OC and UC are further enhanced by the elevated pressures that refrigerant needs to operate. Another example of requiring a wide range of operating pressure ratios is the vacuum pump that is used to pull down the vacuum level in a system (for example, to pump air from a vessel to atmosphere), continuously increasing the pressure ratio as the vacuum level gets higher and higher. For these applications, the UC and OC induced energy losses and gas pulsations are significant, especially the later one, if left undampened, can potentially damage downstream pipelines, equipment and induce severe vibrations and noise within the compressor system.

[0004] To address the after-effects of the mismatch problem, a large pulsation dampener known in the trade as reactive and/or absorptive type as shown in Figure 2a, is usually required at the discharge side of a screw compressor to dampen the gas pulsations and induced NVH. It is generally very effective in gas pulsation control with a reduction of 20-40 dB but is large in size and causes other problems such as inducing more noises due to additional vibrating surfaces, or sometimes causes dampener structure fatigue failures that could result in catastrophic damages to downstream components and equipment. At the same time, discharge dampeners used today create high pressure losses as illustrated in Figure 2b that contribute to poor compressor overall efficiency. For this reason, screw compressors are often cited unfavorably with high gas pulsations, high NVH and low off-design efficiency and bulky size when compared with dynamic types like the centrifugal compressor.

[0005] To overcome the mismatch problem at source, a concept called slide valve has been explored widely since 1960s as demonstrated in Figures 3a-3b. For example, the slide valve concepts are disclosed in US Patent number 3,088,659 to H. R. Nilsson et al and entitled "Means for Regulating Helical Rotary Piston Engine" 4,215,977, or in US Patent number 3,936,239 to David. N. Shaw and entitled "Under-compression and Over-compression Free Helical Screw Rotary Compressor". The idea, often called variable V_i scheme, is to use a slide valve to mechanically vary the internal volume ratio hence compression ratio of the compressor to meet different operating pressure requirements, and to eliminate the under-compression and/or over-compression that are the source of discharge gas pulsations and energy losses. However, these systems typically are very complicated structurally with high cost and low reliability. Moreover, they do not work well for widely used dry screw applications where lubrication is essential between sliding parts.

[0006] In an effort to achieve the same goal of the slide valve variable V_i idea but without its complexity and limitation of applications, a shunt pulsation trap (SPT) technology as shown in Figures 4a-4b was disclosed for example in several co-owned patents (U.S. Patents Nos. 9,140,260; 9,155,292; 9,140,261; 9,243,557; 9,555,342;

and 9,732,754). The idea is to use fluidly gas to compensate the variable load conditions rather than moving the solidly mechanical parts that are sensitive to friction, fatigue failure and response frequency. SPT is capable of achieving the same goal of the slide valve by an automatic feedback flow loop both to communicate between the compressor cavity and outlet (discharge port) and to compensate the cavity compression by adding or subtracting gases (just like inflating or deflating a basketball) in such a way as to eliminate the under-compression or over-compression when discharge port opens. Conventional SPT technology is effective in under-compression mode for suppressing low-frequency pressure pulsation levels and reducing energy consumption by the elimination of back-pressure loss inherent with serial dampening. However, it does not work well in over-compression mode, especially for screw compressors operating over a wide range of pressure ratios.

[0007] Accordingly, it is always desirable to provide a new design and construction of a screw compressor that is capable of achieving high gas pulsation and NVH reduction at source and improving compressor off-design efficiency without externally connected silencer at discharge or using a slide valve while being kept compact in size and suitable for operating reliably for high efficiency, variable pressure ratio applications at the same time.

SUMMARY

[0008] Generally described, the present invention relates to a shunt enhanced compression and pulsation trap (SECAPT) for screw compressor having a compression chamber with a suction port and a discharge port, and a pair of multi-helical-lobe rotors housed in the compression chamber forming a series of moving cavities for trapping, compressing and propelling the trapped gas in the cavities from the suction port to discharge port. The SECAPT comprises an inner casing as an integral part of the compression chamber, and an outer casing surrounding part of the inner casing near the discharge port forming at least one diffusing chamber, therein housed at least one feedback flow loop through at least one flow nozzle (located at one of the moving cavities at least one male lobe span away or totally isolated from the suction port) to communicate between the propelled moving cavities and the discharge port. In this way, the SECAPT automatically compensates cavity pressure, in a similar way as inflating or deflating a basketball by adding or subtracting gas to the cavity, to meet different outlet pressures (hence eliminating the under-compression and/or over-compression when the discharge port opens), partially recovers the potential energy associated with the under-compression (UC), and traps and attenuates gas pulsations and noise before the discharge port opens.

[0009] These and other aspects, features, and advantages of the invention will be understood with reference to the drawing figures and detailed description herein, and will be realized by means of the various elements

and combinations particularly pointed out in the appended claims. It is to be understood that both the foregoing summary and the following brief description of the drawings and detailed description of the example embodiments are explanatory of example embodiments of the invention, and are not restrictive of the invention, as claimed.

BRIEF DESCRIPTION OF THE DRAWINGS

[0010]

FIGS. 1a and 1b are a cross sectional view showing the triggering mechanism of gas pulsation generation at the compressor discharge for an under-compression and an over-compression condition for a prior-art screw compressor.

FIGS. 1c and 1d are P-V diagrams of the associated energy losses for an under-compression and an over-compression condition for a prior-art screw compressor.

FIG. 2a shows the phases of a prior-art compression cycle of a screw compressor with a serial discharge dampener.

FIG. 2b is a P-V diagram of the associated energy losses at the compressor discharge for prior-art serial dampening (with back pressure).

FIGS. 3a and 3b show a typical design of a prior-art screw compressor with a slide valve.

FIG. 4a shows a perspective view of a prior-art shunt pulsation trap (SPT).

FIG. 4b is a cross-sectional view of (A-A) section of prior-art shunt pulsation trap of FIG. 4a showing different shapes of optional injection port nozzles.

FIG. 5 is a flow chart of the phases of a compression cycle of shunt enhanced compression and pulsation traps (SECAPT) according to the present invention, showing an under-compression condition and an over-compression condition.

FIG. 6a is a cross-sectional view of a one-stage SECAPT according to a first example embodiment of the invention, showing an under-compression condition.

FIG. 6b is an unwrapped view of the one-stage SECAPT of FIG. 6a.

FIG. 6c is a cross-sectional view of the one-stage SECAPT of FIG. 6a, showing an over-compression condition.

FIG. 7a shows side and top cross-sectional views of a circular converging nozzle without shape transition and with the same cross-sectional area from the nozzle throat into the cavity of a SECAPT.

FIG. 7b shows side and top cross-sectional views of a circular converging nozzle with the cross-sectional shape transitioning from circular to rectangular while maintaining the same cross-sectional area from the nozzle throat into the cavity of the SECAPT.

FIG. 7c shows side and top cross-sectional views of a circular converging nozzle with the cross-sectional shape transition from circular to rectangular while increasing the cross-sectional area (diverging) from the nozzle throat into the cavity of a SECAPT.

FIG. 8a is a cross-sectional view of a two-stage SECAPT according to a second example embodiment, showing an under-compression condition for both stages.

Fig. 8b is an unwrapped view of the two-stage SECAPT of FIG. 8a.

FIG. 8c is a cross-sectional view of the two-stage SECAPT of FIG. 8a, showing an over-compression condition.

FIG. 8d is a cross-sectional view of the two-stage SECAPT of FIG. 8a, showing an under-compression condition for the first stage and an over-compression condition for the second stage.

FIG. 9a is a cross-sectional view of a one-stage SECAPT according to a third example embodiment, showing the SECAPT in a deep vacuum mode.

FIG. 9b is a cross-sectional view of a two-stage SECAPT according to a fourth example embodiment, showing the SECAPT in a deep vacuum mode.

DETAILED DESCRIPTION OF EXAMPLE EMBODIMENTS

[0011] Although specific embodiments of the present invention will now be described with reference to the drawings, it should be understood that such embodiments are examples only and merely illustrative of but a small number of the many possible specific embodiments which can represent applications of the principles of the present invention. Various changes and modifications obvious to one skilled in the art to which the present invention pertains are deemed to be within the spirit, scope and contemplation of the present invention as further defined in the appended claims.

[0012] It should also be pointed out that though drawing illustrations and description are devoted to a dual

rotor screw compressor for enhancing gas compression and attenuating gas pulsations in the present invention, the principle can be applied to screw vacuum pump and/or other rotor combinations such as a single rotor screw or a tri-rotor screw. The principle can also be applied to other media such as gas-liquid two phase flow as widely used oil-injected screws for refrigeration. In addition, screw expanders are another variation except being used to generate shaft power from a media pressure drop.

[0013] To illustrate the principles of the present invention, Figure 5 is a flow chart of a screw compression cycle with the addition of a shunt enhanced compression and pulsation trap (SECAPT) according to example embodiments of the present invention, linking the internal compression phase to the discharge pressure. In broad terms, a SECAPT is used to assist internal compression (IC), to trap and attenuate gas pulsations and noises, and to improve off-design efficiency, without using a slide valve and/or a traditional serial pulsation dampener. As illustrated in Figure 5, a SECAPT involves modifications to a standard screw compression cycle from a serial mode, that is, from internal compression and dampening in series as shown in the prior art of Figure 2a, to a parallel mode where IC and SECAPT are carried out simultaneously and synergistically during a much longer time interval. Any deviation of the pressure in the compressor cavity from the target outlet pressure, either due to an under-compression $\Delta P_{UC} (=P_{outlet} - P_{cavity})$ or an over-compression $\Delta P_{OC} (=P_{cavity} - P_{outlet})$, would immediately trigger a feedback flow in the form of induced fluid flow (IFF) between the cavity and outlet that adds or subtracts extra gas molecules to or from the cavity in such a way as to diminish the pressure difference (ΔP) BEFORE the discharge valve opens. This way of compensation of the screw cavity pressure is somewhat similar to inflating or deflating a basketball by injecting or releasing gas into or from the cavity. By the compounded compression scheme of IC and SECAPT, any UC or OC pressure deficit or build-up at the compressor discharge will be minimized so that there would be no need to use a downstream dampener (However, an optional absorptive silencer could be used if flow induced broadband noise needs to be attenuated, say for vacuum applications when gas is discharged to atmosphere).

