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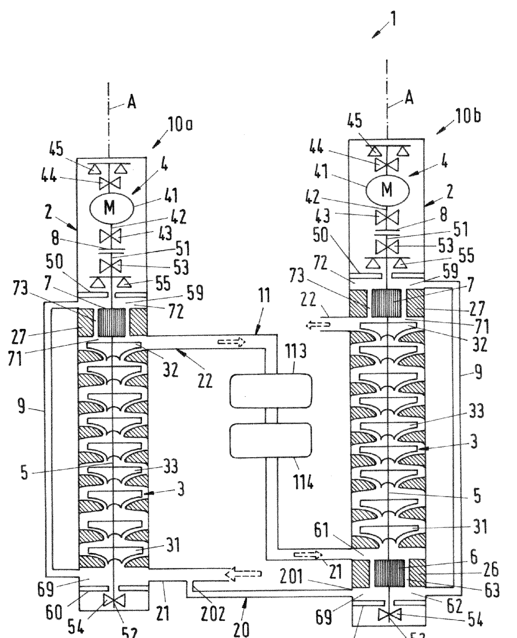
# (54) PUMPING ARRANGEMENT

(57) A pumping arrangement is proposed, comprising a first multistage pump (10a) and a second multistage pump (10b), wherein each multistage pump (10a, 10b) comprises:
a housing (2) with a pump unit (3) arranged in the housing (2), wherein the housing (2) comprises a pump inlet (21) for receiving a fluid with a suction pressure, and a pump outlet (22) for discharging the fluid with a discharge pressure, and wherein the pump unit (3) comprises a plurality of impellers (31, 32, 33) for conveying a fluid from the pump inlet (21) to the pump outlet (22), and
a pump shaft (5) for rotating about an axial direction (A), with the pump shaft (5) extending from a drive end (51) to a non-drive end (52), wherein each impeller (31, 32, 33) is mounted to the pump shaft (5) in a torque proof manner.

The second multistage pump (10b) comprises at least a first mechanical seal (50) for sealing the pump unit (3) at the pump shaft (5), with the first mechanical seal (50) having a process side (59) facing the pump unit (3). The pump outlet (22) of the first multistage pump (10a) is connected to the pump inlet (21) of the second multistage pump (10b), so that the first multistage pump (10a) and the second multistage pump (10b) are arranged in series. The process side (59) of the first mechanical seal of the second multistage pump (10b) is in fluid communication with the pump inlet (21) of the first multistage pump (10a).

Fig.3

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## Description

**[0001]** The invention relates a pumping arrangement comprising a first multistage pump and a second multistage pump in accordance with the preamble of the independent claim.

**[0002]** Multistage pumps for conveying a fluid are used in many different industries, in particular for applications where a high pressure shall be generated. A multistage pump comprises a plurality of impellers, which are arranged on a common shaft. The common shaft is driven for a rotation about an axial direction so that all impellers are commonly rotated about the axial direction. One important industry, in which multistage pumps are used, is the oil and gas processing industry, where multistage pumps are designed e.g. for conveying hydrocarbon fluids, for example for extracting the crude oil from the oil field or for transportation of the oil/gas through pipelines or within refineries. Another application of multistage pumps in the oil and gas industry is the injection of a fluid, for example water and in particular seawater, into an oil reservoir. For such applications, said pumps are designed as (water) injection pumps supplying seawater at high pressure to a well that leads to a subterranean region of an oil reservoir. A typical value for the pressure increase generated by such an injection pump is 200-300 bar (20 - 30 MPa) or even more.

**[0003]** In view of an efficient exploitation of oil and gas fields, there is nowadays an increasing demand for pumping equipment that may be installed directly on the sea ground in particular down to a depth of 500 m, down to 1000 m or even down to more than 2000 m beneath the water surface. Needless to say that the design of such pumping devices is challenging, in particular because these devices shall operate in a difficult subsea environment for a long time period with as little as possible maintenance and service work. This requires specific measures to minimize the amount of equipment involved and to optimize the reliability of the pumping devices.

**[0004]** In particular in deepwater oil fields there are massive amounts of carbon dioxide ($CO_2$) and natural gas on top of the crude oil. The carbon dioxide and the natural gas, which contains methane ($CH_4$), are usually separated from the oil. This is usually done at the water surface (topside) on an FPSO (floating production storage and offloading) unit or onshore. The separated gas can be compressed and reinjected into the reservoir in order to maintain the reservoir pressure or the gas is injected into exhausted gas reservoirs to be stored in the ground. The reinjection into oil reservoirs is a well-known method for increasing the recovery of hydrocarbons from an oil or gas field. The injected fluid maintains or increases the pressure in the reservoir thereby driving the oil or the hydrocarbons towards and out of the production well. This process is known as enhanced oil recovery (EOR).

**[0005]** The separation, treatment and reinjection of carbon dioxide/natural gas at the topside i.e. on a FPSO unit or in an onshore facility requires a significant amount of space. This amount may be for example 70% of the required topside space. One of the main reasons is the low density of the gas at the topside operation pressures. Therefore the idea came up to separate the carbon dioxide/natural gas/methane from the oil at a subsea location e.g. on the sea ground. Thus, the crude oil containing the light components such as carbon dioxide, methane, ethane is separated at the sea ground into a heavier liquid enriched phase, which is delivered to a topside location, and into a lighter $CO_2$ and $CH_4$ enriched phase, which is reinjected into a subterranean region, e.g. the oil reservoir. Due to the hydrostatic pressure at the sea ground the separation will take place for many applications at a pressure and temperature where carbon dioxide is in the supercritical state or in the dense phase.

**[0006]** The dense phase or supercritical phase for a pure fluid is the region beyond the critical point, namely the fluid region where the pressure is higher as the critical pressure and the temperature is higher as the critical temperature.

**[0007]** Basically, both terms "supercritical phase" and "dense phase" designate the same state of matter. Thus, from a physical point of view both terms are synonyms and they will be use as synonyms within the scope of this application. Beside the solid, the liquid and the gaseous state the dense phase or the dense fluid phase is another state of matter, which is characterized by a viscosity similar to that of a fluid in the gaseous phase, but a density closer to that of a fluid in the liquid phase.

**[0008]** Although "supercritical phase (state)" and "dense phase (state)" are synonyms from the physical perspective, they are used - as a kind of convention - with slightly different meaning: the tendency is to use the term "supercritical phase" for a single component fluid (also referred to as pure fluid), and the term "dense phase" fluid for a multi-component fluid.

**[0009]** For a better understanding, Fig. 1 shows as an example schematically the phase diagram of pure $CO_2$ (single component fluid). The horizontal axis T indicates the temperature in degree Celsius, and the vertical axis P indicates the pressure in MPa. The Line SL indicates the phase boundary between the solid phase and the liquid phase. The line SG indicates the phase boundary between the solid phase and the gaseous phase. The line LG indicates the phase boundary between the liquid and the gaseous phase. The point TP is the triple point and the point CP is the critical point. The region SC indicates the area where the fluid is in the supercritical state or supercritical phase.

**[0010]** The process fluid in typical $CO_2$ pumping applications is often a mixture of several components. A typical example of such a mixture is a mixture of $CO_2$ and natural gas.

**[0011]** Natural gas itself is already a mixture of several components such as methane, ethane, propane and so on. Fluid mixtures such as natural gas have a phase envelope in which the liquid phase and the gaseous or vapor phase are in equilibrium with each other over a range of temperature, pressure and composition. In this region the two phases coexist. This equilibrium region is a two-phase region having a liquid and a gaseous phase. The gaseous phase can also be called the vapor phase or the gas phase.

**[0012]** Fig. 2 shows in a schematic representation a typical phase diagram of a multi-component fluid at one defined composition. The horizontal axis T again indicates the temperature, wherein the temperature is increasing to the right. The vertical axis P again indicates the pressure wherein the pressure is increasing upwardly. The region EQ indicates the region where the liquid phase and the gaseous phase coexist. The line DP indicates the dew point curve, which is the boundary between the region EQ and the gaseous phase. The line BP indicates the bubble point curve, which is the boundary between the region EQ and the liquid phase. The curves FL between the line BP and the line DP indicate different molar fractions of the liquid in the equilibrium region EQ.

**[0013]** The dense phase region SC for such a multi-component fluid is beyond the critical point CP and the phase envelope built by the dew point curve DP and the bubble point curve BP, namely above the critical pressure and the critical temperature and outside said phase envelope.

**[0014]** It is only inside the envelope delimited by the bubble point curve BP and dew point curve DP that there is a two phase equilibrium of the liquid and gaseous phase.

**[0015]** Outside of this envelope, the fluid is in single phase condition, namely single phase liquid phase, single phase gaseous phase or single phase dense phase.

**[0016]** An example of a pumping application of a single phase, dense phase $CO_2$ rich multi-component fluid can be found in the upstream offshore oil and gas industry.

**[0017]** As already mentioned in a subsea oil and gas field exploitation the lighter $CO_2$ enriched phase contains a considerable amount of other components, predominantly $CH_4$. This lighter fluid phase as a whole is a mixture of different components and is quite often in the dense phase SC at a temperature and pressure which is above the critical point CP of the multi-component fluid. A typical operation pressure for the separation into the lighter phase and the heavier phase may be for example around 200 bar (20 MPa) where the mixture of carbon dioxide with natural gas may have a density of which is higher than 200 kg/m$^3$, e.g. approximately 400 kg/m$^3$. This means, that the lighter phase has a density at the sea ground, which is a few hundred times larger than the density of air at normal conditions. In addition, the lighter $CO_2$ and $CH_4$ enriched fluid being in the dense phase has a viscosity which is comparable to the viscosity of a gas, a density which is comparable to the density of a liquid and a compressibility, which is comparable to the compressibility of a gas.

**[0018]** In EP 3 771 828 A1 a multistage pump and a pumping arrangement are disclosed, that can reinject such a compressible fluid in a subterranean region. For many applications a significant pressure rise is required for the reinjection of the compressible fluid. Therefore, the pumping arrangement comprising two multistage pumps arranged in series is preferred for such applications rather than a single multistage pump.

**[0019]** Operating two pumps in series has proven to be a good concept, however there are several challenges that have to be addressed. As it is disclosed in EP 3 771 828 A1 in such a pumping arrangement the pump inlet of the second multistage pump is connected to the pump outlet of the first multistage pump, such that the discharge pressure of the first multistage pump is (at least approximately) the suction pressure of the second multistage pump.

**[0020]** Usually, the pump unit of each multistage pump is sealed at the pump shaft by two mechanical seals, namely adjacent to the drive end of the pump shaft and adjacent to the non-drive end of the pump shaft. The first mechanical seal at the drive end is for example arranged between a balance drum and a bearing unit for the pump shaft. Since the back side of the balance drum is connected to the suction side of the second multistage pump by means of a balance line, the process side of the first mechanical seal, which is the side facing the back side of the balance drum is exposed to a pressure which is - disregarding a minor pressure drop over the balance line - the suction pressure of the second multistage pump.

**[0021]** The second mechanical seal at the non-drive end is for example arranged between a bearing unit at the non-drive end of the pump shaft and the first stage impeller of the pump. Thus, the process side of the second mechanical seal, which is the side facing the first stage impeller of the pump unit, is exposed to the suction pressure of the second multistage pump.

**[0022]** Usually, the balance line constitutes a flow connection between the process side of the first mechanical seal and the process side of the second mechanical seal, so that the two process sides of the mechanical seals are in fluid communication with each other. Therefore, the pressure prevailing at the process side of the first mechanical seal is essentially the same as the pressure prevailing at the process side of the second mechanical seal. Said pressure is the suction pressure of the second multiphase pump.

**[0023]** A problem arises, when the first multistage pump trips, e.g. because of a malfunction or an overload. The discharge pressure of the first multistage pump immediately drops down within a very short period of time, for example: 80% of the generated pressure rise, i.e. the difference between the discharge pressure and the suction pressure, falls away within the first two seconds. It takes approximately ten seconds to come to a full stop of the pump. The strong decrease in the discharge pressure of the first pump results in a massive drop of the pressure at the process side of the first and the second mechanical seal of the second multistage pump. The pressure drop may be for example 250 bar (25 MPa).

The barrier fluid system for the mechanical seals of the second multistage pump will not be able to depressurize the barrier fluid at the non-process side of the mechanical seal, e.g. in the drive unit, quickly enough, so that a tremendous pressure difference across the mechanical seals will result. Said pressure drop is detrimental for the mechanical seals and can even cause under dynamic conditions a break of the mechanical seal. The pressure drop may even be larger than the maximum allowed pressure difference over the mechanical seals during operation. Thus, the second multistage pump has to be shut down as quickly as possible. However, since there will always be a delay to shut down the second multistage pump, there is a considerable risk that in particular the mechanical seals of the second multistage pump will be damaged. Even if the second pump is shut down at the same time as the first pump trips, the second pump will be exposed during the ten seconds it takes to come to a stop a damaging overpressure for the mechanical seals of the second pump.

**[0024]** Furthermore, due to the huge pressure difference over the mechanical seals of the second multistage pump a considerable amount of barrier fluid is forced to flow through the mechanical seals to the process side of the respective mechanical seal. The increased consumption of barrier fluid is a disadvantage in particular from an economic perspective.

**[0025]** Another problem arises from pressure spikes at the process side of the mechanical seals of the second multistage pump. Such pressure spikes can be caused by a slug, such as a liquid slug, going through the first pump. A possible reason for such a liquid slug can be, for example, an instabilities in the separator that separates the multiphase fluid into a heavier liquid enriched phase, which is delivered to a topside location, and into a lighter $CO_2$ and $CH_4$ enriched phase, which is supplied to the first multistage pump of the pumping arrangement. A carry-over of liquid in the lighter phase can occur in the separator. This carry-over of liquid can result in a liquid slug, e.g. in a water slug, propagating into the first multistage pump.

**[0026]** The dense gas has of the lighter phase leaving the separator has for example a density of around 400kg/m$^3$. The liquid slugs such as water slugs can have densities of 1000kg/m$^3$.

**[0027]** In case there is a liquid slug passing through the first multistage pump and the drive unit driving the rotation of the pump is operated with a frequency control, i.e. operated to maintain a constant rotational speed, the first multistage pump will generate a considerable pressure spike. When the multistage pump is operated with a constant rotational speed, the discharge pressure is proportional to the density of the fluid. Therefore, the liquid slug will result in a considerable increase in the discharge pressure of the first multistage pump. This results in a pressure spike at the process sides of the mechanical seals of the second multistage pump. The barrier fluid pressure system of said mechanical seals is usually not fast enough to follow the pressure spike and therefore the pressure spike result in a reverse pressure across the mechanical seals, meaning that the pressure prevailing at the process side is considerably larger than the pressure prevailing at the other side (non-process side) of the mechanical seal. Such large pressure drops across the mechanical seal are detrimental for the mechanical seal and can cause a considerably enhanced wear or even a malfunction or a damage of the mechanical seal.

**[0028]** In particular for conveying dense gas at a subsea location, it were desirable to have a multistage pump configured as a seal-less dense gas injection pump, i.e. a multistage pump without mechanical seals. A seal-less multistage pump configured for installation on a sea ground is for example disclosed in EP 3 808 984 A1. This seal-less configuration allows to dispense with the mechanical seals. This would resolve some of the previously described problems and considerably increase the mean time between maintenance (MTBM). However, nowadays a seal-less configuration for conveying dense gas is not yet available so that the protection of the mechanical seals is still an issue, in particular for dense gas applications. Starting from this state of the art, it is therefore an object of the invention to propose a pumping arrangement comprising two multistage pumps arranged in series, which is better protected against the negative impact of pressure spikes as they are for example generated by liquid slugs. The pumping arrangement shall also be suited for subsea applications and for deployment on the sea ground. In particular the pumping arrangement shall be suited to be configured as an injection pump for injecting a compressible fluid being in the dense phase in a subterranean region. Furthermore, it is an object of the invention to propose a subsea pumping arrangement comprising such a multistage pump.

