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Improved dual chamber displacement pumps

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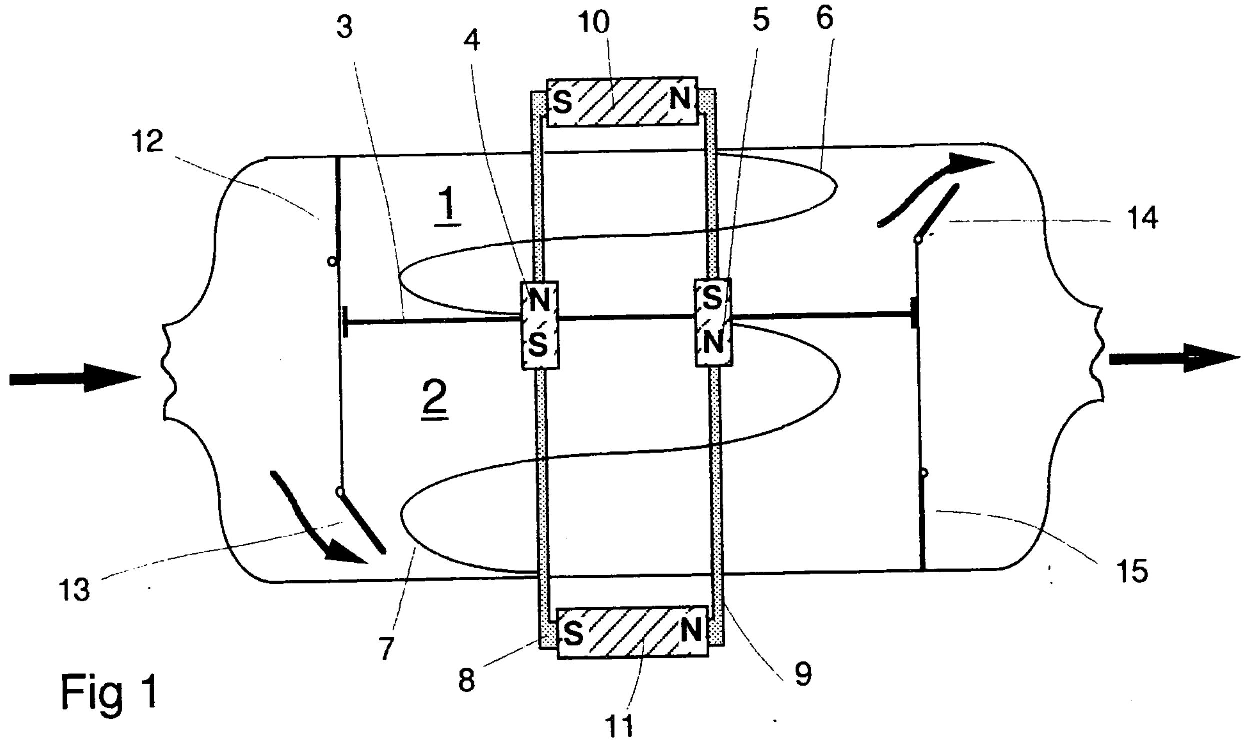
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Fig 1

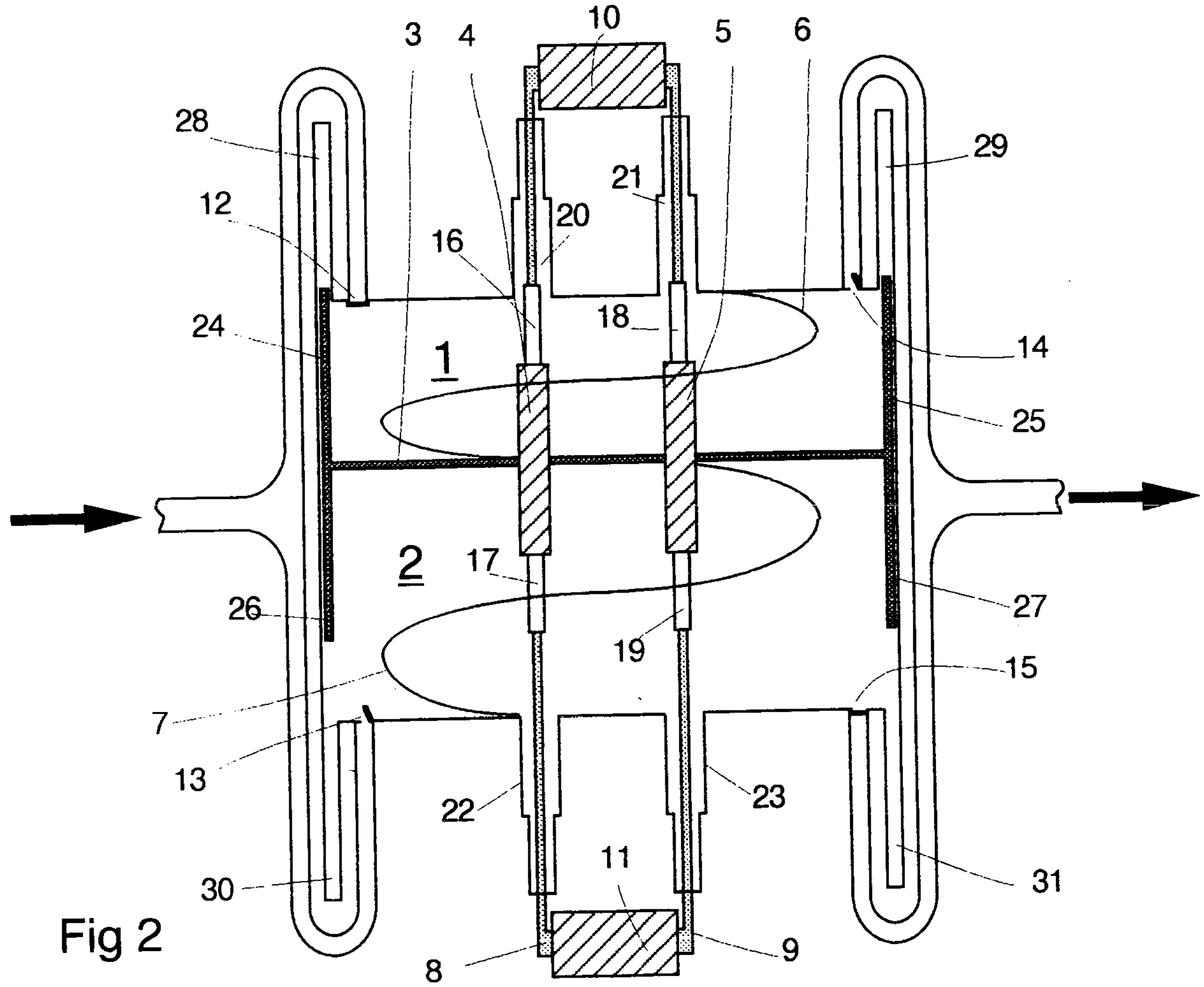
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Fig 2