[0014] Referring to FIGS. 6a to 6c, there is shown a typical arrangement of a screw compressor 10 with a shunt enhanced compression and pulsation trap (SECAPT) apparatus 50 according to a first example embodiment. Typically, the screw compressor 10 has two rotors 12 integrated with two rotor shafts 11, respectively, where rotor shaft 11 is driven by an external rotational driving mechanism (not shown). The rotors 12 are typically driven through a set of timing gears (in case of dry running) or they drive each other directly (for oil injected case). The twin rotors 12 are typically a pair of multi-helical-lobe rotors, one male and one female, housed in the compression chamber 32 forming a series of moving cavities such

as 38 and 39 for trapping, compressing, and propelling the trapped gas in the cavities 38 and 39 from a suction port 36 to a discharge port 37 of the compressor 10. The screw compressor 10 also has an inner casing 20 as an integral part of the compression chamber 32, wherein rotor shafts 11 are mounted on an internal bearing support structure (not shown). The casing structure further includes an outer casing 28 surrounding part of the inner casing 20 near the discharge port 37 forming at least one diffusing chamber 55.

[0015] As a novel and unique feature of the present invention, a SECAPT apparatus 50 is comprised of at least one flow nozzle (trap inlet) 51 branching off from the compression chamber 32 into at least one diffusing chamber 55 and a feedback region (trap outlet) 58 communicating with the compressor outlet 37. As shown in FIG. 6b, the starting line of the flow nozzle (trap inlet) 51 is located at one of the moving cavities 38 or 39 at least one lobe span (or a screw pitch t) away from the suction port 36 closing line and positioned as far away (distance d on FIG. 6b) from the rotating axis 11 as possible and directed at about the same direction as the direction of the rotating rotor 12 to assist rotating (e.g., positioned with a directional axis that is parallel to a tangent to the angular direction of the rotating rotors). FIG. 6b also shows two types of flow nozzles 51 used: on the left is a 2D nozzle with rectangular cross-sectional shape with a converging cross-sectional area distribution along the axis; and on the right are two 3D nozzles with circular cross-sectional shape with a converging cross-sectional area distribution along the axis. Fig. 6a shows the flow pattern for an under-compression mode where the large directional arrows 30 show the direction of the cavity flow as propelled by the rotors 12 from the suction port 36 to the discharge port 37 of the compressor 10, while feedback flow IFF 53 as indicated by the small directional arrows goes from the feedback region (trap outlet) 58 through the diffusing chamber 55, then converging to the flow nozzle (trap inlet) 51 and releasing into the cavity 39 that is open to the flow nozzle 51. On the other hand, Fig. 6c shows the flow pattern for an over-compression mode where the large directional arrows 30 still show the direction of the cavity flow as propelled by the rotors 12 from the suction port 36 to the discharge port 37 of the compressor 10, while feedback flow IFF 54 as indicated by the small arrows goes from the cavity 39 that is now opened to the flow nozzle 51 through the diffusing chamber 55, and releasing into the trap outlet 58 that merges with the discharge flow 30.

[0016] When a screw compressor 10 is equipped with the SECAPT apparatus 50 of the present invention, there exist both a reduction in the gas pulsation and induced noises transmitted from screw compressor outlet to downstream flow as well as an improvement in internal flow field (hence its adiabatic off-design efficiency) for under-compression and/or over-compression operations. The theory of operation underlying the SECAPT apparatus 50 of the present invention can be described as follows. As

illustrated in FIGS. 6a and 6b for an under-compression mode, the SECAPT is designed to assist the internal compression from the moment when the gas pressure of cavity 39 reaches a minimum P_i (but far below the maximum) operating pressure of an application. As the "moving cavity" 39 with gas pressure P_1 is suddenly opened to the trap inlet 51 of the SECAPT with pressure P_4 , a shock-tube-like reaction is triggered (as disclosed in the co-owned U.S. Patent No. 9,155,292). This generates, at the nozzle throat 51 where the sudden opening taking place, an instant gas pulsation in the form of CW-IFF-EW with CW (not shown) and IFF 53 going into the cavity 39 while EW (not shown) coming out of the nozzle 51 towards the trap outlet 58 and compressor discharge port 37.

[0017] There are several advantages provided by the SECAPT when compared to a screw compressor with serially connected traditional dampener. First of all, the required mass is more efficiently transported using a nozzle 51 into the "starved" or under-compressed cavity 39 to minimize fill-in time and pulsation generation at discharge. It can be seen that the required mass flow 53 is first "borrowed" from the outlet area 37 and then "returned" to the outlet area 37 by a shunt feedback flow loop as shown in FIG. 5 so that the induced flow 53 is not lost in the process. The amount of the feedback flow 53 is designed to compensate the internal compression before discharge in such a way that the pressure difference ΔP_{UC} or ΔP_{OC} would be eliminated or reduced close to nearly zero at discharge as shown in FIG. 5. Because the speed of the jet flow at the nozzle throat can be close or equal to the speed of sound for high ΔP_{UC} , much faster than the speed of moving cavity 39, it is possible for the scheme to work for high speed dry screw compressors where variable V_i design does not work well. Secondly from a noise-reduction point of view, using a nozzle 51 as a trap would isolate the high velocity jet noises inside the cavity 39 before discharging as long as the nozzle throat 51 is choked so that no CW and jet induced sound could escape or propagate upstream through the nozzle throat 51. When the nozzle throat 51 is NOT choked, the CW and jet noises inside the cavity 39 will be reduced greatly due to very small throat area for the noise to escape out. Furthermore, the velocity field on the diverging side of the nozzle 51 that is opened to the diffusing chamber 55 and downstream outlet 37 is of much lower velocity, hence the flow induced noises. Thirdly, from an energy conservation point of view, the traditionally lost work associated with UC, shown in prior-art FIG. 1c as the shaded area, could now be partially recovered because the high velocity jet flow 53 is now directed to assist to propel or impulse the rotor 12 with the maximum torque as shown in FIG. 6b, like a Pelton Wheel. In a conventional serial scheme shown in prior-art FIG. 2a, the backflow jet is generally in the direction against the rotor rotating, resulting in doing negative work for the compressor system.

[0018] On the other hand, the theory of operation underlying the SECAPT apparatus 50 for an over-compression

sion mode is different. As illustrated in FIGS. 6c and 6b, the SECAPT is designed to assist the internal compression from the moment when the gas pressure P_1 of cavity 39 is slightly over the minimum operating pressure P_2 at the outlet 37 of the compressor 10 of an application. As the "moving cavity" 39 with gas pressure P_1 is suddenly opened to the trap inlet 51 of the SECAPT with pressure P_2 which is only slightly lower than P_1 , a shock-tube-like reaction is NOT possible anymore due to the small disturbance $\Delta P_{OC} (= P_1 - P_2)$. Instead, only a small feedback flow IFF 54 (as indicated by the small directional arrows in FIG. 6c) is produced from the cavity 39 to the nozzle 51 through the diffusing chamber 55, and released into the trap outlet 58 that merges with the discharge flow 30. Since the internal compression is gradual in nature from the gradual volume reduction of the cavity 39, the induced flow IFF 54 is much smaller in magnitude than the induced flow IFF 53 in the case of under-compression; hence the flow 54 induced jet noises will be much lower for the over-compression case.

[0019] To facilitate and optimize the feedback flow 53 or 54 at the flow nozzle 51 in either direction between the cavity 39 and diffusing chamber 55, more than one nozzle can be used to feed both male and female sides of the cavity 39, and/or the nozzle/s can optionally be in the form of circular hole (3-dimensional nozzle) or slot (2-dimensional nozzle) arranged in parallel with the lobe seal line of the cavity 39 (for illustration purposes, both are shown in FIG. 6b). Moreover, if the circular cross-sectional shape of the nozzle is used, the throat 59 cross section can be designed to be circular/non-transitioned (FIG. 7a), or gradually transitioned to a slot shape into the cavity 39 and oriented generally along the cavity longer side which is parallel with the lobe seal line of the cavity 39 either with the same cross sectional area as the nozzle throat 59 (FIG. 7b) or with a gradually increased cross-sectional area resulting in so called de Laval nozzle (FIG. 7c). Replacing a circular cross-sectional shape (FIG. 7a) with a slot as shown in FIGS. 7b and 7c will also reduce the stage spacing defined as perpendicular to the rotor sealing line, hence gaining more timing for the second stage operation. Furthermore, the slot shape would help flow exchange of the oblong shaped cavity 39 with the diffusing chamber 55, and improve the efficiency of the feed-in flow 53 for under-compression or the feed-out flow 54 for over-compression condition, especially for high speed dry screw application.

[0020] If the range of the pressure ratio variation or the extent of OC and UC is small, a one-stage SECAPT is enough to cover the compounded compression phase when the distance between the nozzle 51 opening to discharge port 37 opening is smaller than one lobe span or screw pitch t as shown in FIG. 6b. However, for some applications where the range of pressure ratio variation or the extent of OC and UC is large, a two-stage SECAPT can be used to cover the compounded compression phase when the distance between the closing of the first nozzle opening to the discharge port opening is the same

or larger than one lobe span or screw pitch t . The principle is that each cavity should always be in communication with the compressor outlet at any instant after being connected, but cavities never communicate with each other. Based on this principle, the start of the 2nd stage nozzle should be located at least one screw pitch t away from the end of the 1st nozzle and within the last screw pitch before the discharge port opening. Likewise, if a two-stage SECAPT is not enough to cover the compounded compression phase, a three-stage SECAPT can be used.