**[0029]** The subject matter of the invention satisfying this object is characterized by the features of the independent claim.

**[0030]** Thus, according to the invention, a pumping arrangement is proposed comprising a first multistage pump and a second multistage pump, wherein each multistage pump comprises: a housing with a pump unit arranged in the housing, wherein the housing comprises a pump inlet for receiving a fluid with a suction pressure, and a pump outlet for discharging the fluid with a discharge pressure, and wherein the pump unit comprises a plurality of impellers for conveying a fluid from the pump inlet to the pump outlet, and

a pump shaft for rotating about an axial direction, with the pump shaft extending from a drive end to a non-drive end, wherein each impeller is mounted to the pump shaft in a torque proof manner.

**[0031]** The second multistage pump comprises at least a first mechanical seal for sealing the pump unit at the pump shaft, with the first mechanical seal having a process side facing the pump unit.

**[0032]** The pump outlet of the first multistage pump is connected to the pump inlet of the second multistage pump, so that the first multistage pump and the second multistage pump are arranged in series. The process side of the first mechanical seal of the second multiphase pump is in fluid communication with the pump inlet of the first multistage pump.

**[0033]** By providing a fluid communication between the process side of the first mechanical seal of the second multistage pump and the pump inlet of the first multistage pump, the process side of the first mechanical seal is decoupled from the discharge pressure of the first multistage pump, and therewith from the suction pressure of the second multistage pump. Thus, if a pressure slug occurs causing a pressure spike in the discharge pressure of the first multistage pump, there is no negative impact on the first mechanical seal of the second multistage pump, because the process side of said first mechanical seal is in fluid communication with the pump inlet of the first multistage pump, so that the pressure prevailing at the process said of said first mechanical seal is always at least approximately the same as the suction pressure of the first multistage pump.

**[0034]** As a consequence the pressure difference across the first mechanical seal of the second multistage pump remains constant and can be adjusted to a very low value. Thus, during operation there is only a small pressure drop over the first mechanical seal and in addition the pressure on the process side of the first mechanical seal is essentially constant. This results in a low wear of the first mechanical seal, in a reliable function of the first mechanical seal as well as in a very low consumption of barrier fluid for the first mechanical seal.

**[0035]** Furthermore, the barrier fluid system for the first mechanical seal of the second multistage pump is not influenced by the discharge pressure of the first multistage pump. This enables a very economical operation of the barrier fluid system.

**[0036]** Preferably, each multistage pump comprises a first balance drum, which is fixedly connected to the pump shaft between the pump unit and the drive end of the pump shaft, the first balance drum defining a first front side facing the pump unit and a first back side, wherein a first relief passage is provided between the first balance drum and a first stationary part configured to be stationary with respect to the housing, the first relief passage extending from the first front side to the first back side, and wherein a balance line is provided and configured for the recirculation of the fluid from the back side to a low pressure side of the multistage pump. By providing in each multistage pump the first balance drum at the pump shaft, the axial thrust generated by the impellers during operation of the respective multistage pump is at least partially compensated by the pressure drop over the first balance drum. This measure considerably reduces the load that has to be carried by the axial or thrust bearing(s) of the respective multistage pump.

**[0037]** According to a preferred embodiment, the second multistage pump comprises a second balance drum, which is fixedly connected to the pump shaft between the pump unit and the non-drive end of the pump shaft, the second balance drum defining a second front side facing the pump unit and a second back side, wherein a second relief passage is provided between the second balance drum and a second stationary part configured to be stationary with respect to the housing, the second relief passage extending from the second front side to the second back side, and wherein the second back side is in fluid communication with the pump inlet of the first multistage pump. The second balance drum constitutes a simple measure to decouple the suction pressure of the second multistage pump, and therewith the discharge pressure of the first multistage pump, from the pressure prevailing at the process side of the first mechanical seal of the second multi stage pump, which equals the suction pressure of the first multistage pump.

**[0038]** Preferably, the balance line of the second multistage pump is configured to connect the first back side of the second multistage pump with the second back side of the second multistage pump. By this measure essentially the same pressure prevails at the second back side of the second balance drum as at the first back side of the first balance drum of the second multistage pump.

**[0039]** Furthermore, it is preferred that the second front side is exposed to the suction pressure of the second multistage pump. Thus, the second balance drum is exposed to the suction pressure of the second multistage pump on the second front side and to the suction pressure of the first multistage pump on the second back side.

**[0040]** According to a preferred configuration the second multistage pump comprises a second mechanical seal for sealing the pump unit at the pump shaft, with the second mechanical seal having a process side facing the pump unit, wherein the second mechanical seal is arranged between the second balance drum and the non-drive end of the pump shaft of the second mechanical seal. According to this configuration the pump unit is sealed at the pump shaft with two mechanical seals, namely by means of the first mechanical seal at the drive end of the pump shaft, and by means of the second mechanical seal at the non-drive end of the pump shaft.

**[0041]** In other preferred embodiments the second multistage pump has no second mechanical seal at the non-drive end of the pump shaft, but only the second balance drum arranged at the non-drive end of the pump shaft.

**[0042]** Furthermore, it is a preferred design, that the plurality of impellers comprises a first set of impellers and a second set of impellers, wherein the first set of impellers and the second set of impellers are arranged in a back-to-back arrangement, so that an axial thrust generated by the first set of impellers is directed opposite to an axial thrust generated by the second set of impellers.

**[0043]** Regarding the back-to-back design it is advantageous that each multistage pump comprises a center bush, which is fixedly connected to the pump shaft between the first set of impellers and the second set of impellers, wherein a balancing passage is provided between the center bush and a third stationary part configured to be stationary with respect to the housing. The center bush with the balancing passage also contributes to reduce the overall axial thrust acting upon the pump shaft in the respective multistage pump.

**[0044]** In addition, the center bush and/or the balance drum(s) (throttle bush) support the rotordynamic stability both with respect to stiffness and damping in particular of rotor vibrations. The rotor is the entity of the rotating parts of the pump unit, i.e. in particular all impellers as well the pump shaft are part of the rotor of the pump unit.

**[0045]** In particular for subsea applications where the pump arrangement is deployed below the water surface, it is preferred that each multistage pump comprises a drive unit arranged in the housing, wherein the drive unit comprises a drive shaft for driving the pump shaft, and an electric motor for rotating the drive shaft about the axial direction, and wherein a coupling is provided for coupling the drive shaft to the pump shaft. The housing with the drive unit inside may then be configured as a pressure housing, which is able to withstand the large hydrostatic pressure at a subsea location, e.g. on the sea ground.

**[0046]** Preferably, in each multistage pump the first mechanical seal is arranged between the first balance drum and the drive unit. Thus, it is preferred that the first mechanical seal is arranged for sealing the pump unit at the drive end of the shaft and thus preventing the fluid from entering the drive unit.

**[0047]** According to a preferred embodiment, each multistage pump is configured as a vertical pump with the pump shaft extending in the direction of gravity, wherein the drive unit is arranged on top of the pump unit.

**[0048]** In a preferred embodiment each multistage pump comprises a throttle bush, which is fixedly connected to the pump shaft between the pump unit and the drive end of the pump shaft, the throttle bush defining a throttle front side facing the pump unit and a throttle back side, wherein a throttle passage is provided between the throttle bush and a stationary throttle part configured to be stationary with respect to the housing, the throttle passage extending from the throttle front side to the throttle back side, wherein a first supply opening is provided at the stationary throttle part to supply fluid to the throttle passage at a location between the throttle front side and the throttle back side with respect to the axial direction, wherein the first supply opening is connectable with an intermediate take-off, which is arranged for delivering the fluid with a pressure higher than the suction pressure of the particular multistage pump, and wherein a second balance line is provided and configured for the recirculation of the fluid from the throttle back side to a low pressure side.

**[0049]** Preferably, the throttle bush is arranged with respect to the axial direction between the first balance drum and the first mechanical seal, so that the throttle front side coincides with the first back side. By connecting the first supply opening with the intermediate take-off the pressurized fluid enters the throttle passage and divides there into a first partial flow directed to the throttle front side and a second partial flow directed to the throttle back side. Both partial flows generated a centering force acting upon the pump shaft due to the Lomakin effect. Thus, if the first supply opening is connected to the intermediate take-off, the throttle bush provides a double acting stabilization of the pump shaft by means of the Lomakin effect. The throttle bush supports the rotordynamic stability both with respect to stiffness and damping in particular of rotor vibrations. The rotor is the entity of the rotating parts of the pump unit, i.e. in particular all impellers as well the pump shaft are part of the rotor of the pump unit.

**[0050]** In addition, it is preferred that a second supply opening is provided at the second stationary part to supply fluid to the second relief passage at a location between the second front side and the second back side with respect to the axial direction, wherein the second supply opening is connectable with the intermediate take-off.

**[0051]** The configuration with the first and the second supply opening has the advantage, that the first and the second multistage pump are interchangeable in the pumping arrangement, even if the pumping arrangement is configured for a subsea application. Each of the two multistage pumps can be used as the first multistage pump of the pumping arrangement, and each of the two multistage pumps can be used as the second multistage pump of the pumping arrangement. The flow connection of the intermediate take-off with the first supply opening as well as the flow connection of the intermediate take-off with the second supply opening can be provided with valve, such as shut-off valves. By adjusting the particular shut-off valves the flow connection between the intermediate take-off and the first as well as the second supply opening can be opened or can be closed. By this measure the position of the particular multistage pump in the pumping arrangement can be easily changed.

**[0052]** Furthermore, it is a preferred design that each multistage pump is configured as a single phase pump for conveying a single phase fluid.

**[0053]** Within this application a "single phase fluid" designates a fluid being in one state of matter. Thus, a single phase fluid is for example a liquid, a gas or a fluid in the dense phase. It has to be noted that a single phase fluid may contain entrainments of another phase. It is known in the art that every single phase liquid pump, can handle a small fraction of a gas phase entrainment. In an analogous manner a single phase dense fluid pump can handle also a small fraction of a liquid entrainment.

**[0054]** The term single phase fluid shall be understood in this manner, namely that the single phase fluid may contain entrainments of one or more other phase(s), for example up to 5 Mol% or up to 5 Vol%.

**[0055]** The primary purpose of e single-phase pump is to pump a single phase fluid.

**[0056]** According to a preferred configuration each multistage pump is configured for conveying a compressible fluid, being in the dense phase at the pump outlet of the second multistage pump.

**[0057]** Within the scope of this application the term "compressible fluid" is used for a fluid having a specific gravity relative to water, which is at most 0.9, and preferably at least 0.2 and at most 0.8. As it is commonly used in the art, the specific gravity is the ratio of the density of said fluid to the density of a reference substance. Within the scope of this application the reference fluid is water.

**[0058]** In addition, it is preferred that the "compressible fluid" has a dynamic viscosity, which is comparable to the viscosity of a gas, and preferably at least 0.005 mPa·s and at most 0.1 mPa·s. The SI unit Millipascal times second corresponds to the also used unit Centipoise (cP), i.e. 1 mPa·s equals 1 cP.

**[0059]** Furthermore, the term "compressible fluid" also encompasses a fluid in the supercritical stage, or the dense phase, respectively. Each fluid in the supercritical phase, i.e. in the dense phase, is a compressible fluid as defined hereinbefore.

**[0060]** According to a preferred embodiment the pumping arrangement is configured for installation on a sea ground.

**[0061]** Preferably, the pumping arrangement is configured for injecting a compressible fluid being in the dense state at the pump outlet of the second multistage pump into a subterranean region. The compressible fluid is e.g. a mixture containing carbon dioxide. By means of a separator the carbon dioxide/natural gas/methane can be separated from the oil at a subsea location e.g. on the sea ground. Thus, the crude oil containing the light components such as carbon dioxide, methane, ethane is separated at the sea ground into a heavier liquid enriched phase, which is delivered to a topside location, and into a lighter $CO_2$ and $CH_4$ enriched phase, which is supplied to the pumping arrangement and reinjected into a subterranean region, e.g. the oil reservoir.

**[0062]** By this measure it is no longer necessary to transport the gas-liquid mixture from the subsea location to a topside location, to at least partially remove the carbon dioxide, to compress the carbon dioxide and to transport the carbon dioxide back to a subsea location for the injection into a reservoir.

**[0063]** Further advantageous measures and embodiments of the invention will become apparent from the dependent claims.

**[0064]** The invention will be explained in more detail hereinafter with reference to embodiments of the invention and with reference to the drawings. There are shown in a schematic representation:

Fig. 1: a schematic representation of the phase diagram of pure $CO_2$ (single component fluid),

Fig. 2: as Fig. 1, but for a multi-component fluid at one defined composition,

Fig. 3: a schematic cross-sectional view of a first embodiment of a pumping arrangement according to the invention,

Fig. 4: a schematic cross-sectional view of a second embodiment of a pumping arrangement according to the invention,

Fig. 5: a schematic cross-sectional view of a third embodiment of a pumping arrangement according to the invention,

Fig. 6: a schematic cross-sectional view of a fourth embodiment of a pumping arrangement according to the invention,

Fig. 7: a schematic cross-sectional view of a variant of the fourth embodiment,

Fig. 8: a schematic cross-sectional view of a fifth embodiment of a pumping arrangement according to the invention,

Fig. 9: a schematic cross-sectional view of a sixth embodiment of a pumping arrangement according to the invention, and

Fig. 10: a schematic representation of an embodiment of a pumping arrangement according to the invention comprising a by-pass piping.

**[0065]** Fig. 3 shows a schematic cross-sectional view of a first embodiment of a pumping arrangement according to the invention, which is designated in its entity with reference numeral 1. The pumping arrangement 1 comprises at least a first multistage pump 10a and a second multistage pump 10b. Each of the multistage pumps 10a, 10b comprises a housing 2 with a pump unit 3 arranged in the housing 2. Each housing 2 comprises a pump inlet 21 for receiving a fluid with a suction pressure and a pump outlet 22 for discharging the fluid with a discharge pressure. The first multistage pump 10a and the second multistage pump 10b are arranged in series. Therefore, the pump outlet 22 of the first multistage pump 10a is connected to the pump inlet 21 of the second multistage pump 10b by means of a connecting line 11. The discharge pressure of the first multistage pump 10a is at least approximately the same as the suction pressure of the second multistage pump 10b.

**[0066]** By way of example, in the following description reference is made to an important application, for which the pumping arrangement 1 is configured as a subsea pumping arrangement 1 configured for an installation on a sea ground. It has to be noted, that the pumping arrangement 1 according to the invention is not restricted to subsea applications or to a deployment on the sea ground. The pumping arrangement 1 according to the invention can also be configured for an installation at topside locations on or above the water surface. For example, the pumping arrangement 1 can be configured to be arranged ashore or on an oil platform, in particular on an unmanned platform, or on a FPSO (Floating Production Storage and Offloading Unit).

**[0067]** Furthermore, in the example referred to hereinafter, the pumping arrangement 1 is designed for conveying a compressible fluid having a specific gravity of at most 0.9, preferably between 0.2 and 0.8. Each of the multistage pumps 10a and 10b is configured as a centrifugal pump. Each multistage pump 10a, 10b comprises a drive unit 4 arranged within the housing 2. The housing 2 is designed as a pressure housing, which is able to withstand the pressure generated by the respective multistage pump 10a, 10b as well as the pressure exerted on the multistage pump 10a, 10b by the environment. The housing 2 may comprise several housing parts, e.g. a pump housing, bearing housing (s), seal housing (s) and a drive housing, which are connected to each other to form the housing 2 surrounding the pump unit 3 and the drive unit 4. It is also possible that a separate pump housing and a separate motor housing are inserted into the housing 2. The housing 2 is configured as a hermetically sealed pressure housing preventing any leakage to the external environment.