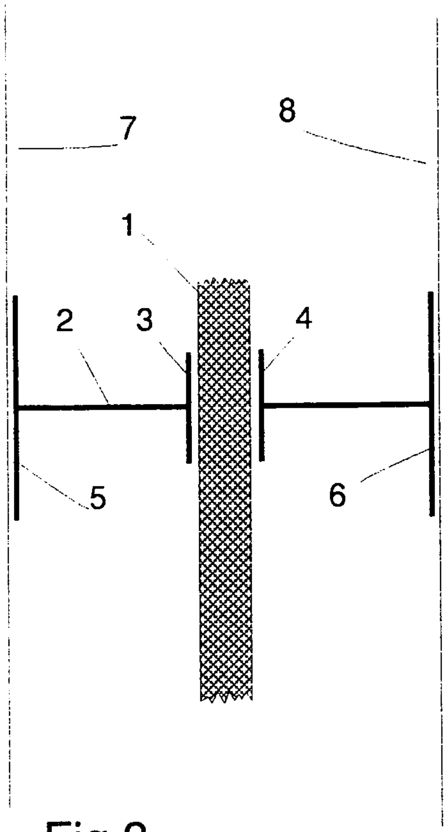

Fig 3

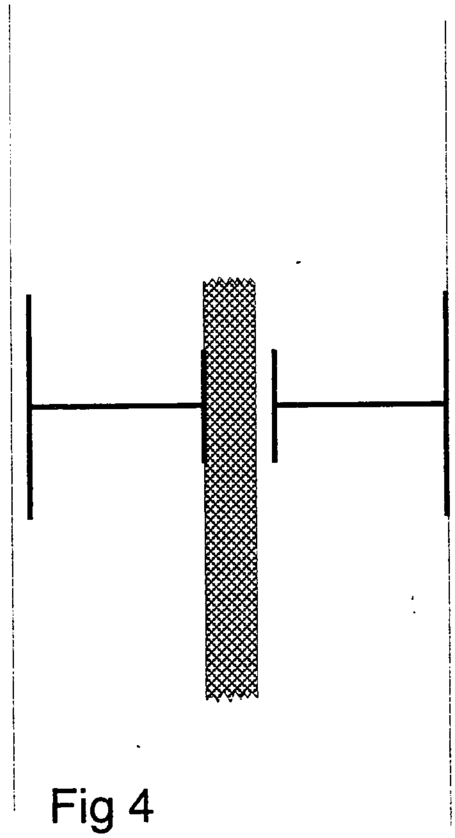

Fig 4

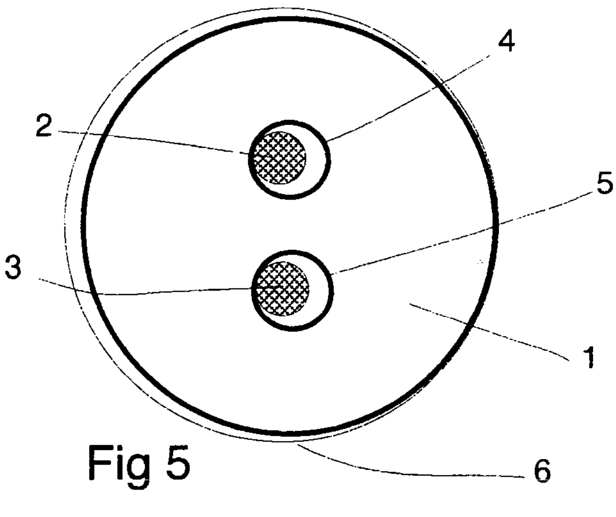

Fig 5

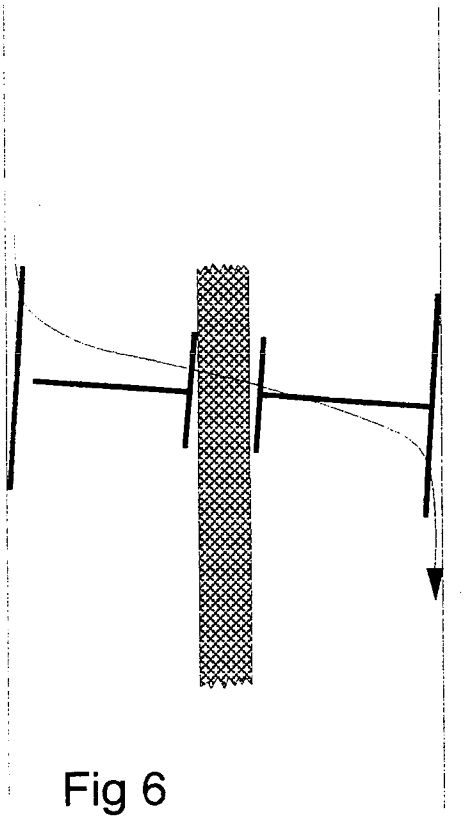

Fig 6