[0021] Referring to FIGS. 8a to 8c, there is shown a typical arrangement of a two-stage SECAPT according to a second example embodiment of a screw compressor 10 with a shunt enhanced compression and pulsation trap (SECAPT) apparatus 60. The construction of the screw compressor 10 and the first stage of the SECAPT apparatus 60 can be the same as for the SECAPT apparatus 50 as discussed above. However, a second stage of SECAPT apparatus 60 is added which is further comprised of at least one flow nozzle 61 (trap inlet) branching off from the compression chamber 32 into at least one diffusing chamber 63 and connected to a feedback region (trap outlet) 68 communicating with the compressor outlet 37. As shown in FIG. 8b, the first nozzle 51 (trap inlet) is still located at least one lobe span (one screw pitch t) away from the suction port 36 closing line and the start of the second nozzle 61 is located at least one screw pitch t away from the closing of the first nozzle 51, both of which are positioned as far away (distance d on FIG. 8b) from the rotating axis 11 as possible and directed at the same direction as the rotating rotor 12 to assist its rotating. FIG. 8a shows the flow pattern for an under-compression mode for both stages where the large directional arrows 30 show the direction of the cavity flow as propelled by the rotors 12 from the suction port 36 to the discharge port 37 of the compressor 10, while feedback flows 53 and 63 as indicated by the small directional arrows goes from the feedback region (trap outlet) 58 through the diffusing chambers 55 and 65, then converging to the flow nozzles 51 and 61 and releasing into the cavities 38 and 39 respectively. On the other hand, FIG. 8c shows the flow pattern for an over-compression mode for both stages where the large directional arrows 30 still show the direction of the cavity flow as propelled by the rotors 12 from the suction port 36 to the discharge port 37 of the compressor 10, while the feedback flows 54 and 64 as indicated by the small directional arrows go from the cavities 38 and 39 that are now opened to the nozzles 51 and 61 through the diffusing chambers 55 and 65, and releasing into the trap outlets 58 and 68 that merge with the discharge flow 30. Furthermore, FIG. 8d shows the flow pattern of an under-compression condition for the first stage and an over-compression for the second stage where the feedback flow 54 for the first stage goes from the feedback region (trap outlet) 58 through the diffusing chamber 55, then to the flow nozzle 51, and is then released into the cavity 38, while the feedback flow 64 for the second stage goes from the cavity

39 that is now opened to the nozzle 61, through the diffusing chamber 65, and is released into the trap outlet 68 that merge with the discharge flow 30.

[0022] In addition to a two-port configuration for a screw compressor application discussed above for the first and second example embodiments, a three-port configuration can be used for a screw vacuum pump application for pulling deep vacuum. In a vacuum pump embodiment, the suction port of the compressor is connected to a process or a vessel where a deep vacuum is to be created while the outlet port of the compressor is connected through a silencer to atmosphere. In addition, a third port is added that is also open to atmosphere and allows cool atmospheric air into the compressor cavity through the SECAPT to extend the pressure ratio range, e.g., from about 4/1 to about 20/1 or more.

[0023] Referring to FIGS. 9a and 9b, there are shown typical arrangements of a one-stage and a two-stage SECAPT, according to third and fourth example embodiments, respectively, of a screw compressor 10 with a shunt enhanced compression and pulsation trap (SECAPT) apparatus 70 and 80, respectively. The difference of the construction of the screw compressor 10 with the SECAPT apparatus 70 and 80 relative to that of the SECAPT apparatus 50 and 60 (of the first and second embodiments) is that an access port (or region) 77 is included (instead of the feedback region) to connect the compressor cavity 38 and/or 39 directly with atmosphere 78 through the SECAPT apparatus 70 and 80 instead of merging with the compressor outlet 37. A typical mode of operation for a one-stage SECAPT 70, for example as shown in FIG. 9a, is first releasing flow (not shown) from the cavity 39 through the nozzle 51 then through the diffusing chambers 55 to the port 77 and into the atmosphere 78 when the operating pressure ratio is less than the design pressure ratio of the compressor 10 to get rid of the over-compression. Then the flow direction (not shown) is automatically switched to pulling cooler atmospheric air from port 77 through the diffusing chambers 55 and nozzle 51 into the compressor cavity 39 when the operating pressure ratio is more than the design pressure ratio of the compressor 10. The cool ambient air mixed with hotter cavity air after internal compression will allow the compressor to reach a much higher pressure ratio beyond its normal operating range, say from about 4/1 to about 20/1 or more.

[0024] As such, various embodiments of the invention provide advantages over the prior art. For example, a screw compressor with a shunt enhanced compression and pulsation trap (SECAPT) in parallel with the compressor internal compression helps eliminate the under-compression and/or over-compression (sources of discharge gas pulsations and energy losses) when discharge port opens. A screw compressor with a shunt enhanced compression and pulsation trap (SECAPT) can be as effective as a slide valve variable V_i design but without mechanical moving parts and limitation to oil-injected applications. A screw compressor with a shunt en-

hanced compression and pulsation trap (SECAPT) can be an integral part of the compressor casing so that it is compact in size by eliminating the serially connected pulsation dampener at discharge. A screw compressor with a shunt enhanced compression and pulsation trap (SECAPT) can be capable of achieving energy savings over a wide range of pressure ratios. A screw compressor with a shunt enhanced compression and pulsation trap (SECAPT) can be capable of achieving reduced gas pulsations and NVH over a wide range of pressure ratios. A screw compressor with a shunt enhanced compression and pulsation trap (SECAPT) can be capable of achieving energy savings and higher gas pulsation attenuation over a wide range of speed and cavity passing frequency. And a screw compressor with a shunt enhanced compression and pulsation trap (SECAPT) can be capable of achieving the same level of adiabatic off-design efficiency as a slide valve over a wide range of pressure and speed.

[0025] It is to be understood that this invention is not limited to the specific devices, methods, conditions, or parameters of the example embodiments described and/or shown herein, and that the terminology used herein is for the purpose of describing particular embodiments by way of example only. Thus, the terminology is intended to be broadly construed and is not intended to be unnecessarily limiting of the claimed invention. For example, as used in the specification including the appended claims, the singular forms "a," "an," and "the" include the plural, the term "or" means "and/or," and reference to a particular numerical value includes at least that particular value, unless the context clearly dictates otherwise. In addition, any methods described herein are not intended to be limited to the sequence of steps described but can be carried out in other sequences, unless expressly stated otherwise herein.

[0026] While the claimed invention has been shown and described in example forms, it will be apparent to those skilled in the art that many modifications, additions, and deletions can be made therein without departing from the spirit and scope of the invention as defined by the following claims.

Claims

1. A screw compressor, comprising:

a compression chamber and a pair of meshing multi-helical-lobe rotors housed within the compression chamber, wherein the compression chamber has a flow suction port and a flow discharge port, wherein the rotors rotate to cooperatively form a series of moving cavities within the compression chamber for trapping and compressing fluid and propelling the trapped fluid from the suction port to the discharge port; and a shunt-enhanced compression and pulsation

- trap (SECAPT) apparatus including a diffusing chamber having a first flow nozzle providing fluid communication between the moving cavities inside the compression chamber and the diffusing chamber and having a feedback region providing fluid communication between the diffusing chamber and the discharge port, wherein the SECAPT defines a first stage of a feedback flow loop, wherein in operation the SECAPT achieves high gas pulsation and NVH reduction and improved compressor off-design efficiency without using a serial pulsation dampener or a slide valve.
2. The screw compressor as claimed in claim 1, wherein the first flow nozzle is positioned at a distance at least one lobe span away, or is totally sealed or isolated, from the suction port, but is positioned before the discharge port.
 3. The screw compressor as claimed in claim 2, further comprising a second flow nozzle that is positioned at a distance at least one lobe span away, or totally sealed or isolated, from the first flow nozzle, but is positioned before the discharge port, and defining a second stage of the feedback flow loop.
 4. The screw compressor as claimed in claim 1, further comprising a third flow nozzle that is positioned at a distance at least one lobe span away, or totally sealed or isolated, from the second flow nozzle, but is positioned before the discharge port, and defining a third stage of the feedback flow loop.
 5. The screw compressor as claimed in claim 1, wherein the first flow nozzle has a circular cross-sectional shape with a converging or a converging-diverging cross-sectional area transitioning along an axis of the nozzle.
 6. The screw compressor as claimed in claim 1, wherein the first flow nozzle has a rectangular cross-sectional shape with a converging cross-sectional area transitioning along an axis of the nozzle.
 7. The screw compressor as claimed in claim 5, wherein the converging cross-sectional area has a continuous transition from a circular cross-sectional shape at a throat of the nozzle to a generally rectangular slot shape at the compression chamber, with a longer side of the rectangular slot shaped nozzle at the compression chamber oriented generally along a longer side of the moving cavity.
 8. The screw compressor as claimed in claim 5, wherein the converging-diverging cross-sectional area has a continuous transition from a circular cross-sectional shape at a throat of the nozzle to a generally rectangular slot shape at the compression chamber, with a longer side of the rectangular slot shaped nozzle at the compression chamber oriented generally along a longer side of the moving cavity.
 9. The screw compressor as claimed in claim 1, wherein the first flow nozzle is positioned a distance away from the rotor axis and aimed in generally the same direction as an angular rotation of one of the rotors.
 10. The screw compressor as claimed in claim 1, wherein the pair of meshing multi-helical-lobe rotors includes a male rotor and a female rotor, and wherein two of the first flow nozzles are provided with one first flow nozzle positioned at the male rotor and with the other first flow nozzle positioned at the female rotor, and wherein the two nozzles are open simultaneously to moving male and female cavities in the compression chamber.
 11. A screw compressor, comprising:
 - a compression chamber and a pair of meshing multi-helical-lobe rotors housed within the compression chamber, wherein the compression chamber as a flow suction port and a flow discharge port, wherein the rotors rotate to cooperatively form a series of moving compression cavities within the compression chamber for trapping and compressing fluid and propelling the trapped fluid from the suction port to the discharge port; and
 - a shunt-enhanced compression and pulsation trap (SECAPT) apparatus including a diffusing chamber having a first flow nozzle providing fluid communication between the moving cavities inside the compression chamber and the diffusing chamber and having an access port providing fluid communication between the diffusing chamber and ambient atmosphere, wherein the SECAPT defines a first stage of a feedback flow loop, wherein in operation the SECAPT achieves deep vacuum with high gas pulsation and NVH reduction and improved compressor off-design efficiency without using a slide valve.
 12. The screw compressor as claimed in claim 11, wherein the first flow nozzle is positioned at a distance at least one lobe span away, or is totally sealed or isolated, from the suction port, but is positioned before the discharge port.
 13. The screw compressor as claimed in claim 11, further comprising a second flow nozzle that is positioned at a distance at least one male lobe span away, or totally sealed or isolated, from the first flow nozzle, but is positioned before the discharge port,

and defining a second stage of the feedback flow loop.