**[0068]** In particular, the pumping arrangement 1 is configured for conveying the compressible fluid being in the dense phase, i.e. in the supercritical phase or state. As to the meaning of "dense phase" and "supercritical phase" reference is made to Fig. 1, Fig. 2 and the explanations in the introduction of this application. The phase diagrams in Fig. 1 and Fig. 2 have already been explained in the introduction. Both in Fig. 1 and in Fig. 2 points E1, E2, E3 are shown, which are connected in each case by a dotted line to point A1 or point A2 or point A3, respectively. The indices 1, 2, 3 refer to three different applications or designs of the pumping arrangement 1. The points E1, E2, E3 indicate the particular state (pressure, temperature) of the fluid at the pump inlet 21 of the first multistage pump 1, and the points A1, A2 and A3, respectively indicate the particular state (pressure, temperature) of the fluid at the pump outlet 22 of the second multistage pump 10b for the same application or design of the pumping arrangement 1. As can be seen for the application represented by the points E1 and A1 the fluid is in the liquid phase at the pump inlet 21 of the first multistage pump 10a and in the dense phase at the pump outlet 22 of the second multistage pump 10b. For the two other applications represented by the points E2, A2 and the points E3 and A3 the fluid is in the dense phase, both at the pump inlet 21 of the first multistage pump 10a and at the pump outlet 22 of the second multistage pump 10b.

**[0069]** In the following description reference is made by way of example to the important application that the pumping arrangement 1 is designed and adapted for being used as a subsea injection pumping arrangement 1 in the oil and gas industry, in particular for injecting a fluid being in the dense phase at least at the pump outlet 22 of the second multistage pump 10b into a subterranean oil and/or gas reservoir to increase recovery of hydrocarbons from the subterranean region. The compressible fluid contains for example carbon dioxide ($CO_2$) and may contain also other constituents, such as natural gas, methane ($CH_4$) or the like. In addition, the compressible fluid may also comprise a certain amount of one or more liquid(s), for example water or oil. Typically, the content of liquid(s) does not exceed ten percent by volume and preferably is less than two percent by volume. Thus, the term "compressible fluid" is not restricted to a single substance, such as $CO_2$ but also encompasses mixtures e.g. mixtures in the dense phase or in the supercritical state. The term "compressible fluid" shall be understood in such a manner that the fluid in its entity behaves like a compressible fluid having a specific gravity relative to water which is at most 0.9 and preferably between 0.2 and 0.8. Preferably, the "compressible fluid" has a dynamic viscosity, which is comparable to the viscosity of a gas, and preferably at least 0.005 mPa·s and at most 0.1 mPa·s. The SI unit Millipascal times second corresponds to the also used unit Centipoise (cP), i.e. 1 mPa·s equals 1 cP. Particularly preferred, the compressible fluid contains at least 20 mol% of $CO_2$. In particular, a fluid in the dense state is a compressible fluid.

**[0070]** At a subsea location on a sea ground the $CO_2$ is for example separated from a stream of crude oil emerging from a production well of a subterranean oil field. More generally, a separation device 100 (see e.g. Fig. 6) separates the crude oil in a heavier phase having a higher density and a lighter phase having a lower density. The lighter phase is enriched with methane and carbon dioxide and the heavier phase comprises predominantly liquid hydrocarbons. The heavier phase is conveyed for example to a topside location for further processing as it is indicated by the arrow T for example in Fig. 6. The lighter phase, which contains a considerable amount of $CO_2$, is fed to the pumping arrangement 1, more precisely to the pump inlet 21 of the first multistage pump 10a and injected into a subterranean region of the oil field from the pump outlet 22 of the second multistage pump 10b as it is indicated by the arrow I for example in Fig. 6. Due to the pressure and temperature at the subsea location the $CO_2$ containing lighter phase is in the dense phase, i.e. in the supercritical state, at least when said lighter phase is discharged from the pumping arrangement 1.

**[0071]** By injecting the fluid in the dense phase into the oil reservoir the hydrocarbons are forced to flow towards and out of the production well. The pumping arrangement 1 can be configured for installation on the sea ground, i.e. for use beneath the water surface, in particular down to a depth of 100 m, down to 1000 m or even down to more than 2000 m beneath the water surface of the sea.

**[0072]** Typically, the pump outlet 22 of the second multistage pump 10b is connected to a pipe (not shown in detail) for delivering the pressurized fluid to a well, in which the fluid is injected. The pressure of the fluid at the pump outlet 22 of the second multistage pump10b is referred to as 'high pressure' whereas the pressure of the fluid at the pump inlet 21 of the first multistage pump 10a is referred to as 'low pressure'. Atypical value for the difference between the high pressure and the low pressure is for example 100 to 200 bar (10 - 20 MPa).

**[0073]** The pumping arrangement 1 is in particular advantageous for applications, where a high injection pressure is required. For such applications it may be more efficient to arrange two or more multistage pumps 10a, 10b in series rather than adding additional stages to a single multistage pump.

**[0074]** The pump outlet 22 of the first multistage pump 10a is connected to the pump inlet 21 of the second multistage pump 10b by means of the connecting line 11. In some embodiments the pump outlet 22 of the first multistage pump 10a is directly connected to the pump inlet 21 of the second multistage pump 10b without any additional device in between. In other embodiments - as it is shown in Fig. 3 - one or more additional device(s) 113, 114 is/are arranged between the pump outlet 22 of the first multistage pump 10a and the pump inlet 21 of the second multistage pump 10b. In the embodiment illustrated in Fig. 3 the connecting line 11 comprises a cooling device 113 and a buffer 114. The cooling device 113 is for example configured as a seawater cooled heat exchanger to remove in particular the heat which is generated by the compression of the compressible fluid. The buffer 114 is configured to reduce pressure fluctuations. The outlet 22 of the first multistage pump 10a is connected to the cooling device 113. From the cooling device 113 the pressurized fluid is guided to the buffer 114. The outlet of the buffer 114 is connected to the inlet 21 of the second multistage pump 10b. The outlet 22 of the second multistage pump 10b is connected to a well (not shown) leading to a subterranean region, in which the fluid, e.g. the carbon dioxide, is injected.

**[0075]** Basically, each of the multistage pumps 10a, 10b can be configured in an analogous manner as it has been disclosed in EP 3 771 828 A1. According to the disclosure of EP 3 771 828 A1, the multistage pump is configured with radial or semi-axial impellers, wherein at least two impellers of the same multistage pump have different specific speeds. The pumping arrangement 1 according to the present invention can also be configured with helico-axial impellers, i.e. each of the multistage pumps 10a, 10b can be configured as a helico-axial pump.

**[0076]** Referring to Fig. 3 the first embodiment of the pumping arrangement 1 according to the invention will be explained in more detail. Firstly, the general configuration of each of the multistage pumps 10a, 10b will be described, which is the same for the first multistage pump 10a and the second multistage pump 10b. After that, the specific features of the second multistage pump 10b will be described.

**[0077]** The pump unit 3 of the multistage pump 10a, 10b comprises a pump shaft 5 extending from a drive end 51 to a non-drive end 52 of the pump shaft 5. The pump shaft 5 is configured for rotating about an axial direction A, which is defined by the longitudinal axis of the pump shaft 5.

**[0078]** The pump unit 3 further comprises a plurality of impellers with a first stage impeller 31, a last stage impeller 32 and optionally a number of intermediate stage impellers 33. In the first embodiment the multistage pump 10a, 10b is a nine stage pump having the first stage impeller 31, the last stage impeller 32 and seven intermediate stage impellers 33, which are all arranged in series on the pump shaft 5. Of course, the number of nine stages is only exemplary. In other embodiments the multistage pump 10a, 10b may comprise more than nine stages, e.g. ten or twelve stages, or less than nine stages for example four or two stages.

**[0079]** The first stage impeller 31 is the first impeller when viewed in the direction of the streaming fluid, i.e. the first stage impeller 31 is located next to the pump inlet 21 receiving the fluid having the suction pressure. The last stage impeller 32 is the last impeller when viewed in the direction of the streaming fluid, i.e. the last stage impeller 32 is located next to the pump outlet 22, at which the fluid is discharged having the discharge pressure.

**[0080]** Each impeller 31, 32, 33 is fixedly mounted on the pump shaft 5 in a torque proof manner. The plurality of impellers 31, 32, 33 is arranged in series and configured for increasing the pressure of the fluid from the suction pressure to the discharge pressure.

**[0081]** The drive unit 4 is configured to exert a torque on the drive end 51 of the pump shaft 5 for driving the rotation of the pump shaft 5 and the impellers 31, 32, 33 about the axial direction A.

**[0082]** The multistage pump 10a, 10b is configured as a vertical pump 10a, 10b, meaning that during operation the pump shaft 5 is extending in the vertical direction, which is the direction of gravity. Thus, the axial direction A coincides with the vertical direction.

**[0083]** In other embodiments the multistage pump may be configured as a horizontal pump, meaning that during operation the pump shaft is extending horizontally, i.e. the axial direction A is perpendicular to the direction of gravity.

**[0084]** A direction perpendicular to the axial direction A is referred to as radial direction. The term 'axial' or 'axially' is used with the common meaning 'in axial direction' or 'with respect to the axial direction'. In an analogous manner the term 'radial' or 'radially' is used with the common meaning 'in radial direction' or 'with respect to the radial direction'. Hereinafter relative terms regarding the location like "above" or "below" or "upper" or "lower" or "top" or "bottom" refer to the usual operating position of the pump 1. Fig. 3-Fig. 5 show the pump 10a, 10b in the usual operating position.

**[0085]** Referring to this usual orientation during operation and as shown in Fig. 3 the drive unit 4 is located above the pump unit 3. However, in other embodiments the pump unit 3 may be located on top of the drive unit 4.

**[0086]** The multistage pump 10a, 10b is designed with an inline arrangement of all impellers 31, 32, 33. In an inline arrangement all impellers 31, 32, 33 are configured such that the axial thrusts generated by the individual rotating impellers 31, 32, 33 are all directed in the same direction, namely downwards in the axial direction A in Fig. 3. In addition, the main flow of the fluid from the pump inlet 21 (low pressure) towards the pump outlet 22 (high pressure) is always directed in the same direction, namely in upward direction. The flow of the fluid through the multistage pumps 10a, 10b is indicated by the dashed arrows without reference numeral.

**[0087]** In some embodiments each of the impellers 31, 32, 33 is configured as a radial impeller or as a semi-axial impeller 31, 32 , 33 as it is disclosed e.g. in EP 3 771 828 A1. As it is commonly used in the art a radial impeller is configured to deflect the flow of fluid from the axial direction in a radial direction, and a semi-axial impeller is configured to deflect the flow of fluid from the axial direction in a direction, which has both an axial component and a radial component different from zero. In other embodiments each of the impellers 31, 32, 33 is configured as a helico-axial impeller as it is known in the art for multiphase pumps.

**[0088]** The multistage pump 10a, 10b further comprises a plurality of bearings. A first radial pump bearing 53, a second radial pump bearing 54 and an axial pump bearing 55 are provided for supporting the pump shaft 5. The first radial pump bearing 53, which is the upper one according to the representation in Fig. 3, is arranged adjacent to the drive end 51 of the pump shaft 5 between the pump unit 3 and the drive unit 4. The second radial pump bearing 54, which is the lower one, is arranged between the pump unit 3 and the non-drive end 52 of the pump shaft 5 or at the non-drive end 52. The axial pump bearing 55 is arranged between the pump unit 3 and the first radial pump bearing 53. The pump bearings 53, 54, 55 are configured to support the pump shaft 5 both in axial and radial direction. The radial pump bearing 53 and 54 are supporting the pump shaft 5 with respect to the radial direction, and the axial bearing 55 is supporting the pump shaft 5 with respect to the axial direction A. The first radial pump bearing 53 and the axial pump bearing 55 are arranged such that the first radial pump bearing 53 is closer to the drive unit 4 and the axial pump bearing 55 is facing the pump unit 3. Of course, it is also possible, to exchange the position of the first radial pump bearing 53 and the axial pump bearing 55, i.e. to arrange the first radial pump bearing 53 between the axial pump bearing 55 and the pump unit 3, so that the axial pump bearing 55 is closer to the drive unit 4.

**[0089]** A radial bearing, such as the first or the second radial pump bearing 53 or 54 is also referred to as a "journal bearing" and an axial bearing, such as the axial pump bearing 55, is also referred to as an "thrust bearing". The first radial pump bearing 53 and the axial pump bearing 55 may be configured as separate bearings, but it is also possible that the first radial pump bearing 53 and the axial pump bearing 55 are configured as a single combined radial and axial bearing supporting the pump shaft 5 both in radial and in axial direction.

**[0090]** The second radial pump bearing 54 is supporting the pump shaft 5 in radial direction. In the embodiment shown in Fig. 3, there is no axial pump bearing provided at the non-drive end 52 of the pump shaft 5. Of course, in other embodiments it is also possible that an axial pump bearing for the pump shaft 5 is provided at the non-drive end 52. In embodiments, where an axial pump bearing is provided at the non-drive end 52, a second axial pump bearing may be provided at the drive end 51 or the drive end 51 may be configured without an axial pump bearing.

**[0091]** Preferably the radial pump bearings 53 and 54 as well as the axial pump bearing 55 are configured as hydrodynamic bearings, and even more preferred as tilting pad bearings 53, 54 and 55, respectively. Specifically preferred at least the first radial pump bearing 53 and the second radial pump bearing 54 are each configured as a radial tilting pad bearing. Of course, it is also possible that the first radial pump bearing 53 and the second radial pump bearing 54 are each configured as fixed multilobe hydrodynamic bearing.

**[0092]** Preferably, the multistage pump 10a, 10b comprises at least one balancing device for at least partially balancing the axial thrust that is generated by the impellers 31, 32, 33 during operation of the multistage pump 1. Here, the multistage pump 10a, 10b comprises a first balance drum 7 (also referred to as throttle bush). During operation the first balance drum 7 at least partially balances the axial thrust that is generated by the impellers 31, 32, 33.

**[0093]** The first balance drum 7 is fixedly connected to the pump shaft 5 in a torque proof manner. The first balance drum 7 is arranged above the upper end of the pump unit 3, namely between the pump unit 3 and the drive end 51 of the pump shaft 5, more precisely between the upper end of the pump unit 3 and the axial pump bearing 55. The first balance drum 7 defines a first front side 71 and a first back side 72. The first front side 71 is the side facing the pump unit 3. In the first embodiment the first front side 71 is facing the last stage impeller 32. The first back side 72 is the side facing the axial pump bearing 55 and the drive unit 4. The first balance drum 7 is surrounded by a first stationary part 27, so that a first relief passage 73 is formed between the radially outer surface of the first balance drum 7 and the first stationary part 27. The first stationary part 27 is configured to be stationary with respect to the housing 2. The first relief passage 73 forms an annular gap between the outer surface of the first balance drum 7 and the first stationary part 27 and extends from the first front side 71 to the first back side 72.

**[0094]** A balance line 9 is provided for recirculating the fluid from the first back side 72 of the first balance drum 7 to a low pressure side of the multistage pump 10a, 10b. In both the first multistage pump 10a and the second multistage pump 10b the balance line 9 is connected to the respective first back side 72. The low pressure side, with which the balance line 9 is in fluid communication, is exposed to the same pressure for both the first multistage pump 10a and the second multistage pump 10b as will be explained in more detail later on. Said low pressure side is the suction side of the first multistage pump 10a, meaning that both the balance line 9 of the first multistage pump 10a and the balance line 9 of the second multistage pump 10b are in fluid communication with the pump inlet 21 of the first multistage pump 10a. Therefore, when neglecting minor pressure drops across the balance lines 9, both at the first back side 72 of the first multistage pump 10a and at the first back side 72 of the second multistage pump 10b a pressure prevails, which equals the suction pressure of the first multistage pump 10a.