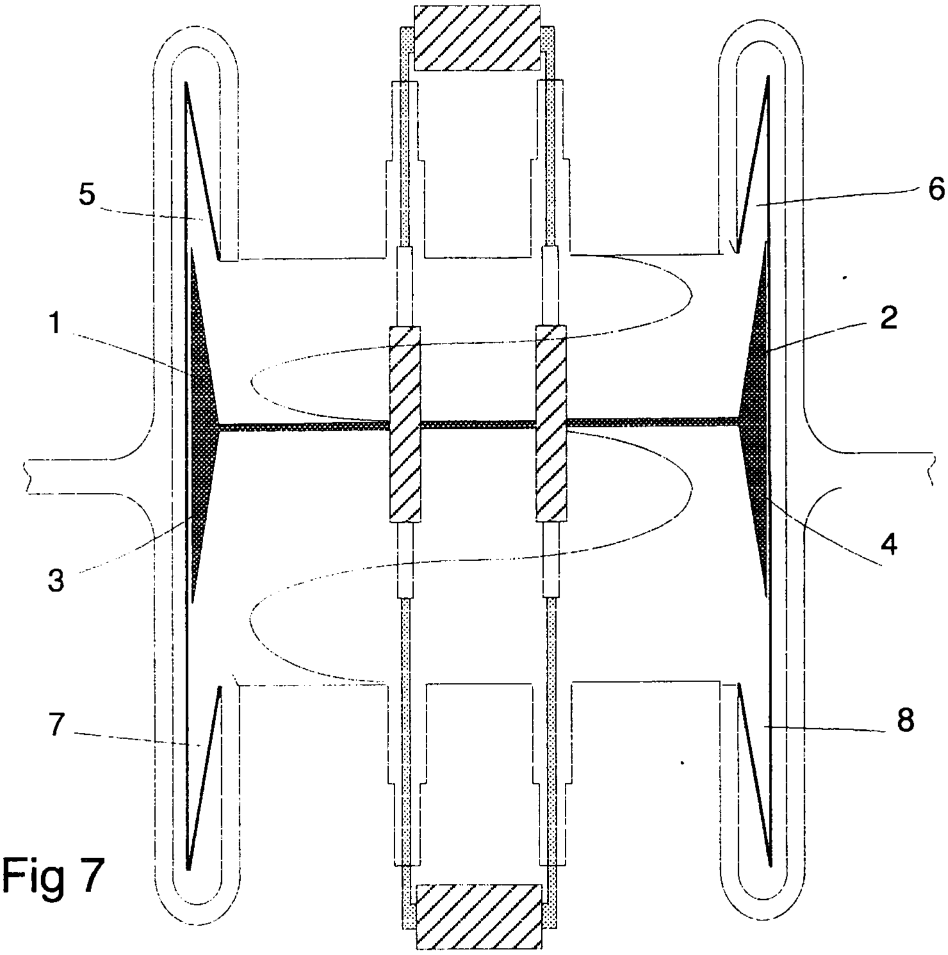

Fig 7

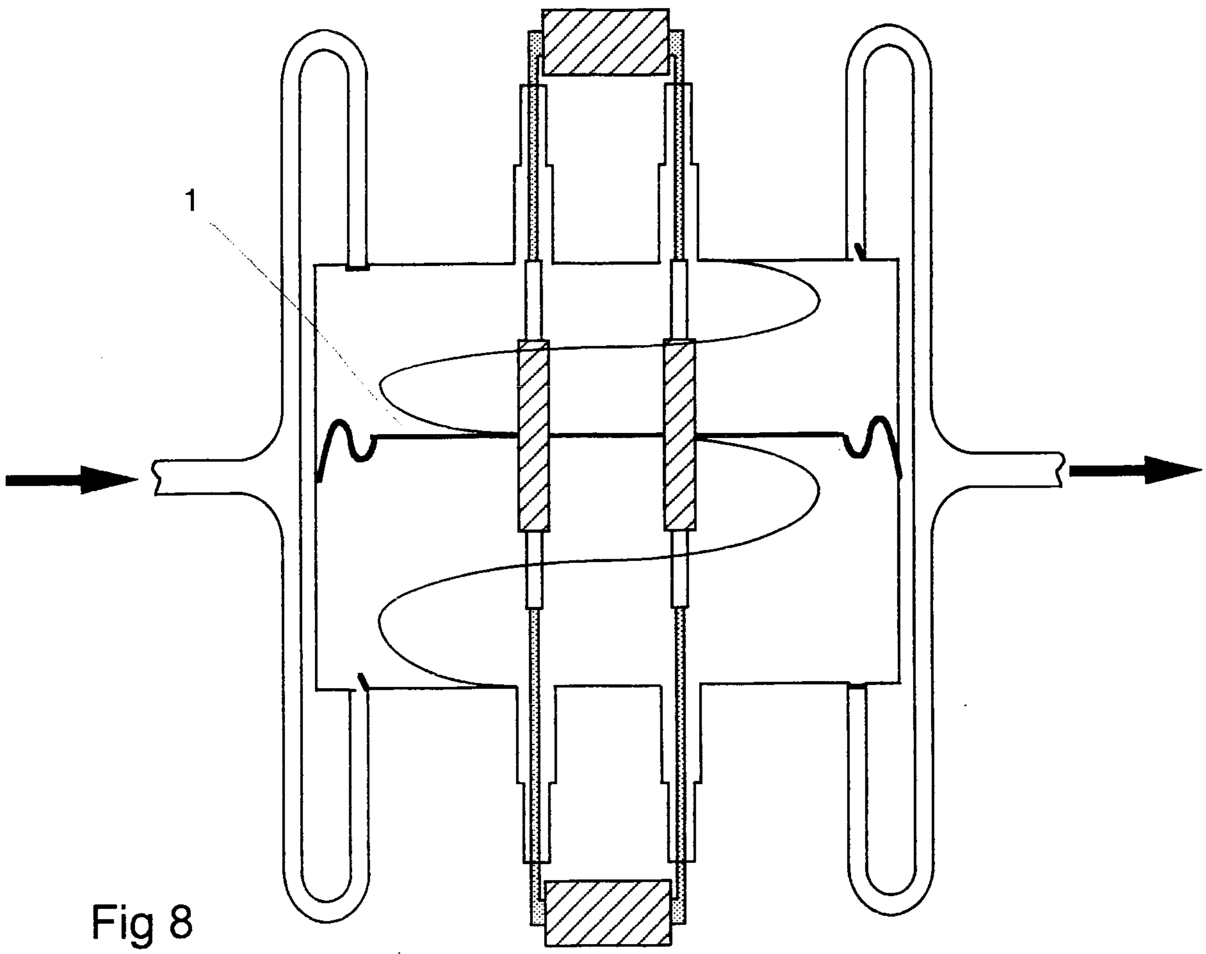

Fig 8

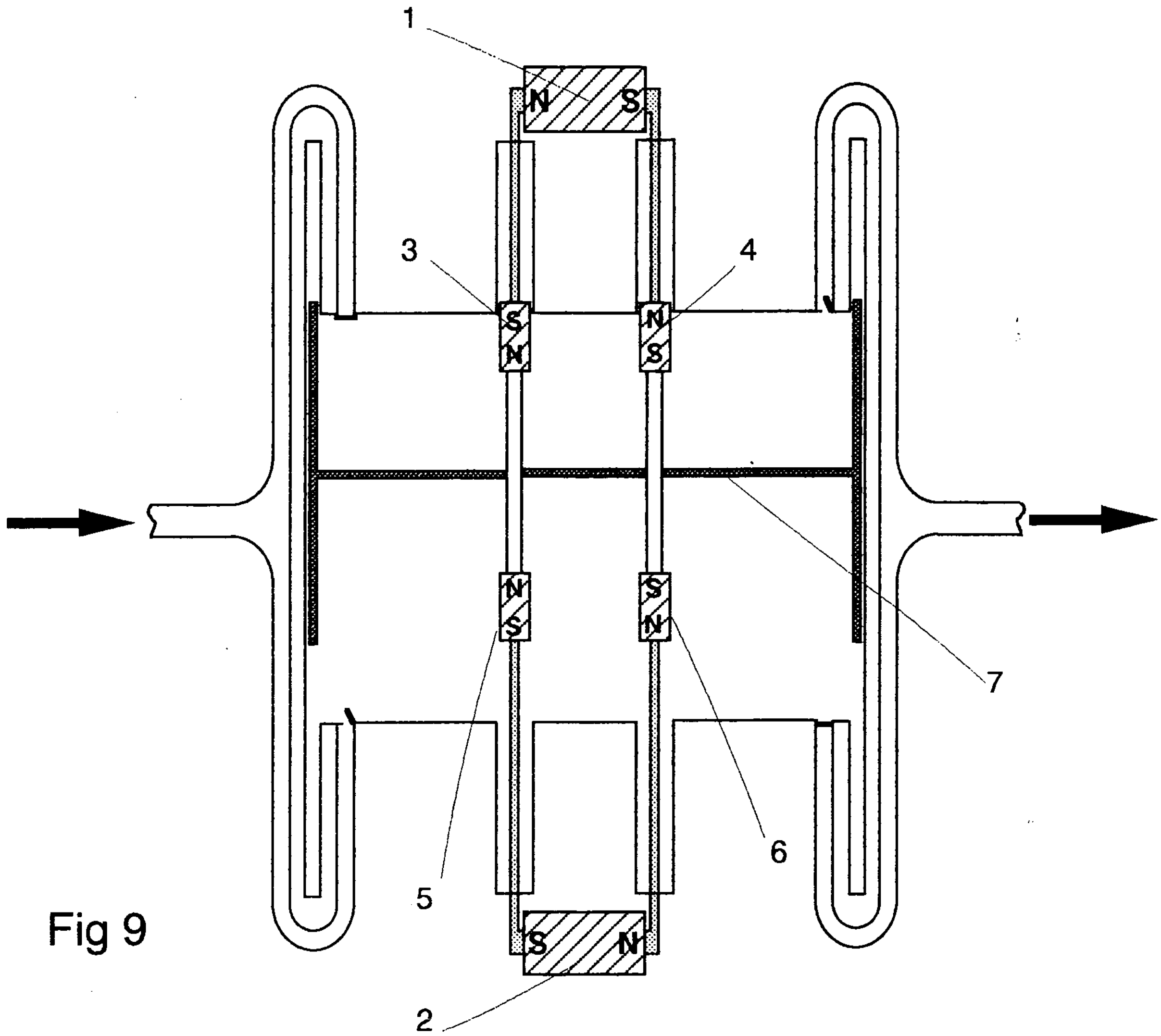

Fig 9

Improved Dual Chamber Displacement Pumps

Technical Field

This invention relates to improvements in dual chamber displacement pumps. The pumps to be described employ the fluid being pumped as the lubricant and do not require additional lubricating fluids.