14. The screw compressor as claimed in claim 11, wherein the first flow nozzle is positioned a distance away from the rotor axis and aimed ingenerally the same direction as an angular rotation of one of the rotors. 5
15. The screw compressor as claimed in claim 11, wherein the pair of meshing multi-helical-lobe rotors includes a male rotor and a female rotor, and wherein two of the first flow nozzles are provided with one first flow nozzle positioned at the male rotor and with the other first flow nozzle positioned at the female rotor, and wherein the two nozzles are open simultaneously to moving male and female cavities in the compression chamber. 10 15

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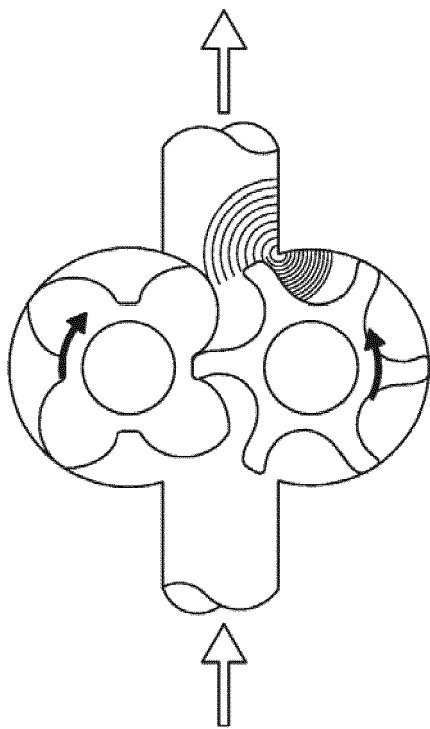