**[0095]** During operation, in each of the multistage pumps 10a, 10b a part of the pressurized fluid passes from the first front side 71 through the first relief passage 73 to the first back side 72, enters the balance line 9 and is recirculated to the low pressure side.

**[0096]** The balance line 9 may be arranged - as shown in Fig. 3 - outside the housing 2. In other embodiments the balance line 9 may be designed as internal line completely extending within the housing 2.

**[0097]** Due to the balance lines 9 the pressure prevailing at the first back side 72 is in both the first multistage pump 1 and the second multistage pump essentially the same - apart from a minor pressure drop caused by the balance lines 9 - as the suction pressure prevailing at the pump inlet 21 of the first multistage pump 10a.

**[0098]** The axial surface of the first balance drum 7 facing the first front side 71 is exposed to a pressure, which is essentially the same as the discharge pressure of the first multistage pump 10a or the second multistage pump 10b, respectively. Of course, due to smaller pressure losses the pressure prevailing at the axial surface of the first balance drum 7 facing the first front side 71 may be somewhat smaller than the respective discharge pressure. However, the considerably larger pressure drop takes place over the first balance drum 7. At the first back side 72 it is essentially the suction pressure of the first multistage pump 10a that prevails during operation of the multistage pump 10a, 10b. Thus, the pressure drop over the first balance drum 7 is essentially the difference between the discharge pressure of the respective multistage pump 10a, 10b and the suction pressure of the first multistage pump 10a.

**[0099]** The pressure drop over the first balance drum 7 results in a force that is directed upwardly in the axial direction A according to the representation in Fig. 3 and therewith counteracts the downwardly directed axial thrust generated by the impellers 31, 32, 33.

**[0100]** In both the first multistage pump 10a and the second multistage pump 10b the respective drive unit 4 comprises an electric motor 41 and a drive shaft 42 extending in the axial direction A. For supporting the drive shaft 42 a first radial drive bearing 43, a second radial drive bearing 44 and an axial drive bearing 45 are provided, wherein the second radial drive bearing 44 and the axial drive bearing 45 are arranged above the electric motor 41 with respect to the axial direction A, and the first radial drive bearing 43 is arranged below the electric motor 41. The electric motor 41, which is arranged between the first and the second radial drive bearing 43, 44, is configured for rotating the drive shaft 42 about the axial direction A. The drive shaft 42 is connected to the drive end 51 of the pump shaft 5 by means of a coupling 8 for transferring a torque to the pump shaft 5.

**[0101]** The drive bearings 43, 44 and 45 are configured to support the drive shaft 42 both in radial direction and in the axial direction A. The first and the second radial drive bearing 43, 44 support the drive shaft 42 with respect to the radial direction, and the axial drive bearing 45 supports the drive shaft 42 with respect to the axial direction A. The second radial drive bearing 44 and the axial drive bearing 45 are arranged such that the second radial drive bearing 44 is arranged between the axial drive bearing 45 and the electric motor 41.

**[0102]** Of course, it is also possible, to exchange the position of the second radial drive bearing 44 and the axial drive bearing 45.

**[0103]** The second radial drive bearing 44 and the axial drive bearing 45 may be configured as separate bearings, but it is also possible that the second radial drive bearing 44 and the axial drive bearing 45 are configured as a single combined radial and axial bearing supporting the drive shaft 42 both in radial and in axial direction A.

**[0104]** The first radial drive bearing 43 is arranged below the electric motor 41 and supports the drive shaft 42 in radial direction. In the embodiment shown in Fig. 1, there is no axial bearing arranged below the electric motor 41. Of course, it is also possible that an axial drive bearing for the drive shaft 42 is - alternatively or additionally - arranged below the electric motor 41, i.e. between the electric motor 41 and the coupling 8.

**[0105]** The electric motor 41 of the drive unit 4 may be configured as a cable wound motor. In a cable wound motor the individual wires of the motor stator (not shown), which form the coils for generating the electromagnetic field(s) for driving the motor rotor (not shown), are each insulated, so that the motor stator may be flooded for example with a barrier fluid. Alternatively, the electric motor 41 may be configured as a canned motor. When the electric drive 41 is configured as a canned motor, the annular gap between the motor rotor and the motor stator of the electric motor 41 is radially outwardly delimited by a can (not shown) that seals the motor stator hermetically with respect to the motor rotor and the annular gap. Thus, any fluid flowing through the gap between the motor rotor and the motor stator cannot enter the motor stator. When the electric motor 41 is designed as a canned motor a dielectric cooling fluid may be circulated through the hermetically sealed motor stator for cooling the motor stator.

**[0106]** Preferably, the electric motor 41 is configured as a permanent magnet motor or as an induction motor. To supply the electric motor 41 with energy, a power penetrator (not shown) is provided at the common housing 2 for receiving a power cable (not shown) that supplies the electric motor 41 with power.

**[0107]** The electric motor 41 may be designed to operate with a variable frequency drive (VFD), in which the speed of the motor 41, i.e. the frequency of the rotation, is adjustable by varying the frequency and/or the voltage supplied to the electric motor 41. However, it is also possible that the electric motor 41 is configured differently, for example as a single speed or single frequency drive.

**[0108]** The drive shaft 42 is connected to the drive end 51 of the pump shaft 5 by means of the coupling 8 for transferring a torque to the pump shaft 5. Preferably the coupling 8 is configured as a flexible coupling 8, which connects the drive shaft 42 to the pump shaft 5 in a torque proof manner, but allows for a relative lateral (radial) and/or axial movement between the drive shaft 42 and the pump shaft 5. Thus, the flexible coupling 8 transfers the torque but no or nearly no lateral vibrations. Preferably, the flexible coupling 8 is configured as a mechanical coupling 8. In other embodiments the flexible coupling may be designed as a magnetic coupling, a hydrodynamic coupling or any other coupling that is suited to transfer a torque from the drive shaft 42 to the pump shaft 5.

**[0109]** The multistage pump 10a, 10b further comprises at least a first mechanical seal 50 for sealing the pump unit 3 at the pump shaft 5 against a leakage of the fluid along the pump shaft 5. With respect to the axial direction A the first mechanical seal 50 is arranged between the first balance drum 7 and the pump bearings 53, 55. By means of the first mechanical seal 50 the fluid is prevented from entering the first radial pump bearing 53 and the axial pump bearing 55 as well as the drive unit 4. The first mechanical seal 50 has a process side 59, which is the side facing the pump unit 3. The first mechanical seal 50 is arranged adjacent -with respect to the axial direction A- to the first balance drum 7, so that the first back side 72 of the first balance drum 7 is arranged between the first balance drum 7 and the first mechanical seal 50. Thus, the pressure prevailing at the process side 59 of the first mechanical seal 50 is the same pressure as the pressure prevailing at the first back side 72 of the first balance drum 7.

**[0110]** In the first embodiment of the pumping arrangement 1 both the first multistage pump 10a and the second multistage pump 10b comprise a second mechanical seal 60 for sealing the pump unit 3 at the pump shaft 5 against a leakage of the fluid along the pump shaft 5. With respect to the axial direction A the second mechanical seal 60 is arranged between the pump unit 3 and the non-drive end 52 of the pump shaft 5, more precisely between the pump unit 3 and the second radial pump bearing 54. By means of the second mechanical seal 60 the fluid is prevented from entering the second radial pump bearing 54. The second mechanical seal 60 has a process side 69, which is the side facing the pump unit 3. The second mechanical seal 60 is arranged adjacent -with respect to the axial direction A- to the second radial pump bearing 54. The process side 69 of the second mechanical seal 60 is in fluid communication with the balance line 9. Thus, the pressure prevailing at the process side 69 of the second mechanical seal 60 is the same pressure as the pressure prevailing at the first back side 72 of the first balance drum 7, and therewith the same pressure that prevails at the process side 59 of the first mechanical seal 50.

**[0111]** Mechanical seals 50, 60 are well-known in the art in many different embodiments and therefore require no detailed explanation. In principle, a mechanical seal 50, 60 is a seal for a rotating shaft and comprises a rotor (not shown) fixed to the pump shaft 5 and rotating with the pump shaft 5, as well as a stationary stator (not shown) fixed with respect to the housing 2. During operation, the rotor and the stator are sliding along each other - usually with a liquid there between - for providing a sealing action to prevent the fluid from escaping to the environment or entering the drive unit 4, or the pump bearings 53, 54, 55, respectively.

**[0112]** For the lubrication and the cooling of the mechanical seals 50, 60 a barrier fluid system (not shown) is provided. Barrier fluid systems for mechanical seals 50, 60 as such are well-known in the art since many years and therefore do not require a detailed explanation. A barrier fluid system comprises a reservoir for a barrier fluid as well as a circuit through which the barrier fluid is moved. The circuit is designed e.g. such that the barrier fluid passes through the mechanical seals 50, 60. The barrier fluid system may also comprise a heat exchanger for cooling the barrier fluid as well as a pressure control device for controlling the pressure of the barrier fluid in the circuit. The pressure of the barrier fluid in the barrier fluid system is controlled such that the pressure of the barrier fluid is at least as high as but preferably higher than the pressure prevailing at the process side 59, 69 of the mechanical seals 50, 60. According to a preferred configuration, the pressure in the barrier fluid system is higher, e.g. 10% and at least 2 bar higher, than the suction pressure at the pump inlet 21 of the first multistage pump 10a.

**[0113]** By this measure there is always a leakage flow of barrier fluid through the mechanical seals 50, 60 into the pump unit 3. Therefore any leakage flow of the fluid from the pump unit 3 through the mechanical seals 50, 60 into the drive unit 4 or the pump bearings 53, 54, 55 is reliably prevented. The amount of barrier fluid, that is lost by the leakage into the pump unit 3 is replaced from the reservoir for the barrier fluid.

**[0114]** Still referring to Fig. 3, specific features of the second multistage pump 10b will now be described.

**[0115]** The second multistage pump 10b comprises a second balance drum 6, which is fixedly connected to the pump shaft 5 between the pump unit 3 and the non-drive end 52 of the pump shaft 5, more precisely, between the first stage impeller 31 of the pump unit 3 and second mechanical seal 60. The second balance drum 6 defines a second front side 61 facing the pump unit 3 and a second back side 62, wherein a second relief passage 63 is provided between the second balance drum 6 and a second stationary part 26 configured to be stationary with respect to the housing 2, the second relief passage 63 extending from the second front side 61 to the second back side 62. The second front side 61 is in fluid communication with the pump inlet 21 of the second multistage pump 10b. The second balance drum 6 is arranged adjacent - with respect to the axial direction A - to the first stage impeller 31 of the pump unit 3.

**[0116]** Since the second mechanical seal 60 is arranged adjacent -with respect to the axial direction A- to the second balance drum 6, the second back side 62 of the second balance drum 6 is arranged between the second balance drum 6 and the second mechanical seal 60. Thus, the pressure prevailing at the process side 69 of the second mechanical seal 60 is the same pressure as the pressure prevailing at the second back side 62 of the second balance drum 6.

**[0117]** Furthermore, a pressure equalizing line 20 is provided configured for a fluid communication between the process side 69 of the second mechanical seal 60 of the second multistage pump 10b and the suction side of the first multistage pump 10a. A first end 201 of the pressure equalizing line 20 is connected to the process side 69 of the second mechanical seal 60 of the second multistage pump 10b, and a second end 202 is connected to the pump inlet 21 of the first multistage pump 10a. By means of the pressure equalizing line 20 it is ensured that during operation the process side 69 of the second mechanical seal 60 of the second multistage pump 10b is always exposed to the suction pressure of the first multistage pump 10a. Thus, the pressure prevailing at the process side 69 of the second mechanical seal 60 of the second multistage pump 10b is decoupled from the discharge pressure of the first multistage pump 10a. Independent from the discharge pressure of the first multistage pump 10a, the pressure prevailing at the process side 69 of the second mechanical seal 60 of the second multistage pump 10b always equals the suction pressure of the first multistage pump 10a.

**[0118]** Furthermore, due to the balance line 9 of the second multistage pump 10b, which constitutes a fluid communication between the process side 59 of the first mechanical seal 50 of the second multistage pump 10b and the process side 69 of the second mechanical seal 60 of the second multistage pump 2, also the pressure prevailing at the process side 59 of the first mechanical seal 50 of the second multistage pump 50 equals the suction pressure of the first multistage pump 10b. Thus, also the process side 59 of the first mechanical seal 50 of the second multistage pump 10b is decoupled from the discharge pressure of the first multistage pump 10a.

**[0119]** Thus, during operation of the pumping arrangement 1, the process sides 59, 69 of both mechanical seals 50, 60 of the second multistage pump 10b are exposed to a pressure which equals the suction pressure of the first multistage pump 10. In particular, said pressure prevailing at the process sides 59, 69 of both mechanical seals 50, 60 of the second multistage pump 10b is decoupled from the discharge pressure of the first multistage pump 10a. Therefore, when a fluid slug, e.g. a water slug, occurs and propagates through the first multistage pump 10a therewith causing a pressure spike at the pump outlet 22 of the first multistage pump 10a, this has no detrimental impact on the mechanical seals 50, 60 of the second multistage pump 10b, because the process sides 59, 69 of the mechanical seals 50, 60 of the second multistage pump 10b are decoupled from the discharge pressure of the first multistage pump 10a. The process sides 59, 69 of the mechanical seals 50, 60 are always exposed to a constant pressure, namely the suction pressure of the first multistage pump 10a, which ensures a reliable, economical and low wear operation of the mechanical seals 50, 60 with a low consumption of barrier fluid.

**[0120]** Fig. 4 shows a schematic cross-sectional view of a second embodiment of a pumping arrangement 1 according to the invention.

**[0121]** In the following description of the second embodiment of the pumping arrangement 1 only the differences to the first embodiment are explained in more detail. The explanations with respect to the first embodiment are also valid in the same way or in analogously the same way for the second embodiment. Same reference numerals designate the same features that have been explained with reference to the first embodiment or functionally equivalent features.

**[0122]** Compared to the first embodiment, it is the main difference, that the second multistage pump 10b of the second embodiment of the pumping arrangement 1 does not have the second mechanical seal 60 at the non-drive end 52 of the pump shaft 5, and does not have the second radial pump bearing 54 at the non-drive end 52 of the pump shaft 5.

**[0123]** Since the second multistage pump 10b is provided with the second balance drum 6 arranged at the non-drive end 52 of the pump shaft 5, there is no need for a separate mechanical seal or a separate radial pump bearing at the non-drive end 52 of the pump shaft 5. Due to the fluid passing through the second relief passage 63 between the second balance drum 6 and the second stationary part 26 there is are centering and bearing forces acting on the pump shaft 5, e.g. due to the Lomakin effect. Furthermore, because of the pressure equalizing line 20 connecting the second back side 62 of the second balance drum 6 with the pump inlet 21 of the first multistage pump 10a, the fluid passing from the second front side 61 through the relief passage 63 to the second back side 62 of the second balance drum 6 is recirculated to the pump inlet 21 of the first multistage pump 10a. Thus, there is no need for the second mechanical seal 6 at the non-drive end 52 of the pump shaft.