The current patent application describes improvements in the dual chamber displacement pumps described earlier by the same inventor in patent number GB2273133.

An essential feature of the displacement pumps described in both patent applications is the mounting of a tubular linear motor runner on to the pump piston or diaphragm crown and the addition of a stator shaft which penetrates the crown.

For both inventions self lubrication is achieved by deliberately allowing a small fraction of the fluid being pumped to flow between adjacent pumping chambers via small gaps between the runner and stator and in the case of the piston version of the pump between the piston and chamber walls.

The invention to be described offers reduced fluid leakage between adjacent chambers. A second advantage of the new invention is that the piston or diaphragm crown has an improved self correcting mechanism if it should move out of symmetrical alignment with the stator.

These improvements are made by extending the length of the former on which the runner part of the motor is mounted and/or adding extended skirts at right angles to both faces of the piston crown.

The pumps as claimed in the earlier patent will be described briefly by way of an example after which the new inventive steps will be explained.

The pumps may be used to shift any form of fluid including gases, liquids, slurries etc but to save repetition in the description, the pumped fluid will be referred to as a gas. It is to be understood that in general the explanations in what follows apply to pumping any form of matter that has flow properties and are not necessarily restricted to gases.

Brief description of the drawings

Figure 1 shows a schematic cross section of a piston pump according to the original invention.

Figure 2 shows a schematic cross section of a piston pump according to the new invention.

Figure 3 shows the desired spatial relationship between the piston and the chambers according to the new invention.

Figure 4 shows the same cross section as figure 3 but with the piston displaced to the right, as could possibly occur if unequal forces in the pump shifted the piston sideways.

Figure 5 shows a horizontal cross section in the plane of the crown under the same circumstances as for figure 4.

Figure 6 shows a simplified vertical cross section of the pump in a situation where rotational movement of the piston in a vertical plane has undesirably occurred.

Figure 7 shows a variation in the design of the piston skirts and corresponding cavities which eliminates the dead space problem.

Figure 8 shows a diaphragm version of the pump.

Figure 9 is an illustrative example of a pump according to the present invention which includes permanent magnets as part of the linear motor.

The prior art

Figure 1 shows a cross section, schematic diagram of a piston version of the original invention. This example is motivated by two tubular linear motors.
1 and 2 are dual chambers of the pump which are swept out by the piston 3. 4 and 5 are the runners which consist of solenoids carrying a.c. currents. 6 and 7 are return springs which also act as current

feed lines to 4 and 5. The springs may be chosen so that their resonance frequencies coincides with the preferred frequency of operation of the pump. 8 and 9 are soft ferromagnetic shafts which pass axially through solenoids 4 and 5. 10 and 11 are d.c. current carrying solenoids surrounding soft magnetic cores which are connected to shafts 8 and 9 such that a complete loop of high magnetic permeability material links the cores of the d.c. solenoids and the a.c. solenoids. 12 and 13 are entry valves, 14 and 15 are exit valves.

All of the solenoids are inductively wound. The d.c. solenoids carry currents which flow such that they produce like magnetic poles facing each other. The a.c. solenoids mounted on the piston crown carry currents which are in anti-phase with each other. Thus if the polarities are as shown in figure 1 the piston moves upwards, driving the gas out of the upper chamber. Simultaneously fresh gas is drawn into the lower chamber. When the a.c. current is reversed the piston is driven down such that fresh gas is drawn into the upper chamber and the accumulated gas is driven out of the lower chamber. A small fraction of the gas leaks through the gap between the runners and shafts and between the circumference of the crown and the chamber walls. This leaked gas acts as a lubricant.

The present invention

The present invention is a dual chamber displacement pump comprising a chamber divided into two opposed working chambers, one on each side of an oscillating diaphragm or piston crown, driven by at least one tubular linear motor, the tubular motor comprising a runner and a stator, the runner being carried by the diaphragm or piston and comprising a solenoid mounted on a former, the stator being in the form of a shaft which passes through the centre of the runner and through the diaphragm or piston crown, wherein the former extends beyond the diaphragm or piston crown and/or wherein the piston is provided with skirts which extend a distance beyond the crown of the piston.

The innovative steps claimed for the present invention will now be explained by way of examples. The first example will relate to piston versions of the pump and the second example to diaphragm versions.

Figure 2 shows a schematic cross section of a piston pump.

Parts 1 to 15 on this diagram perform similar functions to parts 1 to 15 in figure 1.
1 and 2 are dual chambers of the pump which are swept out by the piston 3. 4 and 5 are the runners which consist of solenoids carrying a.c. currents. 6 and 7 are return springs which also act as current feed lines to 4 and 5. 8 and 9 are soft ferromagnetic shafts which pass axially through solenoids 4 and 5. 10 and 11 are d.c. current carrying solenoids surrounding soft magnetic cores which are connected to shafts 8 and 9 such that a complete loop of high magnetic permeability material links the cores of the d.c. solenoids and the a.c. solenoids. 12 and 13 are entry valves, 14 and 15 are exit valves.