FIG. 1a

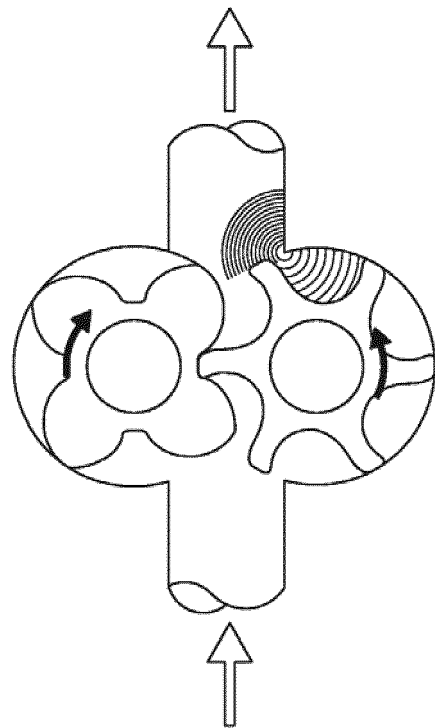


FIG. 1b

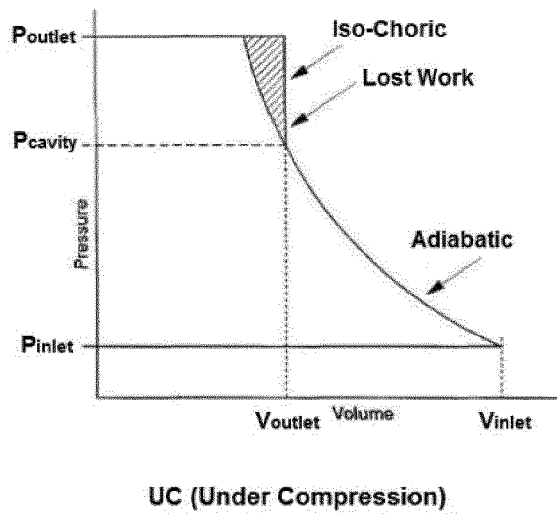


FIG.1c

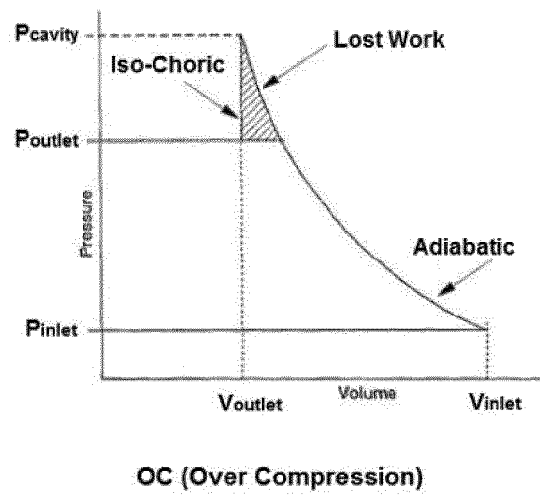


FIG.1d

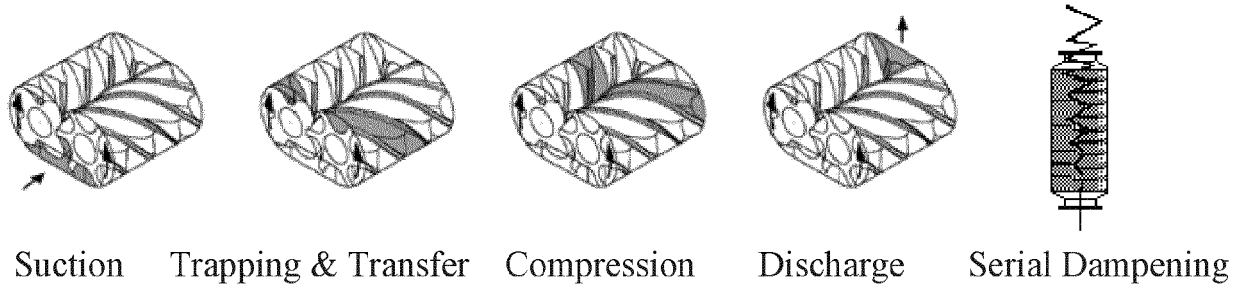


FIG.2a

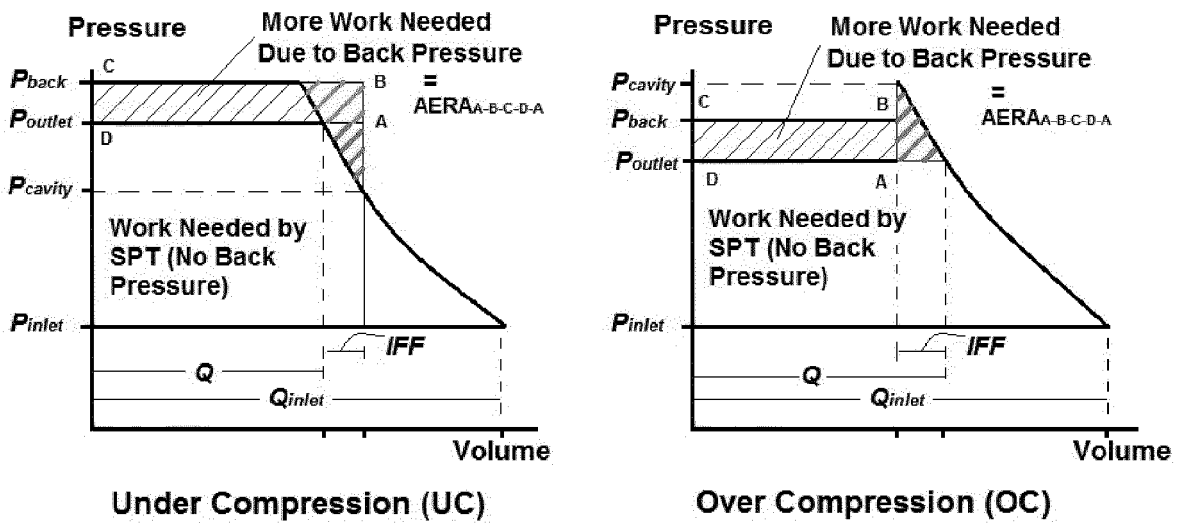