**[0124]** Due to the pressure equalizing line 20 and the balance line 9 of the second multistage pump 10b the pressure prevailing at the first back side 72 of the first balance drum 7 of the second multistage pump 10b equals the pressure at the second back side 62 of the second balance drum 6 of the second multistage pump 10b and equals the suction pressure of the first multistage pump 10a. During operation, the first balance drum 7 of the second multistage pump 10b generates a force on the pump shaft 5, which is directed upwardly according to the representation in Fig. 4 and proportional to the pressure difference between the discharge pressure of the second multistage pump 10b and the suction pressure of the first multistage pump 10a. The second balance drum 6 of the second multistage pump 10b generates a balancing force, which is directed downwardly according to the representation in Fig. 4 and proportional to the pressure difference between the discharge pressure of the first multistage pump 10a and the suction pressure of the first multistage pump 10a. In sum, both balance drums 6, 7 generate a resulting force on the pump shaft 5, which is directed upwardly according to the representation in Fig. 4 and proportional to the pressure difference between the discharge pressure of the second multistage pump 10b and the suction pressure of the second multistage pump 10b. Thus, in sum, the two balance drums 6, 7 of the second multistage pump 10b generate a balancing force, that is essentially the same as it is generated in a conventional multistage pump with only one balance drum.

**[0125]** The configuration of the second embodiment of the pumping arrangement 1 has the additional advantage that both multistage pumps 10a, 10b can have an identical housing 2, but for the second multistage pump 10b the second mechanical seal 6 and the second radial pump bearing 54 at the non-drive end 52 of the pump shaft 5 are replaced with the second balance drum 6.

**[0126]** Fig. 5 shows a schematic cross-sectional view of a third embodiment of a pumping arrangement 1 according to the invention.

**[0127]** In the following description of the third embodiment of the pumping arrangement 1 only the differences to the first and the second embodiment are explained in more detail. The explanations with respect to the first embodiment and the second embodiment are also valid in the same way or in analogously the same way for the third embodiment. Same reference numerals designate the same features that have been explained with reference to the first and the second embodiment or functionally equivalent features.

**[0128]** As can be seen in Fig. 5, in each of the first multistage pump 10a and the second multistage pump 10b the plurality of impellers 31, 32, 33 comprises a first set of impellers 31, 33 and a second set of impellers 32, 33, wherein the first set of impellers 31, 33 and the second set of impellers 32, 33 are arranged in a back-to-back arrangement. The first set of impellers 31, 33 comprises the first stage impeller 31 and the three intermediate stage impellers 33 of the next three stages and the second set of impellers 32, 33 comprises the last stage impeller 32 and the three intermediate stage impellers 33 of the three preceding stages. In other embodiments the first set of impellers may comprise a different number of impellers than the second set of impellers.

**[0129]** In a back-to-back arrangement the first set of impellers 31, 33 and the second set of impellers 32, 33 are arranged such that the axial thrust generated by the action of the rotating first set of impellers 31, 33 is directed in the opposite direction as the axial thrust generated by the action of the rotating second set of impellers 32, 33. As indicated in Fig. 5 by the dashed arrows without reference numeral, the fluid enters the first multistage pump 10a through the pump inlet 21 located at the lower end of the pump section 3, passes the stages one (first stage), two, three and four, is then guided through a crossover line 34 to the suction side of the fifth stage at the upper end of the pump unit 3, passes the stages five, six, seven and eight (last stage), and is then discharged through the pump outlet 22, which is arranged between the upper end and the lower end of the pump unit 3. From the pump outlet of the first multistage pump 10 a the fluid is guided through the connecting line 11 to the pump inlet 21 of the second multistage pump 10b and flows through the second multistage pump 10b in an analogous manner as it has been described for the first multistage pump 10a.

**[0130]** For many applications the back-to-back arrangement for the multistage pumps 10a, 10b is preferred because the axial thrust acting on the pump shaft 5, which is generated by the first set of impellers 31, 33 counteracts the axial thrust, which is generated by the second set of impellers 32, 33. Thus, said two axial thrusts compensate each other at least partially.

**[0131]** Also in the third embodiment each multistage pump 10a, 10b is provided with the first balance drum 7 arranged near the drive end 51 of the pump shaft 5, and the second multistage pump 10 is additionally provided with the second balance drum 6 arranged near the non-drive end 52 of the pump shaft 5.

**[0132]** In each multistage pump 10a, 10b the axial surface of the first balance drum 7 facing the first front side 71 is exposed to an intermediate pressure between the suction pressure and the discharge pressure of the respective multistage pump 10a, 10b. In the third embodiment shown in Fig. 5 said intermediate pressure is the suction pressure of the fifth stage prevailing at the outlet of the crossover line 34 during operation of the multistage pump 10a, 10b. Of course, due to smaller pressure losses the pressure prevailing at the axial surface of the first balance drum 7 facing the first front side 71 may be somewhat smaller than said intermediate pressure. However, the considerably larger pressure drop takes place over the first balance drum 7. At the first back side 72 it is - for each multistage pump 10a, 10b - essentially the suction pressure prevailing at the pump inlet of the first multistage pump 10a that prevails during operation of the pumping arrangement 1. Thus, the pressure drop over the balance drum 7 is essentially the difference between the intermediate pressure of the respective multistage pump 10a, 10b and the suction pressure of the first multistage pump 10a.

**[0133]** The pressure drop over the first balance drum 7 results in a force that is directed upwardly in the axial direction A and therewith counteracts the downwardly directed axial thrust generated by the first set of impellers 31, 33, namely the first stage impeller 31 and the intermediate stage impellers 33 of the second, third and fourth stage.

**[0134]** For further reducing the overall axial thrust acting on the pump shaft 5 each multistage pump 10a, 10b comprises a center bush 35, which is arranged between the first set of impellers 31, 33 and the second set of impellers 32, 33 with respect to the axial direction A. The center bush 35 is fixedly connected to the pump shaft 5 in a torque proof manner and rotates with the pump shaft 5. The center bush 35 is arranged on the pump shaft 5 between the last stage impeller 32, which is the last impeller of the second set of impellers, and the intermediate stage impeller 33 of the fourth stage, which is the last impeller of the first set of impellers, when viewed in the direction of increasing pressure, respectively. The center bush 35 is surrounded by a third stationary part 36 being stationary with respect to the housing 2. A annular balancing passage 37 is formed between the outer surface of the center bush 35 and the third stationary part 36.

**[0135]** The function of the center bush 35 and the balancing passage 37 is in principle the same as the function of the first balance drum 7 and the relief passage 73. At the axial surface of the center bush 35 facing the last stage impeller 32 the high pressure prevails, which is essentially the same as the discharge pressure of the respective multistage pump 10a, 10b, and at the other axial surface facing the intermediate stage impeller 33 of the fourth stage a lower pressure prevails, which is essentially the same as the intermediate pressure for the respective multistage pump 10a, 10b, when neglecting the small pressure losses caused by the crossover line 34. Therefore, the fluid may pass from the last stage impeller 32 through the balancing passage 37 to the intermediate stage impeller 33 of the fourth stage.

**[0136]** The pressure drop over the center bush 35 essentially equals the difference between the discharge pressure and the intermediate pressure of the respective multistage pump 10a, 10b. Said pressure drop over the center bush 35 results in a force that is directed downwardly in the axial direction A and therewith counteracts the upwardly directed axial thrust generated by the second set of impellers 33, 32, namely the intermediate stage impellers 33 of the fifth, sixth and seventh stage and the last stage impeller 32.

**[0137]** It has to be understood that in third embodiment of the pumping device 1 can also be configured such, that the second multistage pump 10b has the second mechanical seal 60 at the non-drive end 52 of the pump shaft 5, and/or the second radial pump bearing 54 at the non-drive end 52 of the pump shaft 5 in an analogous manner as it has been described with respect to the first embodiment.

**[0138]** Fig. 6 shows a schematic cross-sectional view of a fourth embodiment of a pumping arrangement 1 according to the invention.

**[0139]** In the following description of the fourth embodiment of the pumping arrangement 1 only the differences to the previously described embodiments of the pumping arrangement 1 according to the invention are explained in more detail. The explanations with respect to the previously described embodiments are also valid in the same way or in analogously the same way for the fourth embodiment. Same reference numerals designate the same features that have been explained with reference to the previously described embodiments or functionally equivalent features.

**[0140]** In Fig. 6 the separation device 100 is also shown. At the subsea location on a sea ground the separation device 100 receives for example a stream of crude hydrocarbons emerging from a production well of a subterranean oil field as it is indicated by the arrow with the reference numeral W. The separation device 100 separates the crude oil in a heavier phase having a higher density and a lighter phase having a lower density. The lighter phase is enriched with methane and carbon dioxide and the heavier phase comprises predominantly liquid hydrocarbons. The heavier phase is conveyed for example through a delivery pipe 150 to a topside location for further processing as it is indicated by the arrow T. The lighter phase, which contains a considerable amount of $CO_2$, is fed through a feed line 160 to the pumping arrangement 1, more precisely to the pump inlet 21 of the first multistage pump 10a as it is indicated by the arrow with the reference numeral L. Optionally, an additional cooling device 115 can be provided between the separation device 100 and the pump inlet 21 of the first multistage pump 10a for cooling the lighter phase L prior to entering the first multistage pump 10a.

**[0141]** In the fourth embodiment of the pumping arrangement 1 both multistage pumps 10a and 10b are configured with a back-to-back arrangement of the impellers 31, 32, 33 as it has been explained with respect to the third embodiment (Fig. 5). Furthermore, both multistage pumps 10a, 10b are configured with the second mechanical seal 60 for sealing the pump unit 3 at the pump shaft 5 against a leakage of the fluid along the pump shaft 5, as it has been explained with respect to the first embodiment (Fig. 3). With respect to the axial direction A the second mechanical seal 60 is arranged between the pump unit 3 and the non-drive end 52 of the pump shaft 5, more precisely between the pump unit 3 and the second radial pump bearing 54.

**[0142]** The second multistage pump 10b comprises a throttle bush 80, which is fixedly connected to the pump shaft 5 between the first balance drum 7 and the first mechanical seal 50 with respect to the axial direction A. The throttle bush 80 defines a throttle front side 81 facing the first balance drum 7 and a throttle back side 82 facing the first mechanical seal 50. The throttle front side 81 coincides with the first back side 72 of the first balance drum 7. A throttle passage 83 is provided between the throttle bush 80 and a stationary throttle part 84 configured to be stationary with respect to the housing 2. The throttle passage 83 extends from the throttle front side 81 to the throttle back side 82. The stationary throttle part 84 can be configured for example as a liner surrounding the throttle bush 80 and fixed with respect to the housing 2.

**[0143]** Basically, the function of a throttle bush is the same as the function of a balance drum. Sometimes the terms "balance drum" and "throttle bush" are considered as synonyms. Within this application the two terms are used mainly for a better linguistic differentiation of the components.

**[0144]** In the fourth embodiment of the pumping arrangement 1 the balance line 9 for recirculating the fluid from the first back side 72 of the first balance drum 7 to a low pressure side of the particular multistage pump 10a, 10b is connected to a location, where the suction pressure of the particular multistage pump 10a or 10b prevails. In both multistage pumps 10a, 10b the balance line 9 is connected to the respective first back side 72. The low pressure side, with which the balance line 9 is in fluid communication, is exposed to the suction pressure of the particular multistage pump 10a or 10b. Thus, in the first multistage pump 10a the balance line 9 is connected to a location, where the suction pressure of the first multistage pump 10a prevails, for example the process side 69 of the second mechanical seal 60. In the second multistage pump 10b the balance line 9 is connected to a location, where the suction pressure of the second multistage pump 10b prevails, for example the process side 69 of the second mechanical seal 60. During regular operation the suction pressure of the second multistage pump 10b at least approximately equals the discharge pressure of the first multistage pump 10a.

**[0145]** The suction pressure is different for the first and the second multistage pump 10a, 10b. However, for each of the multistage pumps 10a, 10b the pressure drop over the first balance drum 7 equals the difference between the intermediate pressure prevailing at the outlet of the crossover line 34 during operation of the multistage pump 10a, 10b and the suction pressure of the particular multistage pump 10a or 10b. Thus, the balancing of the axial thrust in each of the multistage pumps 10a, 10b corresponds to the usual balancing of the axial thrust in a conventional multistage pump with back-to-back arrangement of the impellers 31, 32, 33. This is particularly advantageous for the balancing of the axial thrust.

**[0146]** In both multistage pumps 10a, 10b a second balance line 90 is provided. Referring now in particular to the second multistage pump 10b, the second balance line 90 is provided for recirculating the fluid from the throttle back side 82 of the throttle bush 80 to the process side 69 of the second mechanical seal 60 of the second multistage pump 10b. Since during operation the process side 69 of the second mechanical seal 60 of the second multistage pump 10b is always exposed to the suction pressure of the first multistage pump 10a it is ensured by means of the second balance line 90 that also the process side 59 of the first mechanical seal 50 of the second multistage pump 10b is always exposed to the suction pressure of the first multistage pump 10a during operation of the pumping arrangement 1.

**[0147]** Thus, both the process side 59 of the first mechanical seal 50 and the process side 69 of the second mechanical seal 60 are always exposed to the suction pressure of the first multistage pump 10a.

**[0148]** The pressure prevailing at the second front side 61 of the second balance drum 6 of the second multistage pump 10b is the suction pressure of the second multistage pump 10b, which equals the discharge pressure of the first multistage pump 10a.

**[0149]** The pressure prevailing at the first front side 71 of the first balance drum 7 of the second multistage pump 10b is the intermediate pressure prevailing at the outlet of the crossover line 34 during operation, wherein the intermediate pressure is larger than the suction pressure of the second multistage pump 10b and smaller than the discharge pressure of the second multistage pump 10b.

**[0150]** The pressure prevailing at the first back side 72 of the first balance drum 7, which coincides with the throttle front side 81, is essentially the same as the suction pressure of the second multistage pump 10b. The pressure prevailing at the throttle back side 82 is essentially the same as the suction pressure of the first multistage pump 10a. Therefore, a part of the fluid passes from the throttle front side 81 through the throttle passage 83 to the throttle back side 82 and is then recirculated to the process side 69 of the second mechanical seal 60 of the second multistage pump 10b. Because of the fluid passing through the throttle passage 83 there are additional centering and bearing forces acting on the pump shaft 5, e.g. due to the Lomakin effect.

**[0151]** In particular for subsea applications it is desirable that two multistage pumps 10a, 10b are as similar as possible. Therefore, also the first multistage pump 10a comprises the second balance line 90, the stationary throttle part 84 and the second stationary part 26. However, the throttle 80 and the second balance drum 6 are not provided in the first multistage pump 10a because of the following reason. If there were the throttle 80 and/or the second balance drum 6, there would not be a pressure drop over the throttle 80 or the second balance drum 6, i.e. the pressure prevailing at the throttle front side 81 would be at least essentially the same as the pressure at the throttle back side 82, and the pressure prevailing at the second front side 61 of the second balance drum 6 would be at least essentially the same as the pressure at the second back side 62. If there is no pressure difference or pressure drop over the second balance drum or the throttle bush, respectively, this could be detrimental for the rotor dynamic, e.g. a destabilization of the pump shaft 5 could result.

**[0152]** In the first multistage pump 10a the balance line

9 and the second balance line 90 connect the same rooms or chambers as viewed from the perspective of the pressure, i.e. at the entrance of the balance line 9 the same pressure prevails as at the entrance of the second balance line 90, and at the exit of the balance line 9 the same pressure prevails as at the exit of the second balance line 90.

**[0153]** The advantage of the configuration is that for manufacturing the pumping arrangement 1 two identical multistage pumps can be provided. In one of the multistage pumps, which constitutes the first multistage pump 10a, the throttle 80 and the second balance drum 6 are removed. In the other multistage pump, which constitutes the second multistage pump 10b, the throttle 80 and the second balance drum 6 is maintained or provided, respectively.

**[0154]** For the same reason it is preferred to provide the first multistage pump 10a with a supply opening 23, which is arranged in the housing 2 at the process side 69 of the second mechanical seal 60. The supply opening 23 is provided with a first shut-off valve 301 to open or to close a fluid communication from the process side 69 of the second mechanical seal 60 to the outside of the housing 2.