16 and 17 are extensions to the former on which solenoid 4 is wound. 18 and 19 are extensions to the former on which solenoid 5 is wound. The end faces of the chamber include shaped cavities 20, 21, 22 and 23 into which the crown solenoids and former ends fit at the end points of each stroke.

The piston has a two long cylindrical skirts which extend at right angles to the plane of the crown into the upper and lower chambers. 24 and 25 are parts of the upper skirt and 26 and 27 are parts of the lower skirt as seen in a cross section which cuts through the vertical axis of the piston. A channel is cut into the end walls of each chamber where they meet the chamber side walls. These channels or cavities encompass the ends of the piston skirts at the end of each stroke. 28 and 29 are parts of the upper channel. 30 and 31 are parts of the lower channel.

The beneficial effects of the extended skirts and extended formers will be explained with the aid of figures 3, 4, 5 and 6. Figures 3, 4 and 6 are simplified side elevation cross sections of the pump seen at right angles to figure 2 so that only one shaft is visible. Figure 5 is a horizontal cross section taken through the plane of the piston crown.

Figure 3 shows the desired spatial relationship between the piston and the chambers with the ferromagnetic shaft, piston and chamber walls all sharing a common vertical axis of symmetry in this plane. In this diagram, 1 is a ferromagnetic shaft, 2 is the piston crown, 3 and 4 the former on which one of the solenoids is mounted. (For clarity the solenoid is omitted.) 5 and 6 are parts of the piston skirt. 7 and 8 are parts of the vertical chamber wall. A small amount of the pumped gas passes through the the gaps between 7 and 5, 6 and 8, 3 and 1, 1 and 4. The planned leakage takes place all the way round the circumferences of the shaft and piston skirts but to simplify the discussion when relating to figures 3, 4 and 6 only the leakage through thin fictional laminae in the plane of the diagrams will be considered.

If the piston is currently travelling upwards the pressure difference between the two chambers causes the leaked gas to flow downwards. The rate of leakage is limited by the viscous drag between the gas and the surfaces over which the gas flows. The rate of flow of a gas down a uniform tube is known to be inversely proportional to the length of the tube. Thus the first benefit of including a piston with extended skirts and an extended former is to reduce the rate of gas leakage between chambers.

If for example the vertical length from the rim of the upper skirt to the rim of the lower skirt is ten times the thickness of the crown at its perimeter then the volume rate of leakage is reduced by 90% compared with an un-skirted piston. Leakage reduction benefits, of the same order of magnitude, between the central shaft and piston are obtained by extending the length of the former.

Figure 4 shows the same simplified cross section as figure 3 but with the piston displaced to the right, as could possibly occur if unequal forces in the pump shifted the piston sideways.
Figure 5 shows a horizontal cross section in the plane of the crown under the same circumstances. In figure 5, 1 is the piston crown, 2 and 3 the ferromagnetic shafts, 4 and 5 the formers and 6 the chamber walls.
As the piston moves sideways from the symmetrically disposed position shown in figure 3 to the asymmetrical position shown in figures 4 and 5 the gas in the gaps between the moving parts is forced to flow from the currently decreasing parts of the gap to the corresponding currently increasing parts of the gap. Due to the viscous drag and inertia of the gas this process takes a finite time. During the transfer period the gas pressure in the narrowed part of the gap exceeds that in the ideal gap and the pressure in the widened part of the gap is less than in the ideal gap. Consequentially there is a net sideways force during this period which opposes any change in gap width from the ideal. This negative feedback force is a function of the pressure differences and of the area on which the gases exert their pressure. Thus in the case cited above where the skirts increase the effective length of the piston by a factor of ten then the negative feedback force is ten times that on an un-skirted piston. A similar beneficial effect is obtained by extending the length of the former.

From the point of view of an observer moving with the piston the gas in the gaps is seen to move downwards as the piston moves up. The downwards velocity of the gas is increased by the pressure difference between the two chambers. The resultant velocity is limited by viscous drag. As a section of the gap closes there is a reduction in velocity in that section and a corresponding increase in velocity as a section of the gap increases.

This can be summarised as: narrowed gaps cause a drop in velocity, widening gaps increase velocity.

Those with a knowledge of fluid dynamics will recognise a similarity between this differential velocity gas flow and and the differential velocity air flow between the upper and lower surfaces of an aircraft wing. In the case of the aircraft wing the resultant effect is an uplift force, for the pump the effect is a resultant force which has a negative feed back effect that tends to re-centralise the piston.

The difference in pressures caused by this phenomenon is only modest but for a given rate of gas flow the magnitude of the resultant negative feedback force is amplified by extending the length of the former and piston skirts.

Thus the second advantage of increasing the length of the former and of adding extended piston skirts is that these modifications increase the stability of the system by helping to self centre the piston, if it moves out of alignment, as it traverses the length of the dual chambers.

Figure 6 shows a simplified vertical cross section of the pump in a situation where rotational movement of the piston in a vertical plane has undesirably occurred. In this instance the path of easiest gas flow is shown by the downward pointing arrow which starts at the top left hand side of the piston, moves round to the front of the piston and exits at the bottom right hand side of the piston. A similar path of least resistance exists to the rear of the piston. A consideration of the pressures and forces involved similar to that argued for figures 4 and 5 indicates that the resultant forces caused by the unequal gas flows create a restoring couple which tends to realign the piston parallel to the shafts and chamber walls.