FIG.2b

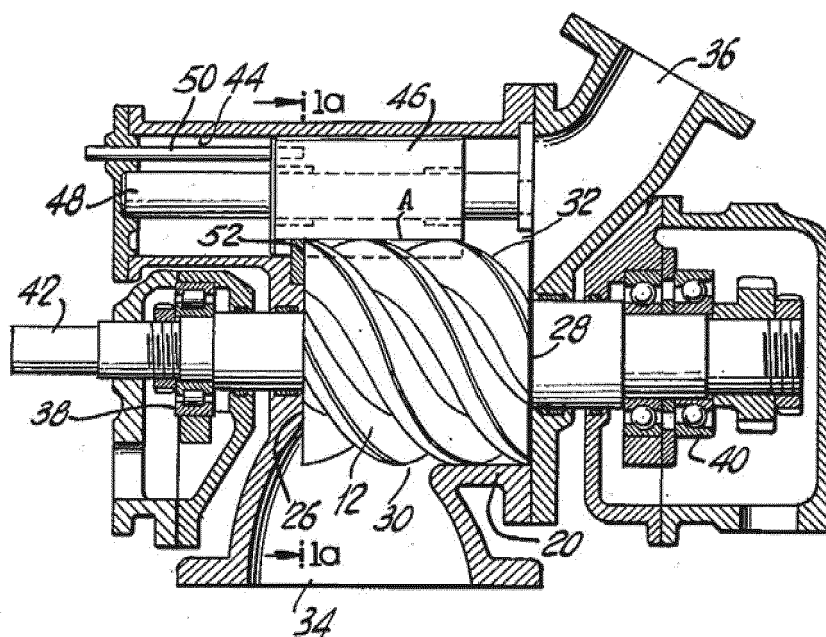


FIG. 3a

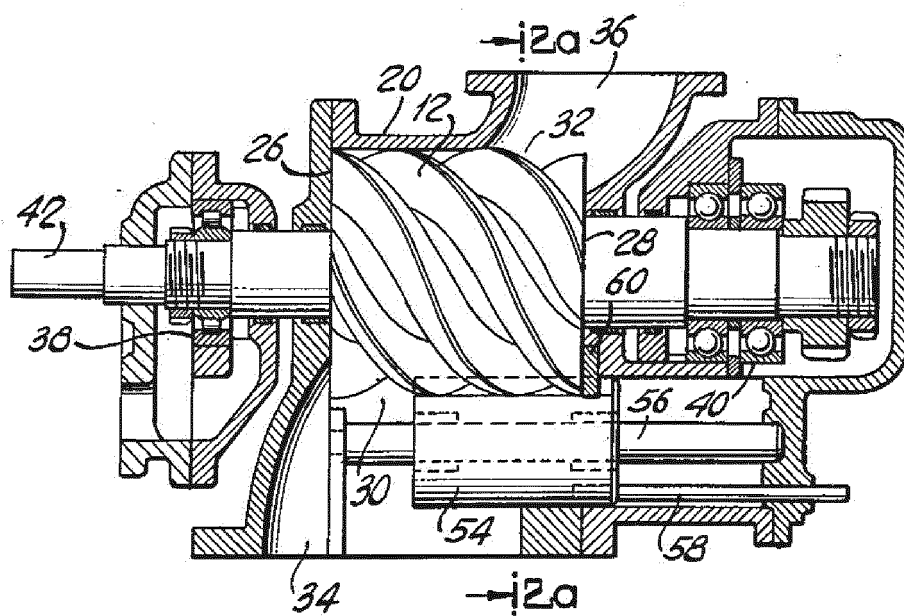


FIG. 3b

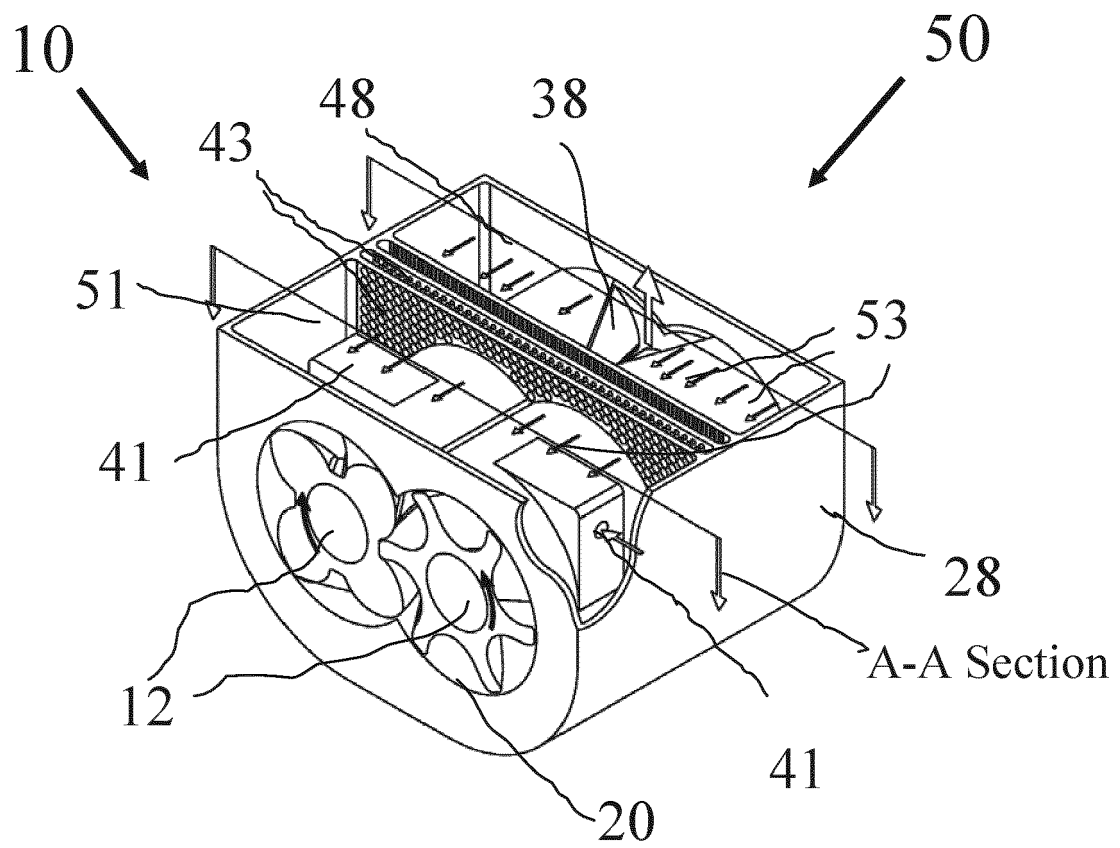


FIG.4a

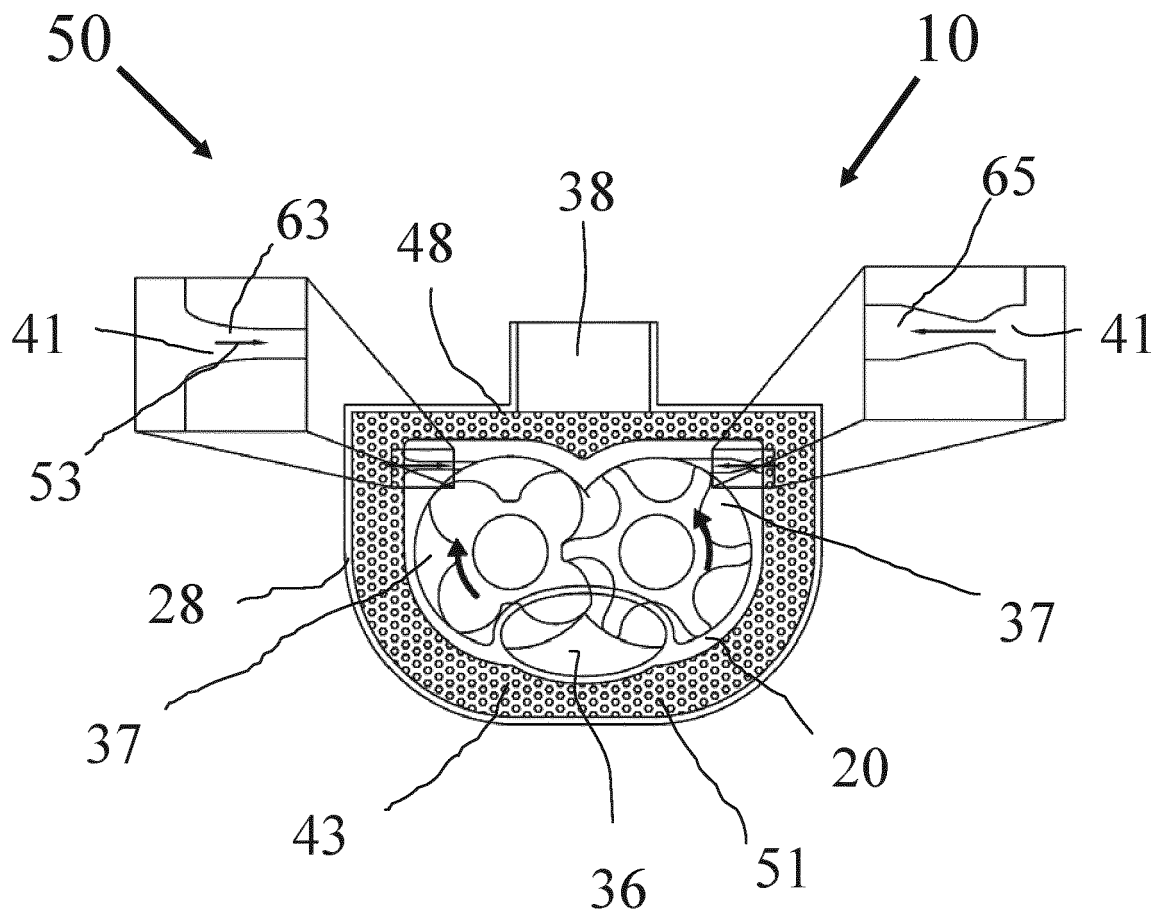


FIG.4b