**[0155]** Furthermore, the pressure equalizing line 20 is provided with a second shut-off valve 302 to open or to close a fluid communication through the equalizing line 20.

**[0156]** In the configuration as it is shown in Fig. 6 the first shut-off valve 301 is closed during operation and the second shut-off valve 302 is open. In case the first multistage pump 10a should be used as the second multistage pump, the supply opening 23 with the first shut-off valve 301 can be used for the equalizing line 20 and the second shut-off valve 302 can be closed.

**[0157]** It has to be noted that the fourth embodiment of the pumping arrangement can also be configured with an inline arrangement of all impellers 31, 32, 33 in an analogous manner as it has been described with respect to the first embodiment (Fig. 3).

**[0158]** Also in the fourth embodiment of the pumping arrangement 1 it is ensured during operation, that the process side 59 of the first mechanical seal 50 as well as the process side 69 of the second mechanical seal 60 are in both multistage pumps 10a and 10b exposed to the suction pressure prevailing at the pump inlet 21 of the first multistage pump 10a.

**[0159]** For the fourth embodiment of the pumping arrangement 1 the housing 2 of the first multistage pump 10a can be configured identical to the housing 2 of the second multistage pump 10b.

**[0160]** In particular, when the pumping arrangement 1 is deployed at a topside or an onshore location, it is also possible to replace the first shut-off valve 301 with a blind flange, which can be removed, if the position of the two multistage pumps shall be exchanged.

**[0161]** Fig. 7 shows a schematic cross-sectional view of a variant of the fourth embodiment of the pumping arrangement 1 according to the invention.

**[0162]** In this variant the first multistage pump 10a is provided with the throttle bush 80 and with the second balance drum 6, however the stationary throttle part 84 and the second stationary part 26 surrounding the second balance drum 6 are removed or not provided. Removing the stationary parts 84 and 26 and maintaining the rotating parts, namely the throttle bush 80 and the second balance drum 6 has the advantage that the balance of the rotor (pump shaft 5 and all components rotating together with the pump shaft 5) is not modified.

**[0163]** Fig. 8 shows a schematic cross-sectional view of a fifth embodiment of a pumping arrangement 1 according to the invention.

**[0164]** In the following description of the fifth embodiment of the pumping arrangement 1 only the differences to the previously described embodiments of the pumping arrangement 1 according to the invention are explained in more detail. The explanations with respect to the previously described embodiments are also valid in the same way or in analogously the same way for the fifth embodiment. Same reference numerals designate the same features that have been explained with reference to the previously described embodiments or functionally equivalent features.

**[0165]** In the fifth embodiment of the pumping arrangement 1 both multistage pumps 10a and 10b are configured with a back-to-back arrangement of the impellers 31, 32, 33 as it has been explained with respect to the third embodiment (Fig. 5). Furthermore, both multistage pumps 10a, 10b are configured with the second mechanical seal 60 for sealing the pump unit 3 at the pump shaft 5 against a leakage of the fluid along the pump shaft 5, as it has been explained with respect to the first embodiment (Fig. 3). Furthermore, both multistage pumps 10a, 10b are provided with the throttle bush 80 surrounded by the stationary throttle part 84, and with the second balance drum 6 surrounded by the second stationary part 26.

**[0166]** It is a particular advantage of the fifth embodiment that the two multistage pumps 10a, 10b are configured to be exchangeable, meaning that each of the two multistage pumps 10a, 10b can be arranged to be the first multistage pump 10a, and each of the two multistage pumps 10a, 10b can be arranged to be the second multistage pump 10b. The position of the two multistage pumps in the pumping arrangement 1 is interchangeable.

**[0167]** As it has been explained with respect to the fourth embodiment of the pumping arrangement (Fig. 6, Fig. 7) in the first multistage pump 10a the throttle 80 and the second balance drum 6 are not provided (Fig. 6), or the stationary throttle part 84 and the second stationary part 26 surrounding the second balance drum 6 are not provided in order to avoid a detrimental effect to the rotor dynamics. In the fifth embodiment of the pumping arrangement 1 an alternative solution is provided to avoid said negative impact on the rotor dynamics.

**[0168]** Reference is now made in particular to the first multistage pump in Fig. 8. In order to ensure that during operation there is always a flow through the throttle passage 83 as well as through the second relief passage 63 of the second balance drum 6 in the first multistage pump 10a, pressurized fluid is supplied both to the throttle passage 83 and to the second relief passage 63.

**[0169]** A first supply opening 86 is provided at the stationary throttle part 84 between the throttle front side 81 and the throttle back side 82 with respect to the axial direction A. Through the first supply opening 86 fluid can be supplied to the throttle passage 83 at a location, which is between the throttle front side 81 and the throttle back side 82 regarding the axial direction A.

**[0170]** In an analogous manner a second supply opening 66 is provided at the second stationary part 26 between the second front side 61 and the second back side 62 with respect to the axial direction A. Through the second supply opening 66 fluid can be supplied to the second relief passage 63 at a location, which is between the second front side 61 and the second back side 62 regarding the axial direction A.

**[0171]** The first supply opening 86 and the second supply opening 66 are each connected to an intermediate take-off 87, which is configured and arranged to provide pressurized fluid having a pressure that is larger than the suction pressure of the first multistage pump 10a and smaller that the discharge pressure of the first multistage pump 10a. As it is shown in Fig. 8 the intermediate take-off 87 is for example arranged at a discharge side of the first stage impeller 31. Of course, it is also possible to arrange the intermediate take-off 87 at a discharge side of any of the intermediate stage impellers 32 or at a discharge side of the last stage impeller 33 or at any other location, where the fluid having a pressure larger than the suction pressure of the first multistage pump 10a can be taken from the pumping arrangement 1.

**[0172]** As it is shown in Fig. 8 the intermediate take-off 87 is connected to a common supply line 88, which branches in a first supply line 85 and a second supply line 65. The first supply line 85 is connected to the first supply opening 86 and the second supply line 65 is connected to the second supply opening 66.

**[0173]** Furthermore, the common supply line 88 is provided with a third shut-off valve 881 to open or to close a fluid communication through the common supply line 88. If the third shut-off valve 881 is in the open position, the fluid can pass through the common supply line 88 and enter the first supply line 85 as well as the second supply line 65. If the third shut-off valve 881 is in the closed position, the fluid cannot enter the first and the second supply line 85, 65.

**[0174]** In addition, the first supply line 85 is provided with a fourth shut-off valve 851 to open or to close a fluid communication through the first supply line 85. If the fourth shut-off valve 851 is in the open position, the fluid can pass through the first supply line 85 to the throttle passage 83. If the fourth shut-off valve 851 is in the closed position, the fluid cannot pass through the first supply line 85 to the throttle passage 83.

**[0175]** The balance line 9 and the second balance line 90 are connected by a connection pipe 91. The connection pipe 91 is provided with a fifth shut-off valve 901 to open or to close a fluid communication through the connection pipe 91.

**[0176]** During operation, in the first multistage pump 10a the third shut-off valve 881, the fourth shut-off valve 851 and the fifth shut-off valve 901 are in the open position and the first shut-off valve 301 is in the closed position.

**[0177]** The fluid at the intermediate take-off 87 has a pressure that equals the discharge pressure of the first stage impeller 31, i.e. a pressure that is larger than the suction pressure of the first multistage pump 10a. The pressurized fluid is guided from the intermediate take-off 87 through the common supply line 88 and enters both the first and the second supply line 85 and 65.

**[0178]** The pressurized fluid passing through the first supply line 85 is supplied through the first supply opening 86 to the throttle passage 83. Since both at the throttle front side 81 and at the throttle back side 82 a pressure prevails that equals the suction pressure of the first multistage pump 10a, the flow entering through the first supply opening 86 is divided into two partial flows. The first partial flow flows to the throttle front side 81 and the second partial flow flows to the throttle back side 82. Thus, according to the representation in Fig. 8 the first partial flow through the throttle passage 83 is directed downwardly in the axial direction A and the second partial flow through the throttle passage 83 is directed upwardly in the axial direction A. Both partial flows through the throttle passage 83 generate a centering effect on the pump shaft 5 due to the Lomakin effect.

**[0179]** The pressurized fluid passing through the second supply line 65 is supplied through the second supply opening 66 to the second relief passage 63. Since both at the second front side 61 and at the second back side 62 a pressure prevails that equals the suction pressure of the first multistage pump 10a, the flow entering through the second supply opening 66 is divided into two partial flows. The first partial flow flows to the second front side 61 and the second partial flow flows to the second back side 62. Thus, according to the representation in Fig. 8 the first partial flow through the second relief passage 63 is directed upwardly in the axial direction A and the second partial flow through the second relief passage 63 is directed downwardly in the axial direction A. Both partial flows through the second relief passage 63 generate a centering effect on the pump shaft 5 due to the Lomakin effect.

**[0180]** As already said, the first multistage pump 10a and the second multistage pump 10b are preferably configured to be exchangeable. Therefore, the first and the second multistage pump 10a and 10b are essentially identically configured. Also the second multistage pump 10b comprises the first and the second supply opening 86, 66, the first and the second supply line 85, 65, the intermediate take-off 87, the common supply line 88, the connection pipe 91, and the third, fourth and fifth shut-off valve 881, 851, 901.

**[0181]** During operation of the pumping arrangement 1 in the second multistage pump the third shut-off valve 881, the fourth shut-off valve 851 and the fifth shut-off valve 901 are each in the closed position. Thus, there is no flow through the first and the second supply line 85, 65 and no flow between the balance line 9 and the second balance line 90 through the connection pipe 91.

**[0182]** It has to be noticed that each of the two multi-stage pumps can be used as the first multistage pump 10a and as the second multistage pump 10b. If the position of the two multistage pumps is exchanged in the pumping arrangement 1, it requires only a change in the adjustment of the shut-off valves 301, 302, 881, 885 and 901.

**[0183]** When the pumping arrangement 1 is deployed at a topside or an onshore location, where the accessibility of the pumping arrangement 1 is not a larger problem, it is also possible without scarifying the interchangeability of the two multistage pumps to remove preferably from the second multistage pump 10b all those components which are not required for the operation of the second multistage pump 10b. For example, the fist supply line 85, the second supply line 65, the common supply line 88 and the connection pipe 91 can be removed from the second multistage pump 10b. The openings such as the first supply opening 86, the intermediate take-off 87 and the second supply opening 66 at the second multistage pump 10b can be closed by means of covers, e.g. with blind flanges.

**[0184]** As an additional feature the fifth embodiment of the pumping arrangement 1 comprises a minimum flow control arrangement to prevent a surging of the first and the second multistage pump 10a, 10b. It has to be noted that also the first, second, third and fourth embodiment of the pumping arrangement 1 can be configured with a minimum flow control arrangement.

**[0185]** Regarding surge control or surge protection respectively, it is known in the art to provide a line through which at least a part of the pumped fluid can be recycled from the pump outlet to the pump inlet. The flow of the fluid at the pump outlet is monitored and when said flow becomes too low or to unstable, at least a part of the fluid is recycled from the pump outlet to the pump inlet.

**[0186]** As it is shown in Fig. 8, the minimum flow control arrangement comprises a first minimum flow line 300 with a first control valve 301 and a second minimum flow line 310 with a second control valve 311. The first control valve 301 is configured to adjust the flow through the first minimum flow line 300 to any value between zero flow and a maximum flow. The second control valve 311 is configured to adjust the flow through the second minimum flow line 310 to any value between zero flow and a maximum flow. Preferably, the control valves 301 and 311 are remotely controlled and adjusted, for example by a surge control unit (not shown) or by a pump control unit (not shown).

**[0187]** By means of the first minimum flow line 300 a fluid communication between the pump outlet 22 of the first multistage pump 10a and the pump inlet 21 of the first multistage pump 10a can be opened, closed and adjusted. For example, the first minimum flow line 300 branches off the connecting line 11 and is connected to the feed line 160 upstream of the additional cooling device 115. Downstream of the branch-off of the first minimum flow line 300 the connecting line 11 is provided with a first check valve 111 configured to prevent a backflow from the second multistage pump 10b into the first minimum flow line 300.

**[0188]** By means of the second minimum flow line 310 a fluid communication between the pump outlet 22 of the second multistage pump 10b and the pump inlet 21 of the second multistage pump 10b can be opened, closed and adjusted. For example, the pump outlet 22 of the second multistage pump 10b is connected to an outlet line 222 for discharging the fluid, e.g. to supply the fluid to the location, where it is injected. The second minimum flow line 310 branches off the outlet line 222 and opens out into the connecting line 11 at a location downstream of the first check valve 111 and upstream of the cooling device 113. Downstream of the branch-off of the second minimum flow line 310 the outlet line 222 is provided with a second check valve 221 configured to prevent a backflow in the outlet line 222.

**[0189]** Fig. 9 shows a schematic cross-sectional view of a sixth embodiment of a pumping arrangement 1 according to the invention.

**[0190]** In the following description of the sixth embodiment of the pumping arrangement 1 only the differences to the previously described embodiments of the pumping arrangement 1 according to the invention are explained in more detail. The explanations with respect to the previously described embodiments are also valid in the same way or in analogously the same way for the sixth embodiment. Same reference numerals designate the same features that have been explained with reference to the previously described embodiments or functionally equivalent features.

**[0191]** The sixth embodiment of the pumping arrangement 1 is similar to the fifth embodiment, however in the sixth embodiment each of the first multistage pump 10a and the second multistage pump 10b is configured with an inline arrangement of all impellers 31, 32, 33. The inline arrangement has been described for example for the first embodiment (Fig. 3). In the sixth embodiment of the pumping arrangement 1, each of the multistage pumps 10a, 10b is configured as a five stage pump comprising the first stage impeller 31, the last stage impeller 32 and three intermediate stage impellers 33. Of course, the number of five stages is only exemplary. In other embodiments each of the multistage pumps 10a, 10b can comprise more than five stages or less than five stages.

**[0192]** Fig. 10 shows a schematic representation of an embodiment of a pumping arrangement 1 according to the invention comprising a by-pass piping, which is designated in its entity with reference numeral 400. It has to be noted that each of the six embodiments of the pumping arrangement 1 previously described may be configured to comprise the by-pass piping 400 as it will be explained with reference to Fig. 10.

**[0193]** The by-pass piping 400 allows for by-passing either the first multi-stage pump 10a, or the second multi-stage pump 10b, or both the fist an the second multistage pump 10a, 10b. The by-pass piping 400 comprises a first by-pass line 401, a second by-pass line 402, a third by-pass line 403 and preferably seven blocking valves 410, 420, 430, 440, 450, 460, wherein the third blocking valve 430 is optionally provided. The three by-pass lines 401, 402, 403 are all connected to a central location 405, where the three by-pass lines 401, 402, 403 merge.

**[0194]** The first by-pass line 401 is connected to the feed line 160 at a location downstream of the seperating device 100 and upstream of the branch-off of the first minimum flow line 300, i.e. upstream of the pump inlet 21 of the first multistage pump 10a. From said location the first by-pass line 401 extends to the central location 405.

**[0195]** The second by-pass line 402 is connected to the outlet line 222, which is connected to the pump outlet 22 of the second multistage pump 10b. Preferably, the second by-pass line 402 is connected to the outlet line 222 downstream of the second check valve 221. From there the second by-pass line 402 extends to the central location 405.

**[0196]** The third by-pass line 403 is connected to the connecting line 11, preferably at a location upstream of the cooling device 113 and downstream of the first check valve 111. From there the third by-pass line 402 extends to the central location 405.