Referring back to figure 2. As the piston moves towards the end of its upward stroke the upper piston skirt and extensions to the formers move into the corresponding cavities in the upper half of the pump chamber. If the cavities allow a snug fit then gas will be trapped in the cavities and compressed. This trapped gas is in a dead space and does not flow out of the pump during the current stroke. This may or may not be a desirable effect depending on the particular demands made on a pump for a specific application. It may be beneficial for example if the elastic properties of the gas are exploited to use the compressed gas in the cavities as a supplement or replacement for return springs which may otherwise be needed to absorb the excess kinetic energy of the piston at the end of the stroke. This is a useful design variable which may be called upon if ferromagnetic return springs produce complications in the shape of the essential magnetic fields required to create the motive force.

Figure 7 shows a variation in the design of the piston skirts and corresponding cavities which eliminates the dead space problem. 1 and 2 are cross sections of parts of the upper skirt. 3 and 4 are cross sections of parts of the lower skirt. 5 and 6 are cross sections of the upper channel or cavity, 7 and 8 cross sections of the lower channel. As shown in this diagram the skirts taper towards their rim with a corresponding tapering of the cavities in which they reside at the end points of the stroke. This arrangement allows gas inside a cavity to flow out as the piston skirt homes into the cavity.

Figure 8 shows a diaphragm version of the pump. In this version the diaphragm, 1 replaces the piston. Otherwise the pump, in this specific example, works in the same manner as the piston version described above. Leakage between chambers is now restricted to gas flow through the gap between the stators and runners. Leakage reduction and diaphragm crown self centring benefits are achieved with this version of the pump by extending the lengths of the formers as described above.

In the examples described above the burden of lubrication has been carried solely by the leakage of gas. This burden has been minimised by the choice of the type of linear motor used as well as by equipping the diaphragm or piston crown with extended piston skirts and/or extended length solenoid formers.

As a precautionary measure, the lubrication mechanisms described above may be backed up by coating one or or both of any of the pairs of sliding surfaces of the pump with a long lasting solid lubricant, for example using a solid lubricant based on p.t.f.e. Experts with a knowledge of modern lubricants will be aware of the choice of suitable solid lubricants available.

It is to be understood that the scope of the current invention is not limited to the type of linear motor described so far but that the invention is extended to include all of the designs of linear motors referred to in patent number GB2273133 and to linear motors in general. It is extended to include linear motors which include different combinations and permutations of d.c. current carrying electromagnets, a.c. current carrying electromagnets and permanent magnets. It is also extended to include linear motors which are commonly described in the literature, "as resembling normal rotating machines that have been cut and opened out flat." It is also extended to include linear motors where the stator through the centre of the piston or diaphragm crown is used solely to centralise the crown.

The scope of the current invention is also extended to include all of the types of dual chamber pumps referred to in patent number GB2273133.

Figure 9 is an illustrative example of a pump according to the present invention which includes permanent magnets as part of the linear motor. 1 and 2 in figure 9 are electromagnets carrying alternating currents. 3, 4, 5 and 6 are hollow cylindrical permanent magnets mounted on the formers. The fixed polarities of the permanent magnets and current polarities of the electromagnets as shown on figure 9 drive the piston 7 upwards. Half a cycle later when the polarities of the electromagnets change the piston is driven down. Engineers with a knowledge of magnetic alloys will be aware of the choice of magnetic materials available for use as the permanent magnets for this pump.

In this example the trapped gas in the dead space in the cavities at the ends of the chamber acts as an elastic energy absorber and return springs are unnecessary.

The scope of the current invention is extended to include at least:

i) All dual chamber displacement pumps which offer two contributing volumes, one on each side of the piston crown and include one or more motors with the movable part or runner of each motor being mounted on the crown of the piston or diaphragm and the stator taking the form of a shaft which passes through the runner at right angles to the crown.

ii) Versions of the pump in which springs are used to control the cut-off pressure.

iii) Dual chamber pumps which are internally coated or painted in order to resist corrosive attack from the fluids being pumped.

iv) Dual chamber pumps which are internally coated or painted in order to reduce resistance to flow of the fluid being pumped.

v) Dual chamber pumps which include electromagnets that can be tuned by the end user to optimize pumping performance.

vi) Dual chamber pumps which include heating elements to reduce the viscosity of fluids being pumped.

vii) Cascaded versions of the pump in which two or more of the dual chamber pumps as outlined in the previous paragraph are connected in series. As explained in GB2273133 typical pumps of this type would have successive pairs of chambers having decreasing swept out volumes if the pumped fluid takes the form of a compressible gas or vapour.

viii) Peristaltic versions of the diaphragm dual chamber pump.

ix) Vacuum pumping versions of the pump which include side valves to vent off gas into the local environment to prevent excessive build up of pressure inside cascaded versions of the vacuum pump in the early stages of evacuation.

x) Vacuum pumping versions which include one or more sets of electromagnetically operated inlet and/or outlet valves to allow the pump to operate at very low pressures.

xi) Fluid mixing versions of the pump which have two entry ports, allowing two different fluids to be input and include flow control mechanisms to allow the two fluids to be mixed in different controllable proportions.

xii) Versions of the pump which can have their cavity sizes varied to allow them to operate at their resonance volume for a given frequency of pump strokes.

xiii) Versions of the pump in which the valves can be remotely manipulated to allow the direction of pumping to be reversed so that what in the first instance was the inlet port of the pump now becomes the outlet port.

xiv) Versions of the pump which include mechanisms which detect the rate of flow of the fluid and/or exit fluid pressure and increase the rate of pumping if necessary to allow for aging of the pump.

xv) Versions of the pump in which some of the coils at least, which produce the essential magnetic fields required for pump operation are immersed in a cooling fluid which cool the coils to a sufficiently low temperature that the coils operate as superconductors.

xvi) Stirling cycle refrigerators which include pumps of the type claimed in this application.