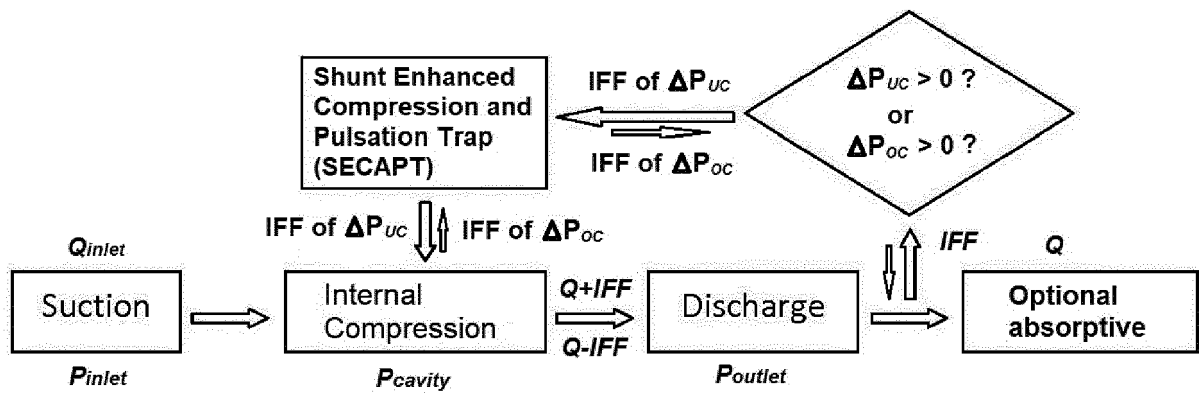


FIG.5

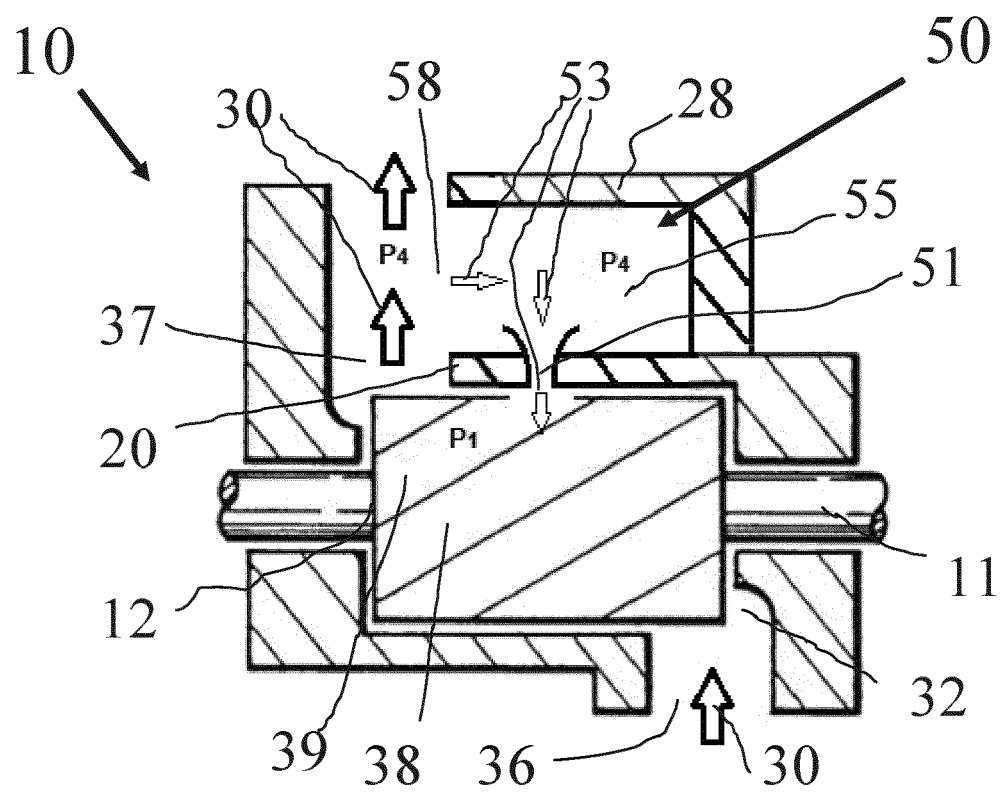


FIG.6a

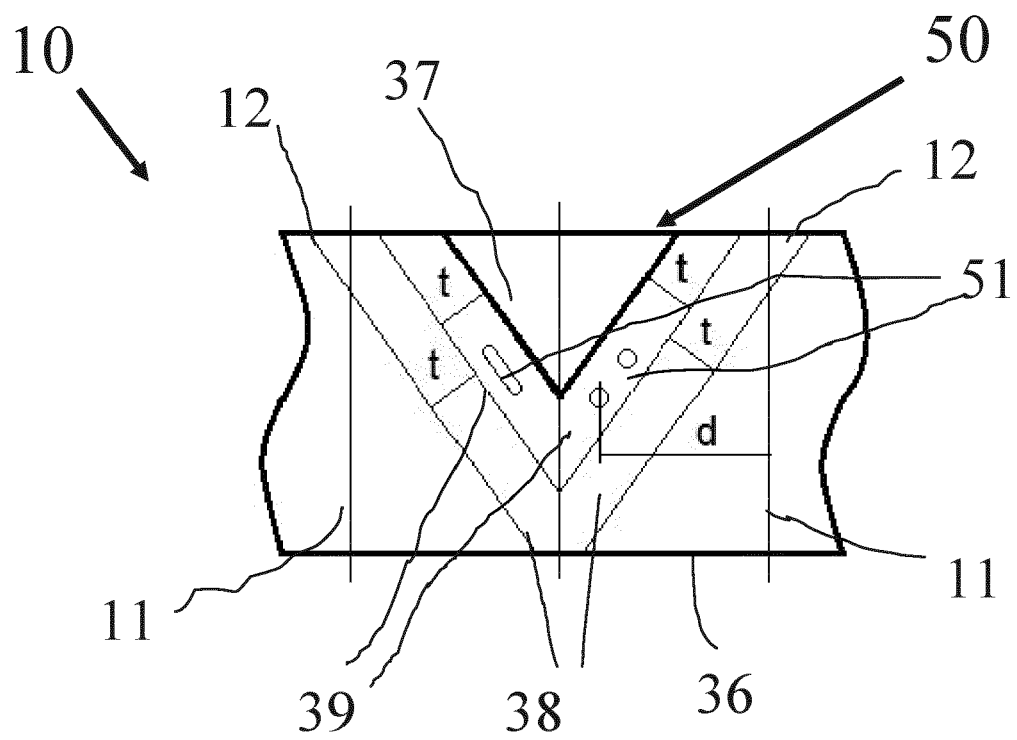


FIG.6b

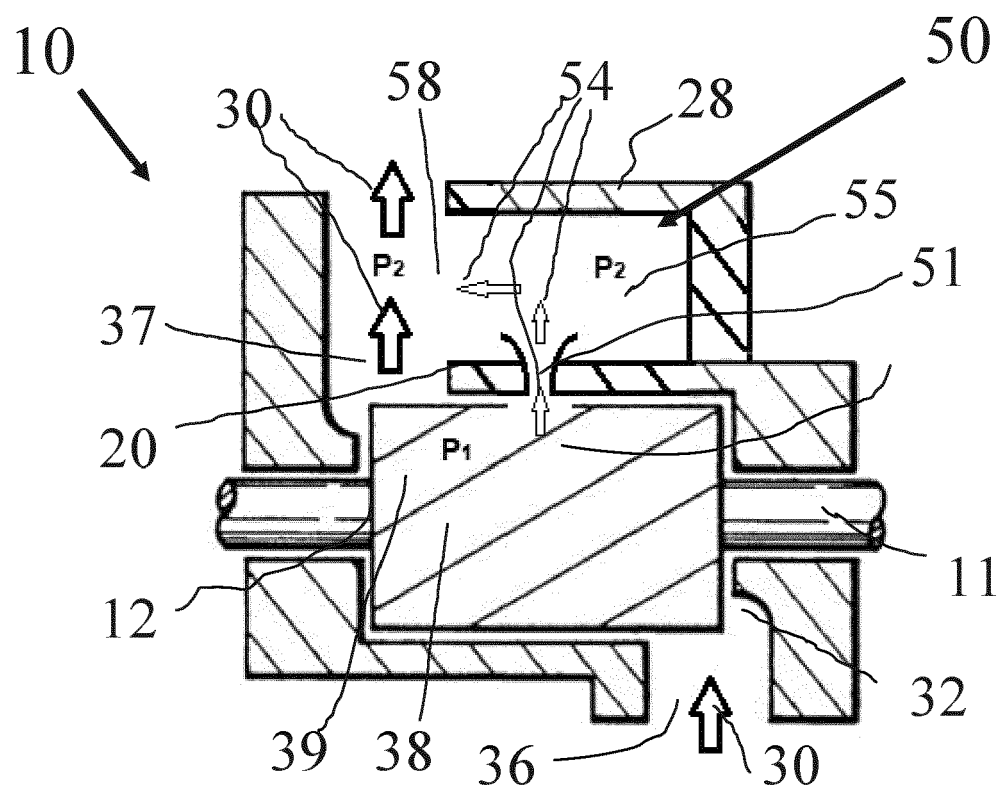
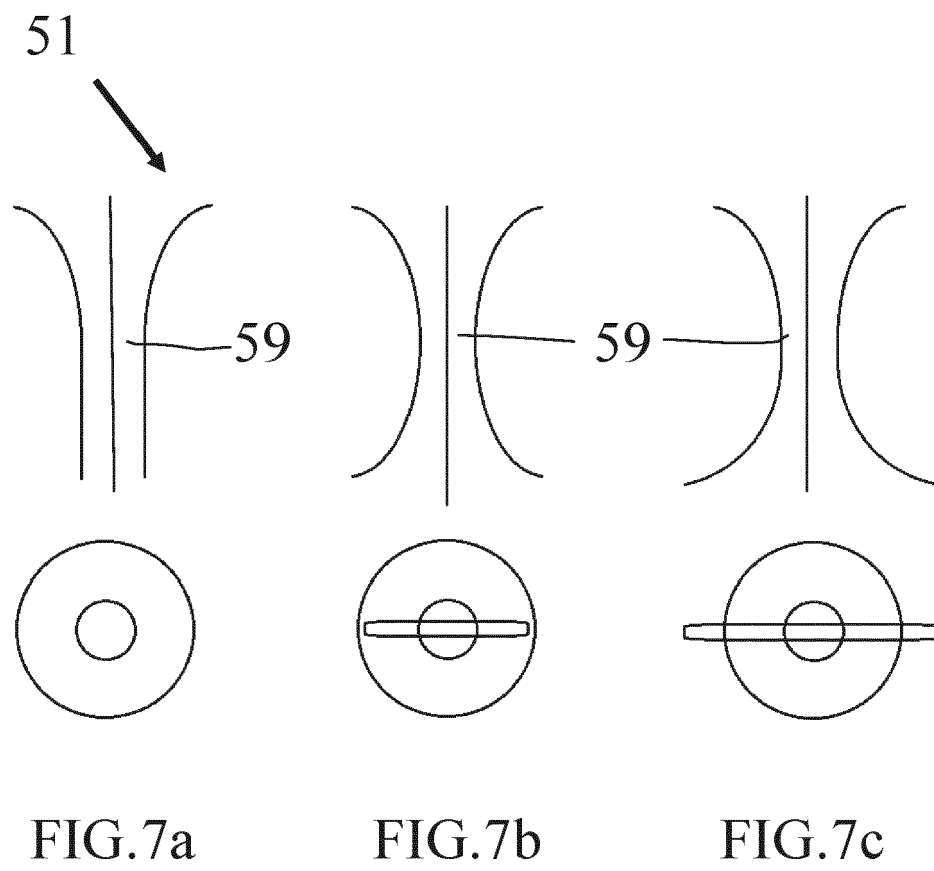