**[0197]** Each of the blocking valves 410, 420, 430, 440, 450, 460, 470 can be switched between an open position, in which the fluid can pass through the blocking valve 410, 420, 430, 440, 450, 460, 470, and a closed position, in which the fluid cannot pass through the blocking valve 410, 420, 430, 440, 450, 460, 470.

**[0198]** The first blocking valve 410 is arranged in the feed line 160 downstream of the location, where the first by-pass line 401 branches off the feed line 160, and upstream of the branch-off of the first minimum flow line 300.

**[0199]** The second blocking valve 420 is arranged in the connecting line 11 downstream of the first check valve 111 and upstream of the branch-off of the third by-pass line 403.

**[0200]** The third blocking valve 430 is arranged in the connecting line 11 downstream of the branch-off of the third by-pass line 403 and upstream of the cooling device 113, preferably upstream of the branch-off of the second minimum flow line 310.

**[0201]** The forth blocking valve 440 is arranged in the outlet line 222 downstream of the second check valve 221 and upstream of the branching-off of the second by-pass line 402.

**[0202]** The fifth blocking valve 450 is located in the second by-pass line 402. The sixth blocking valve 460 is located in the third by-pass line 403. The seventh blocking valve 470 is located in the first by-pass line 401.

**[0203]** In case both multistage pumps 10a and 10b shall operate in series, what is considered as the normal operation, the first blocking valve 410, the second blocking valve 420, the third blocking valve 430 and the forth blocking valve 440 are in the open position. The fifth blocking valve 450, the sixth blocking valve 460 and the seventh blocking valve 470 are in the closed position, so that the first by-pass line 401, the second by-pass line 402 and the third by-pass line 403 are all closed and the fluid cannot pass through the by-pass lines 401, 402, 403.

**[0204]** The bypassing of the first multistage pump 10a and/or the second multistage pump 10b will now be described in each case starting from the normal operation as reference.

**[0205]** In case the first multistage pump 10a shall be bypassed and only the second multistage pump 10b shall be operated, the first blocking valve 410 and the second blocking valve 420 are switched to the closed position. The fifth blocking valve 450 in the second by-pass line 402 remains in the closed position. The third blocking valve 430 and the forth blocking valve 440 remain in the open position. The sixth blocking valve 460 and the seventh blocking valve 470 are switched to the open position. Downstream of the separation device 100 the fluid passes through the feed line 160 and enters the first by-pass line 401. At the central location 405 the fluid enters the third by-pass line 403 and is guided through the third by-pass line 403 to the connecting line 11 and enters the connecting line 11 between the second blocking valve 420 and the third blocking valve 430. The connecting line 11 supplies the fluid to the pump inlet 21 of the second multistage pump 10b.

**[0206]** In case the second multistage pump 10b shall be bypassed and only the first multistage pump 10a shall be operated, the third blocking valve 430 and the fourth blocking valve 440 are switched to the closed position. The seventh blocking valve 470 in the first by-pass line 401 remains in the closed position. The first blocking valve 410 and the second blocking valve 420 remain in the open position. The sixth blocking valve 460 and the fifth blocking valve 450 are switched to the open position. Downstream of the separation device 100 the fluid passes through the feed line 160 to the pump inlet 21 of the first multistage pump 10a and is discharged from the outlet 22 of the first multistage pump 10a to the connecting line 11. Upstream of the closed third blocking valve 430 the fluid enters the third by-pass line 403. At the central location 405 the fluid enters the second by-pass line 402 and is guided through the second by-pass line 402 to the outlet line 222.

**[0207]** In case both the first multistage pump 10a and the second multistage pump 10b shall be bypassed, the first blocking valve 410 and the fourth blocking valve 440 are switched to the closed position. The sixth blocking valve 460 in the third by-pass line 403 remains in the closed position. The second blocking valve 420 and the third blocking valve 430 may remain in the open position or may be switched to the closed position. This does not make any difference, because there is no flow through the connecting line 11. The seventh blocking valve 470 and the fifth blocking valve 450 are switched to the open position. Downstream of the separation device 100 the fluid passes through the feed line 160 and enters the first by-pass line 401. At the central location 405 the fluid enters the second by-pass line 402 and is guided through the second by-pass line 402 to the outlet line 222.

**[0208]** In addition, Fig. 10 illustrates an embodiment of the separation device 100. The embodiment of the separation device 100 can be combined with each of the embodiments of the pumping arrangement 1 previously described. The separation device 100 comprises a gravity separator 110 and an inline cyclone separator 120, also referred to as cyclonic separator 120. In the gravity separator 110 the gas-liquid separation is driven by gravity. The liquid phase settles down at the bottom of a vessel and the gaseous phase is enriched at the upper part of the vessel. In the cyclone separator 120 the separation is mainly caused by centrifugal forces. The fluid fed to the cyclone separator 120 is set into rotation to cause the separation. Such inline cyclone separators are for example disclosed in WO 2017/137080, WO 2014/060048 or WO 02/056999.

**[0209]** At a subsea location, for example, a stream of crude oil emerging from a production well (not shown) of a subterranean oil field is fed to the gravity separator 110 of the separation device 100 as indicated by the arrow with the reference numeral W. The gravity separator 110 separates the crude oil in a heavier, liquid enriched phase having a higher density and a lighter, predominantly gaseous phase having a lower density. The lighter phase is enriched with methane and carbon dioxide and the heavier phase comprises predominantly liquid hydrocarbons. The lighter phase is discharged from the gravity separator 110 to the feed line 160 leading to the pump inlet 21 of the first multistage pump 10a. In the feed line 160 the inline cyclone separator 120 is arranged. The cyclone separator 120 at least considerably reduces the amount of droplets of liquid in the lighter phase. Downstream of the cyclone separator 120 the lighter phase is guided by the feed line 160 to the first multistage pump 10a. The liquid, that is separated from the lighter phase in the cyclone separator 120 is discharged to a liquid line 130, which is connected to the delivery pipe 150, to which the heavier phase is discharged from the gravity separator 110. Thus, the liquid separated in the cyclone separator 120 is merged into the heavier phase in the delivery pipe 150. A delivery pump 170 is provided to convey the heavier phase through the delivery pipe 150 for example to a topside location such as a FPSO. The delivery pump 170 is for example configured as a centrifugal single phase pump for conveying a liquid.

**[0210]** A delivery control valve 180 can be provided in the delivery pipe 150 to control or to adjust the flow of the heavier phase discharged from the gravity separator 110. Preferably, the delivery control valve 180 is arranged upstream of the location, where the liquid line 130 ends in the delivery pipe 150.

**[0211]** According to a preferred application, the pumping arrangement 1 according to the invention is configured for an installation on a sea ground.

**[0212]** The pumping arrangement 1 is in particular advantageous for applications, where a high injection pressure is required. For such applications it may be more efficient to arrange two or more multistage pumps 10a, 10b in series rather than adding additional stages to a single multistage pump.

**[0213]** Furthermore, the pumping arrangement 1 according to the invention may also comprise more than two multistage pumps 10a, 10b. Preferably, all multistage pumps of the pumping arrangement 1 are arranged in series.

## Claims

**1.** A pumping arrangement comprising a first multistage pump (10a) and a second multistage pump (10b), wherein each multistage pump (10a, 10b) comprises:

a housing (2) with a pump unit (3) arranged in the housing (2), wherein the housing (2) comprises a pump inlet (21) for receiving a fluid with a suction pressure, and a pump outlet (22) for discharging the fluid with a discharge pressure, and wherein the pump unit (3) comprises a plurality of impellers (31, 32, 33) for conveying a fluid from the pump inlet (21) to the pump outlet (22), and

a pump shaft (5) for rotating about an axial direction (A), with the pump shaft (5) extending from a drive end (51) to a non-drive end (52), wherein each impeller (31, 32, 33) is mounted to the pump shaft (5) in a torque proof manner,

wherein the second multistage pump (10b) comprises at least a first mechanical seal (50) for sealing the pump unit (3) at the pump shaft (5), with the first mechanical seal (50) having a process side (59) facing the pump unit (3),

and wherein the pump outlet (22) of the first multistage pump (10a) is connected to the pump inlet (21) of the second multistage pump (10b), so that the first multistage pump (10a) and the second multistage pump (10b) are arranged in series,

**characterized in that** the process side (59) of the first mechanical seal of the second multiphase pump (10b) is in fluid communication with the pump inlet (21) of the first multistage pump (10a).

2. A pumping arrangement in accordance with claim 1, wherein each multistage pump (10a, 10b) comprises a first balance drum (7), which is fixedly connected to the pump shaft (5) between the pump unit (3) and the drive end (51) of the pump shaft (5), the first balance drum (7) defining a first front side (71) facing the pump unit (3) and a first back side (72), wherein a first relief passage (73) is provided between the first balance drum (7) and a first stationary part (27) configured to be stationary with respect to the housing (2), the first relief passage (73) extending from the first front side (71) to the first back side (72), and wherein a balance line (9) is provided and configured for the recirculation of the fluid from the back side (72) to a low pressure side of the multistage pump (1).

3. A pumping arrangement in accordance with anyone of the preceding claims, wherein the second multistage pump (10b) comprises a second balance drum (6), which is fixedly connected to the pump shaft (5) between the pump unit (3) and the non-drive end (52) of the pump shaft (5), the second balance drum (6) defining a second front side (61) facing the pump unit (3) and a second backside (62), wherein a second relief passage (63) is provided between the second balance drum (6) and a second stationary part (26) configured to be stationary with respect to the housing (2), the second relief passage (63) extending from the second front side (61) to the second back side (62), and wherein the second back side (62) is in fluid communication with the pump inlet (21) of the first multistage pump (10a).

4. A pumping arrangement in accordance with claim 2 and claim 3, wherein the balance line (9) of the second multistage pump (10b) is configured to connect the first back side (72) of the second multistage pump (10b) with the second back side (62) of the second multistage pump (10b).

5. A pumping arrangement in accordance with anyone of claims 3-4, wherein the second front side (61) is exposed to the suction pressure of the second multistage pump (10b).

6. A pumping arrangement in accordance with anyone of claims 3-5, wherein the second multistage pump (10b) comprises a second mechanical seal (60) for sealing the pump unit (3) at the pump shaft (5), with the second mechanical seal (60) having a process side (69) facing the pump unit (3), and wherein the second mechanical seal (60) is arranged between the second balance drum (6) and the non-drive end (52) of the pump shaft (5) of the second mechanical seal (60).

7. A pumping arrangement in accordance with anyone of the preceding claims, wherein in each multistage pump (10a, 10b) the plurality of impellers (31, 32, 33) comprises a first set of impellers (31, 33) and a second set of impellers (33, 32) wherein the first set of impellers (31, 33) and the second set of impellers (33,32) are arranged in a back-to-back arrangement, so that an axial thrust generated by the first set of impellers (31, 32) is directed opposite to an axial thrust generated by the second set of impellers (33, 32).

8. A pumping arrangement in accordance with claim 7, wherein each multistage pump comprises a center bush (35), which is fixedly connected to the pump shaft (5) between the first set of impellers (31, 33) and the second set of impellers (33, 32), wherein a balancing passage (37) is provided between the center bush (35) and a third stationary part (36) configured to be stationary with respect to the housing (2).

9. A pumping arrangement in accordance with anyone of the preceding claims, wherein each multistage pump (10a, 10b) comprises a drive unit (4) arranged in the housing (2), wherein the drive unit (4) comprises a drive shaft (42) for driving the pump shaft (5), and an electric motor (41) for rotating the drive shaft (42) about the axial direction (A), and wherein a coupling (8) is provided for coupling the drive shaft (42) to the pump shaft.

10. A pumping arrangement in accordance with claim 9, wherein in each multistage pump (10a, 10b) the first mechanical seal (50) is arranged between the first balance drum (7) and the drive unit (4).

11. A pumping arrangement in accordance with anyone of claims 9-10, wherein each multistage pump (10a, 10b) is configured as a vertical pump with the pump shaft (5) extending in the direction of gravity, and wherein the drive unit (4) is arranged on top of the pump unit (3).

12. A pumping arrangement in accordance with anyone of the preceding claims, wherein each multistage pump (10a, 10b) comprises a throttle bush (80), which is fixedly connected to the pump shaft (5) between the pump unit (3) and the drive end (51) of the pump shaft (51), the throttle bush (80) defining a throttle front side (81) facing the pump unit (3) and a throttle back side (82), wherein a throttle passage (83) is provided between the throttle bush (80) and a stationary throttle part (84) configured to be stationary with respect to the housing (2), the throttle passage (83) extending from the throttle front side (81) to the throttle back side (82), wherein a first supply opening (86) is provided at the stationary throttle part (84) to supply fluid to the throttle passage (83)

at a location between the throttle front side (81) and the throttle back side (82) with respect to the axial direction (A), wherein the first supply opening (86) is connectable with an intermediate take-off (86), which is arranged for delivering the fluid with a pressure higher than the suction pressure of the particular multistage pump (10a; 10b), and wherein a second balance line (90) is provided and configured for the recirculation of the fluid from the throttle back side (82) to a low pressure side.

**13.** A pumping arrangement in accordance with claim 12, wherein a second supply opening (66) is provided at the second stationary part (26) to supply fluid to the second relief passage (63) at a location between the second front side (61) and the second back side (62) with respect to the axial direction (A), wherein the second supply opening (66) is connectable with the intermediate take-off (86).

**14.** A pumping arrangement in accordance with anyone of the preceding claims, configured for installation on a sea ground.

**15.** A pumping arrangement in accordance with anyone of the preceding claims, configured for injecting a compressible fluid being in the dense state at the pump outlet (22) of the second multistage pump (10b) into a subterranean region.