CLAIMS

1 A dual chamber displacement pump comprising a chamber divided into two opposed working chambers, one on each side of an oscillating diaphragm or piston crown, driven by at least one tubular linear motor, the tubular motor comprising a runner and a stator, the runner being carried by the diaphragm or piston and comprising a solenoid mounted on a former, the stator being in the form of a shaft which passes through the centre of the runner and through the diaphragm or piston crown, wherein the former extends beyond the diaphragm or piston crown and/or wherein the piston is provided with skirts which extend a distance beyond the crown of the piston.

2 A pump according to claim 1 with the ends of the chambers having recesses or cavities which accommodate the extensions to the former(s) and/or piston skirts at the end of each stroke.

3 A pump according to claim 2 but with the former on which each crown solenoid is mounted having tapered walls of a diminishing thickness towards its ends and the associated recesses in the chamber end walls also tapering in a complimentary manner inside the recesses so as to allow the former to be driven fully into a complimentary recess at the end of the piston or diaphragm stroke without trapping a significant amount of the pumped fluid inside the recess.

4 A pump according to claims 2 or 3 but with the piston skirts having walls of a diminishing thickness towards their free ends and the associated recesses in the chamber end walls also tapering in a complimentary manner inside the recesses so as to allow a skirt to be driven fully into a complimentary recess at the end of the piston stroke without trapping a significant amount of the pumped fluid inside the recess.

5 A pump according to any of the above claims with the stator shaft or shafts including or being made from ferromagnetic material.

6 A pump according to any of the above claims in which the shaft is a ferromagnetic material with magnetic flux being generated in the shaft by one or more permanent or electromagnets coupled to the shaft at fixed points.

7 A pump according to claim 6 with the magnetic flux created in the stator being alternating in nature.

8 A pump according to any of the above claims with runners including one or more permanent magnets.

9 A pump according to any of the above claimswith runners including electromagnets carrying direct currents.

10 A pump according to any of claims 1-8 with the runners including electromagnets carrying alternating currents.

11 A pump according to any of the above claims which is lubricated, in part at least, by a small fraction of the pumped fluid acting as a lubricant between the runner and stator and between the piston skirt and the adjacent chamber walls.

12 A pump according to any of the above claims which uses elastic components such as return springs to control the cut-off pressure.

13 A pump according to any of the above claims and having inlet and outlet valves having roles which can be remotely changed so as to reverse the direction of fluid flow.

14 A pump according to any of the above claims in which a heating element is included or which utilises existing circuit components to produce a heating affect with the intention of heating the fluid being pumped.

15 A pump according to any of the above claims which has some or all of the surfaces in sliding contact coated with solid lubricant.

16 A pump according to any of the above claims which has some or all of the internal components painted or otherwise surface treated in order to prevent corrosion and/ or aid the flow of fluid.

17 A pump according to any of the above claims which has a plurality of linear motors driving the piston or diaphragm.

18 A pump according to any of the above claims which is similar to a peristaltic pump in that the dual chambers are separated by a flexible diaphragm motivated by two or more tubular linear motors which operate out of phase such that during each cycle the shape of a chamber becomes constricted at the inlet valve end and the constriction then becomes more general, travelling towards the exit valve end of the pump, this state being followed by an equivalent phased expansion.

19 A pump according to any of the above claims which has electromagnetically operated valves which are synchronised to open and close at the necessary stages of the pumping cycle in order to allow the pump to operate with greater efficiency at low pumped fluid pressures.

20 A pump according to any of the above claims which includes side valves which allow fluid to escape directly to the local environment when a pre-set pressure inside the chamber is exceeded.

21 A pump according to any of the above claims which can be tuned or adjusted by means of altering the positions of the electromagnets.

22 A pump according to any of the above claims which can be tuned or adjusted by means of altering the electromagnet currents.

23 A pump according to any of the above claims which include sensors for detecting magnetic field strength or rate of change of field strength.

24 A pump according to any of the above claims which include sensors for detecting magnetic field strength or rate of change of field strength and feed back mechanisms which allow the pump to correct itself if unequal forces or aging cause inefficient operation or malfunction of the pump.

25 A pump according to any of the above claims which includes chambers with movable walls that can be tuned mechanically or automatically in response to feedback signals in order to increase the efficiency of the pump.

26 A pump according to any of the above claims in which the containing walls of the pump are shaped to match with the shape of the piston or diaphragm in the maximum displaced position in order to minimise the volume of dead space inside each chamber.

27 A pump according to any of the above claims in which two different fluids have access to the respective chambers through two adjacent inlet ports.

28 A pump according to claim 27 in which two different fluids have access to the respective chambers and which includes a mechanism for altering the relative pumping pressures of the two chambers in order to allow the fluids to be mixed in different proportions after emergence from the pump.

29 A pump according to any of the above claims which is used to displace the working gas in a reverse Stirling refrigerator.

30 A pump according to any of the above claims which includes some of the electrically conducting components at least cooled to a sufficiently low temperature that the conducting material acts as a superconductor.

31 A pump assembly comprising at least two pumps according to any of the above claims connected in series with successive dual pump chambers having progressively smaller volumes.

32 A pump substantially as described with reference to figures 2 to 9 of the accompanying drawings.