FIG.6c



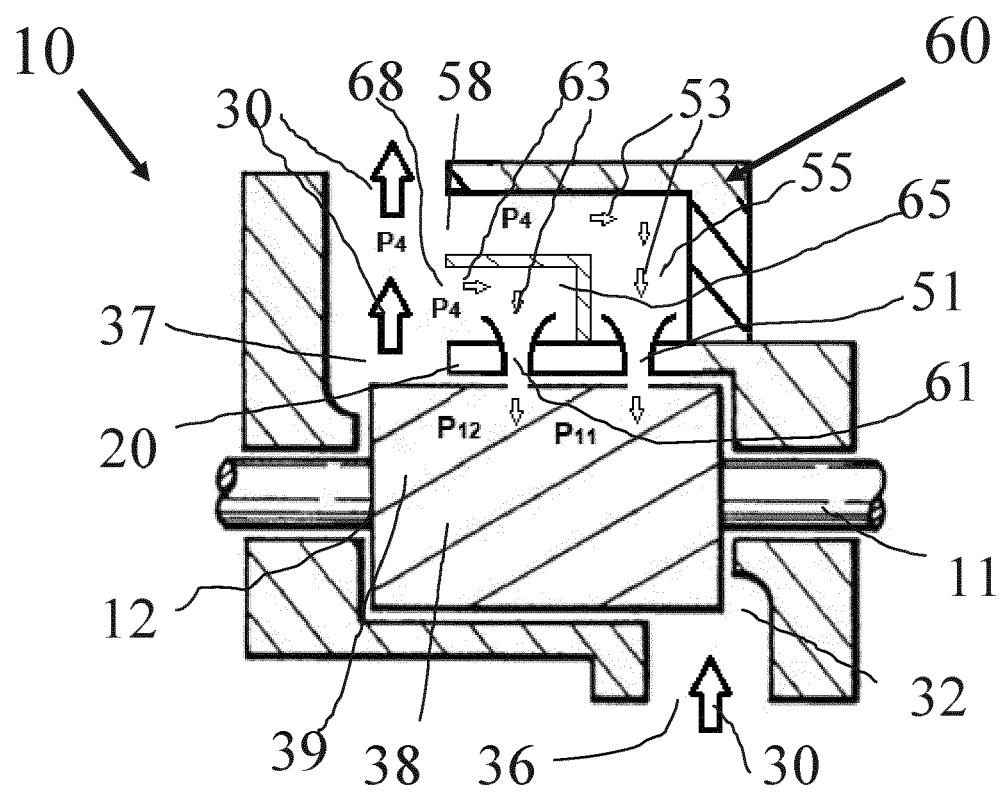


FIG.8a

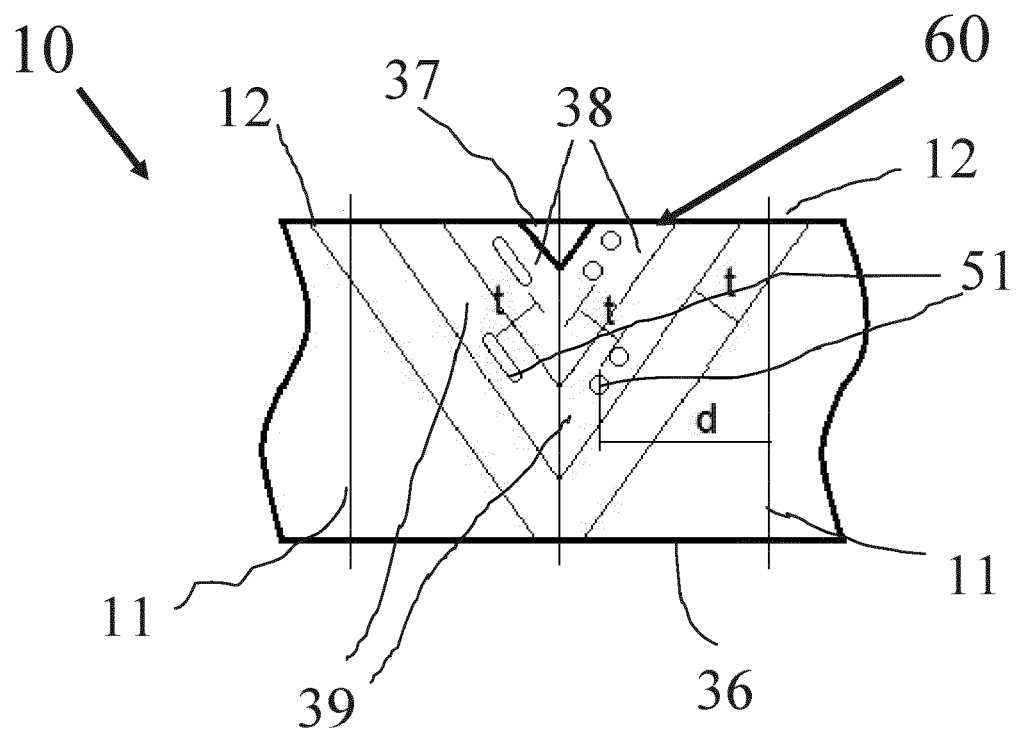


FIG.8b

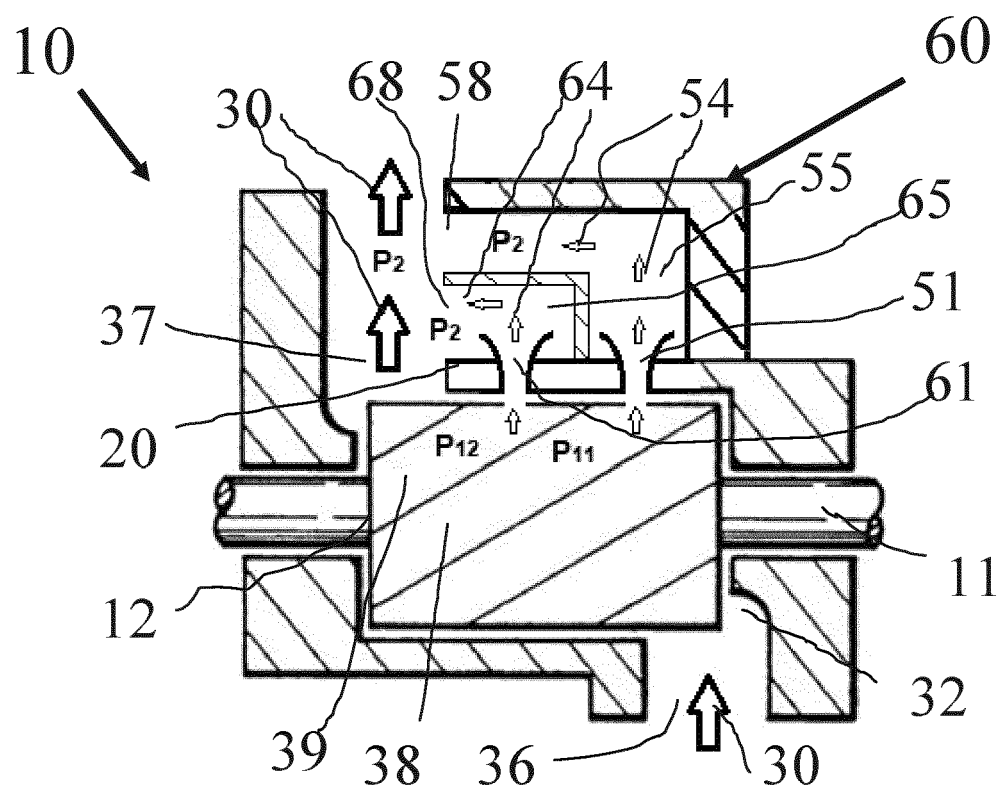


FIG.8c

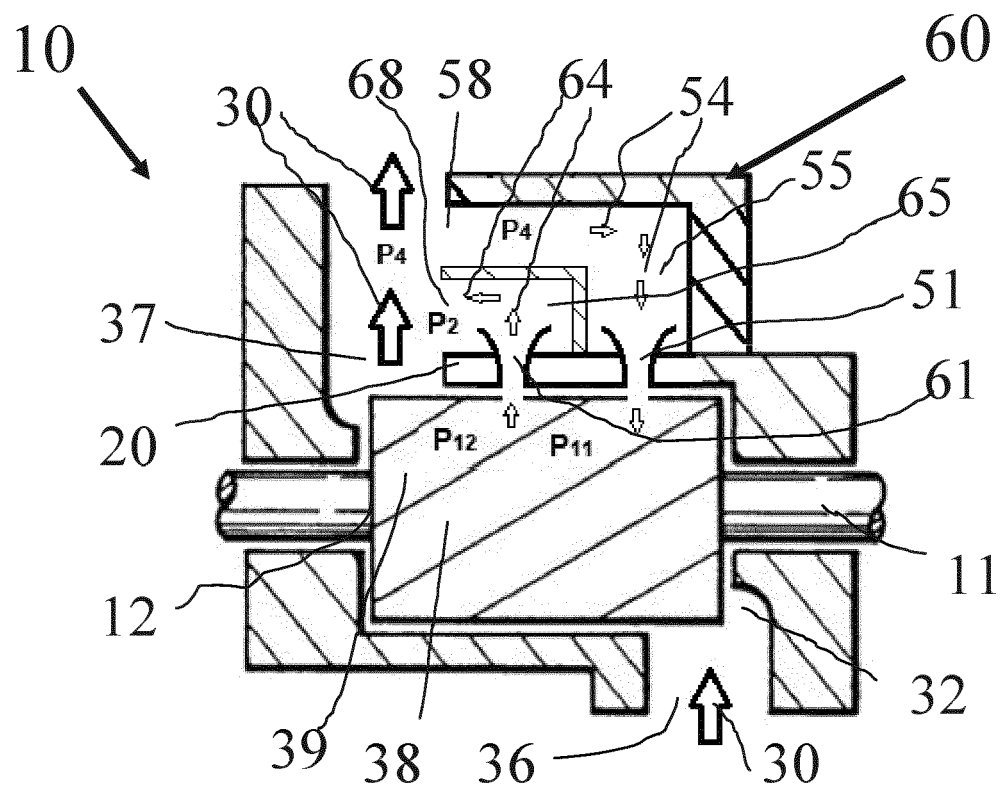


FIG.8d

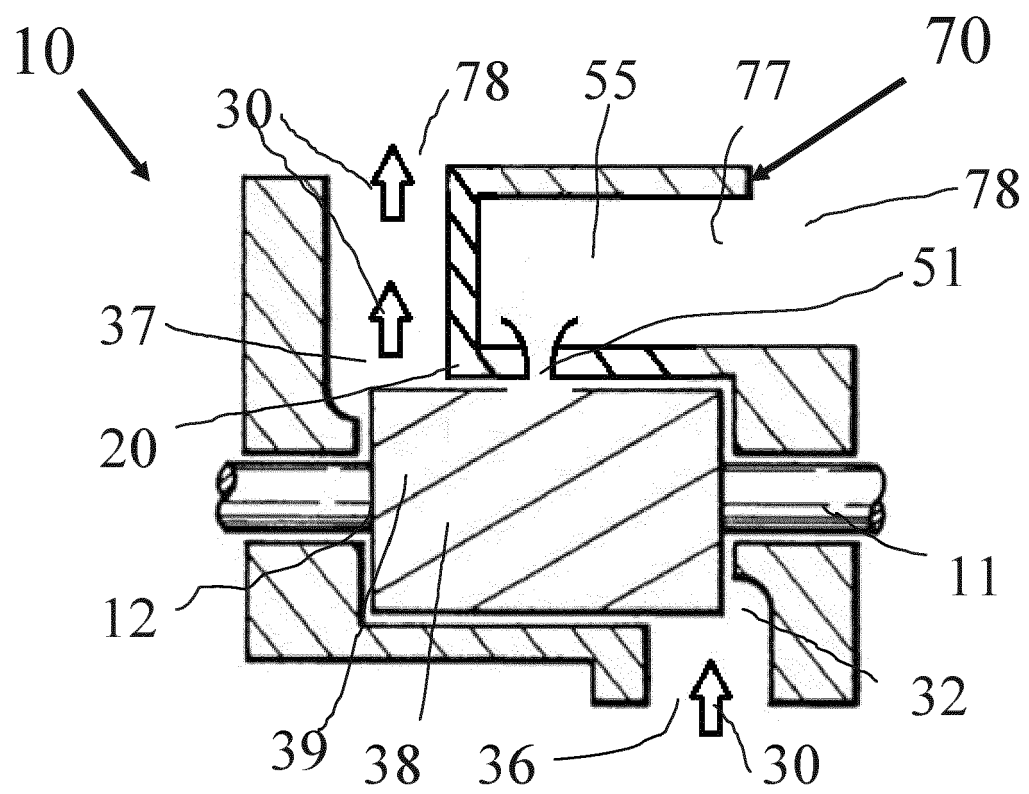


FIG.9a

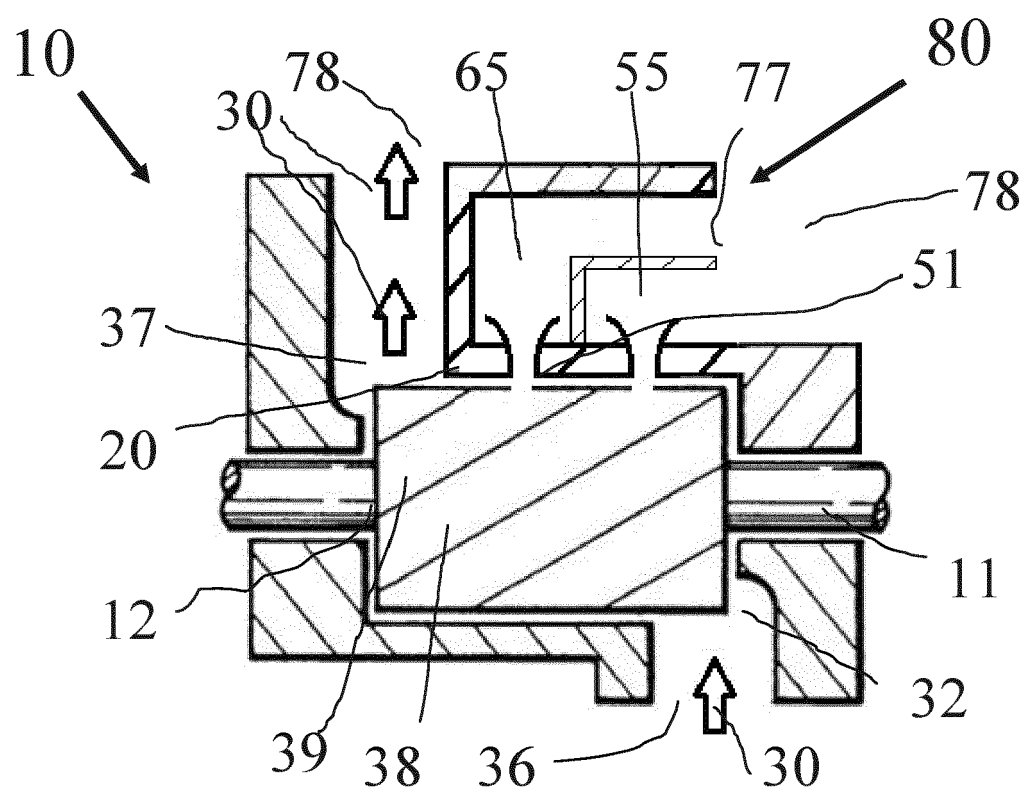


FIG.9b



EUROPEAN SEARCH REPORT

Application Number

EP 22 15 7685

DOCUMENTS CONSIDERED TO BE RELEVANT

Category	Citation of document with indication, where appropriate, of relevant passages	Relevant to claim	CLASSIFICATION OF THE APPLICATION (IPC)
X,D	US 9 140 260 B2 (HUANG PAUL XIUBAO [US]; YONKERS SEAN WILLIAM [US] ET AL.) 22 September 2015 (2015-09-22) * figures 7B, 8, 17 * * column 1, line 17 - line 27 * * column 8, line 15 - column 9, line 44 * * column 11, line 51 - column 12, line 3 * -----	1-15	INV. F04C18/08 F04C18/12 F04C29/06
X	CN 106 481 564 A (PAUL XIUBAO HUANG) 8 March 2017 (2017-03-08) * figure 5a * * paragraph [0069] * -----	1,2,5-10	
A	EP 3 670 916 A2 (INGERSOLL RAND CO [US]) 24 June 2020 (2020-06-24) * abstract * * figures * -----	1-15	
			TECHNICAL FIELDS SEARCHED (IPC)
			F04C
The present search report has been drawn up for all claims			
Place of search		Date of completion of the search	Examiner
Munich		13 September 2022	Durante, Andrea
CATEGORY OF CITED DOCUMENTS			
X : particularly relevant if taken alone Y : particularly relevant if combined with another document of the same category A : technological background O : non-written disclosure P : intermediate document			
T : theory or principle underlying the invention E : earlier patent document, but published on, or after the filing date D : document cited in the application L : document cited for other reasons & : member of the same patent family, corresponding document			

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ON EUROPEAN PATENT APPLICATION NO.**

EP 22 15 7685

5 This annex lists the patent family members relating to the patent documents cited in the above-mentioned European search report.
The members are as contained in the European Patent Office EDP file on
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13-09-2022

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EP 3670916	A2	24-06-2020	CN 111425394 A 17-07-2020
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