Fig.1

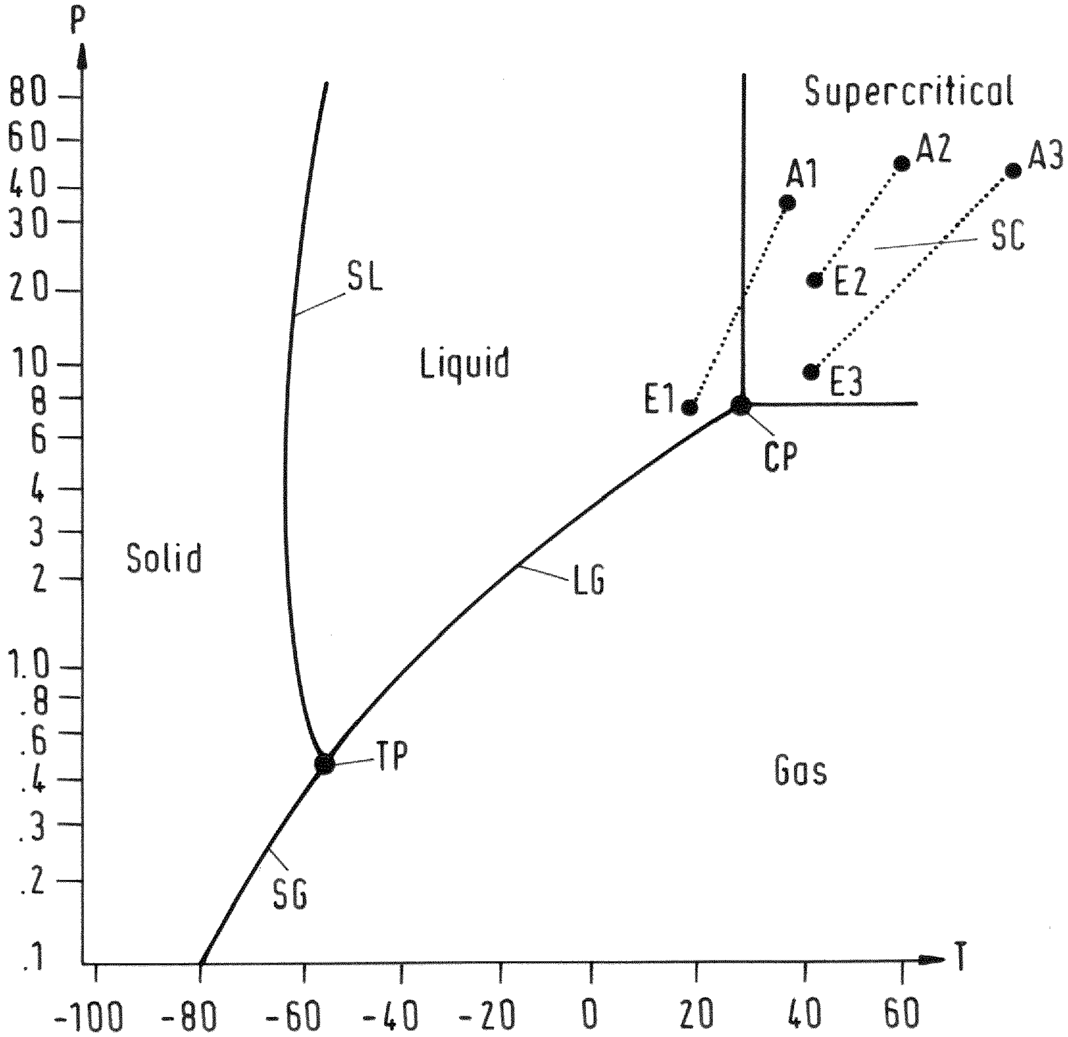
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Fig.2

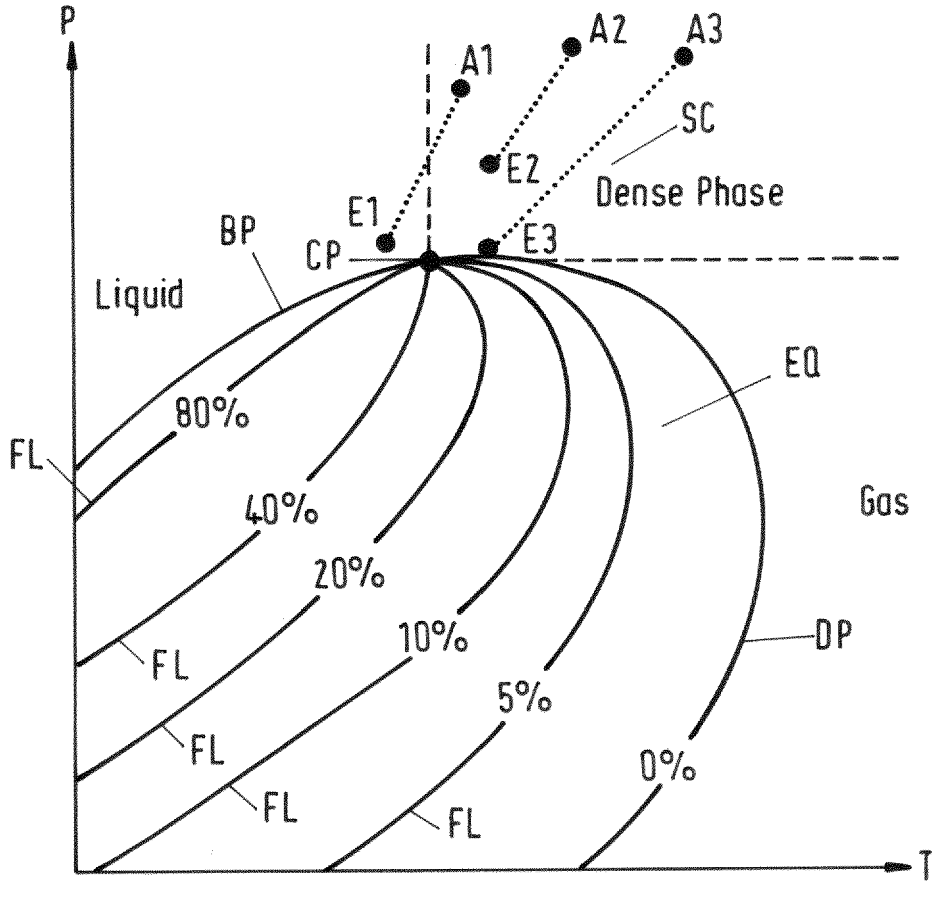
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Fig.3

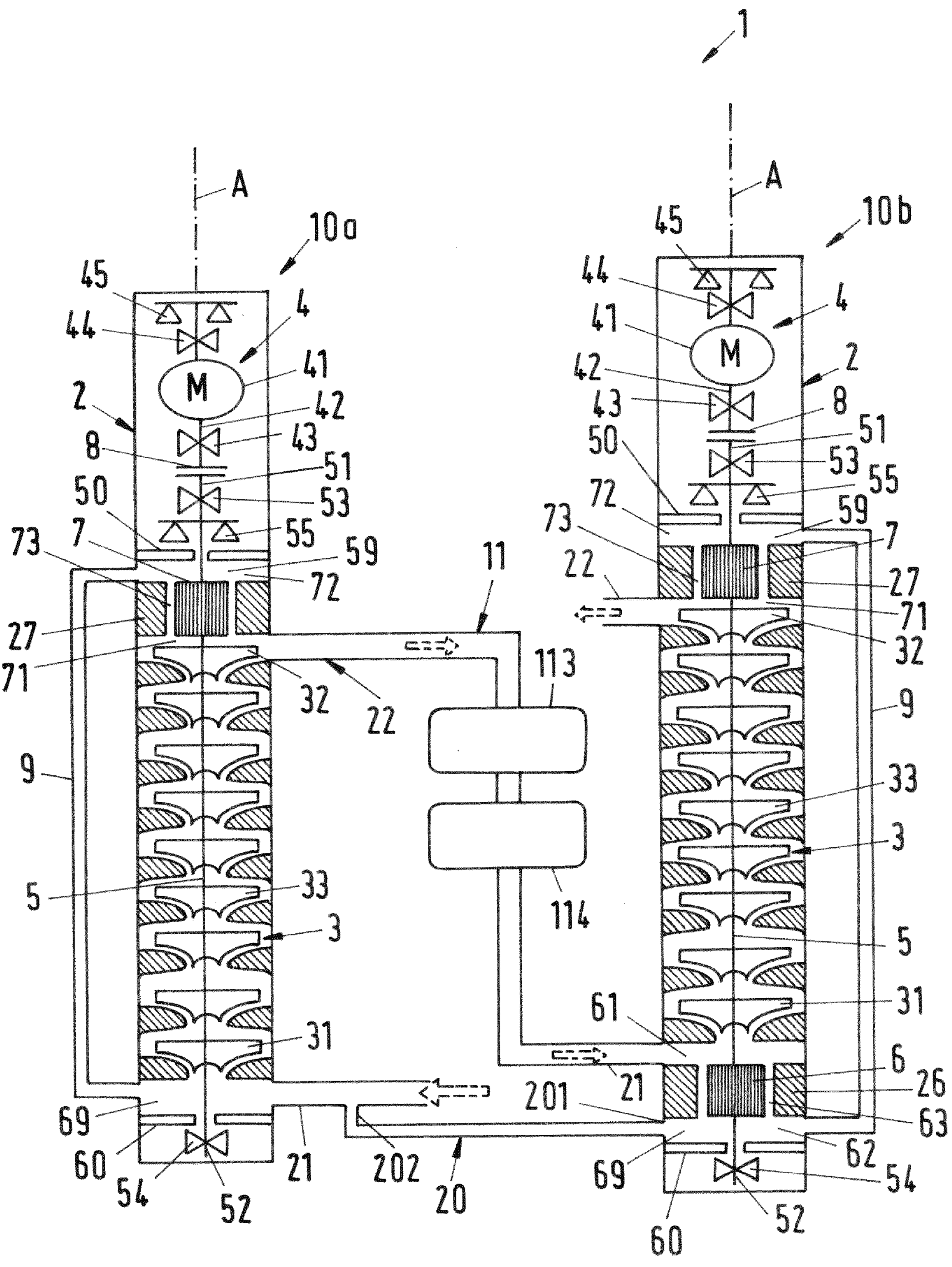
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Fig.4

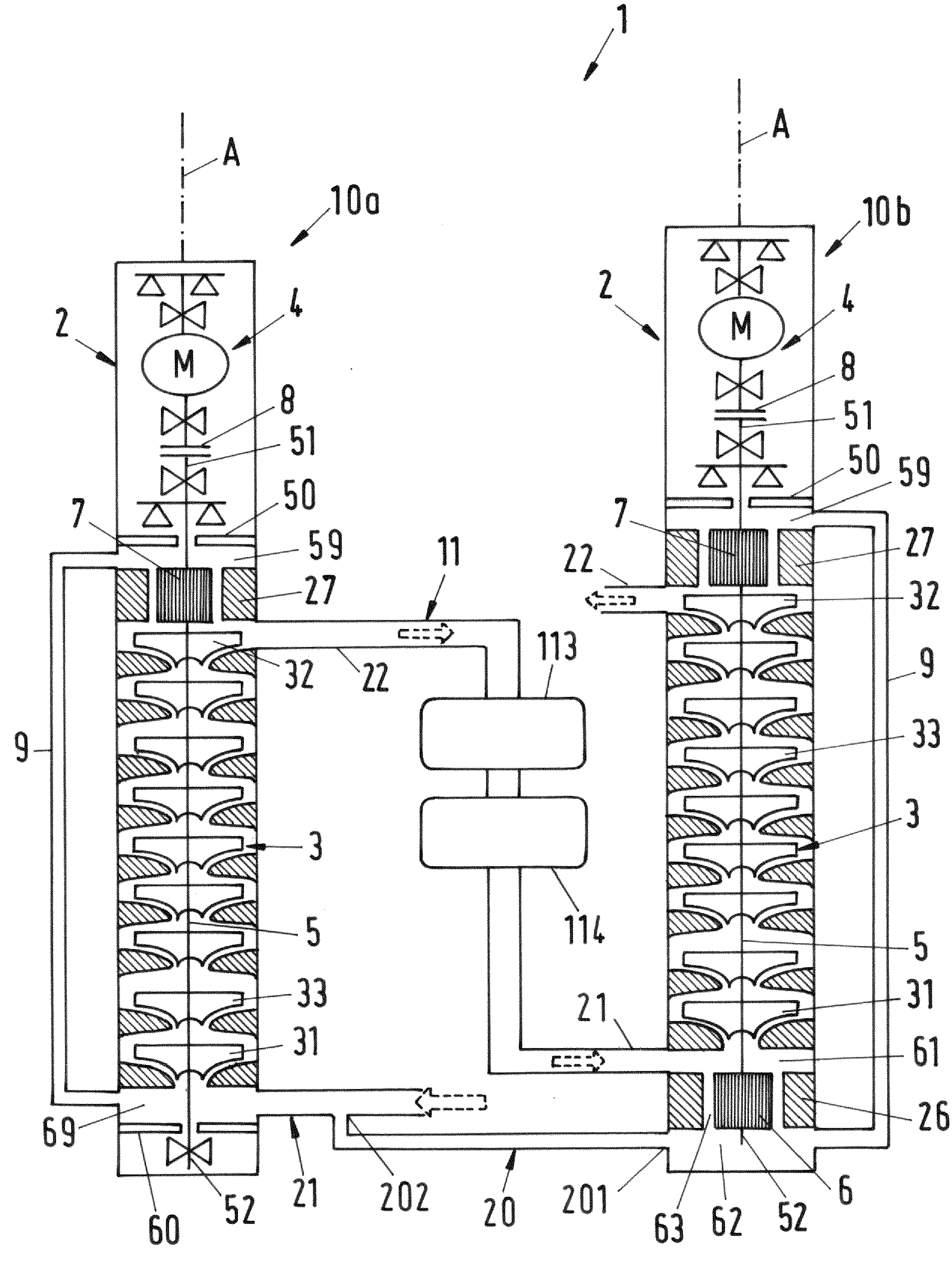
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Fig.5

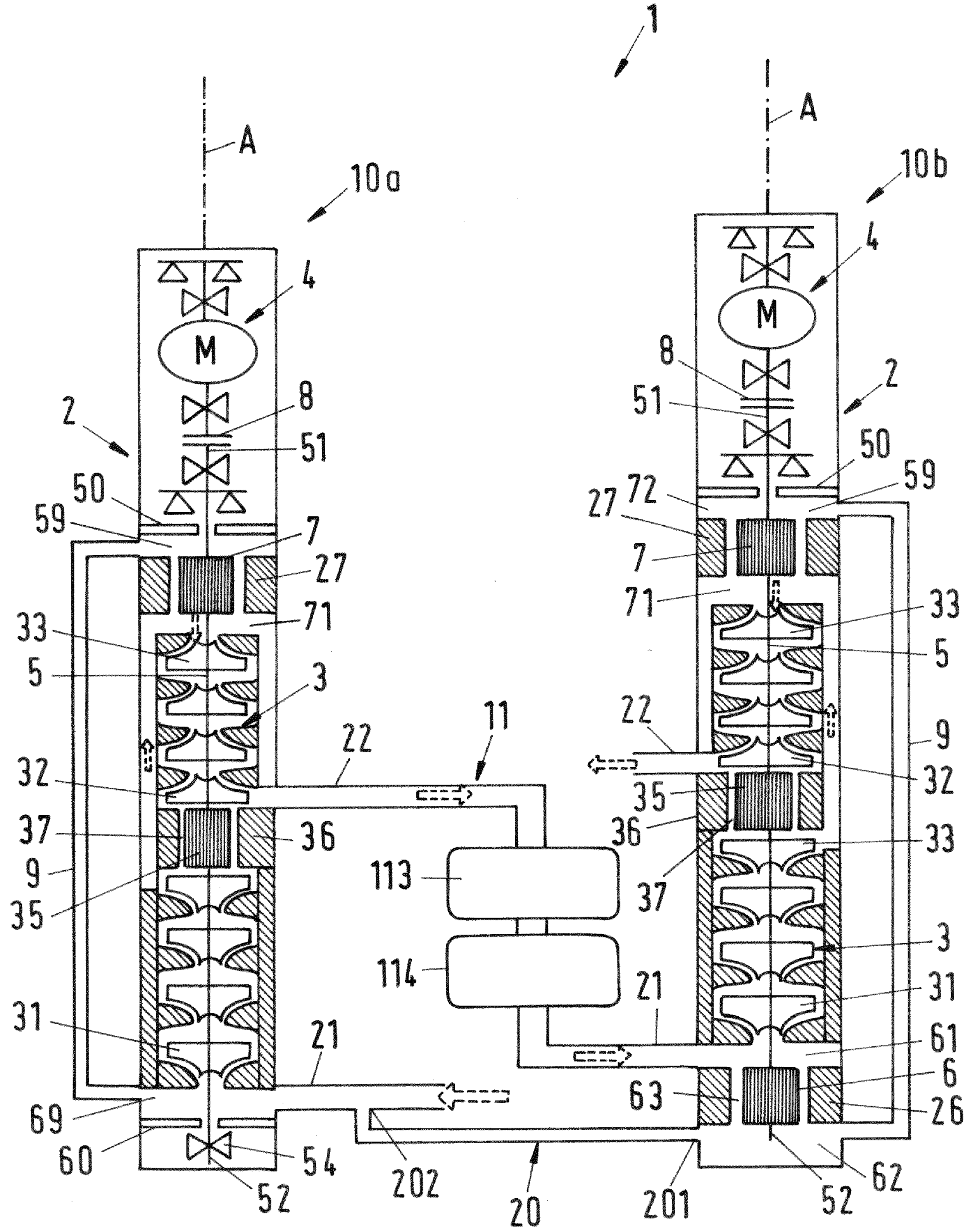
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Fig.6

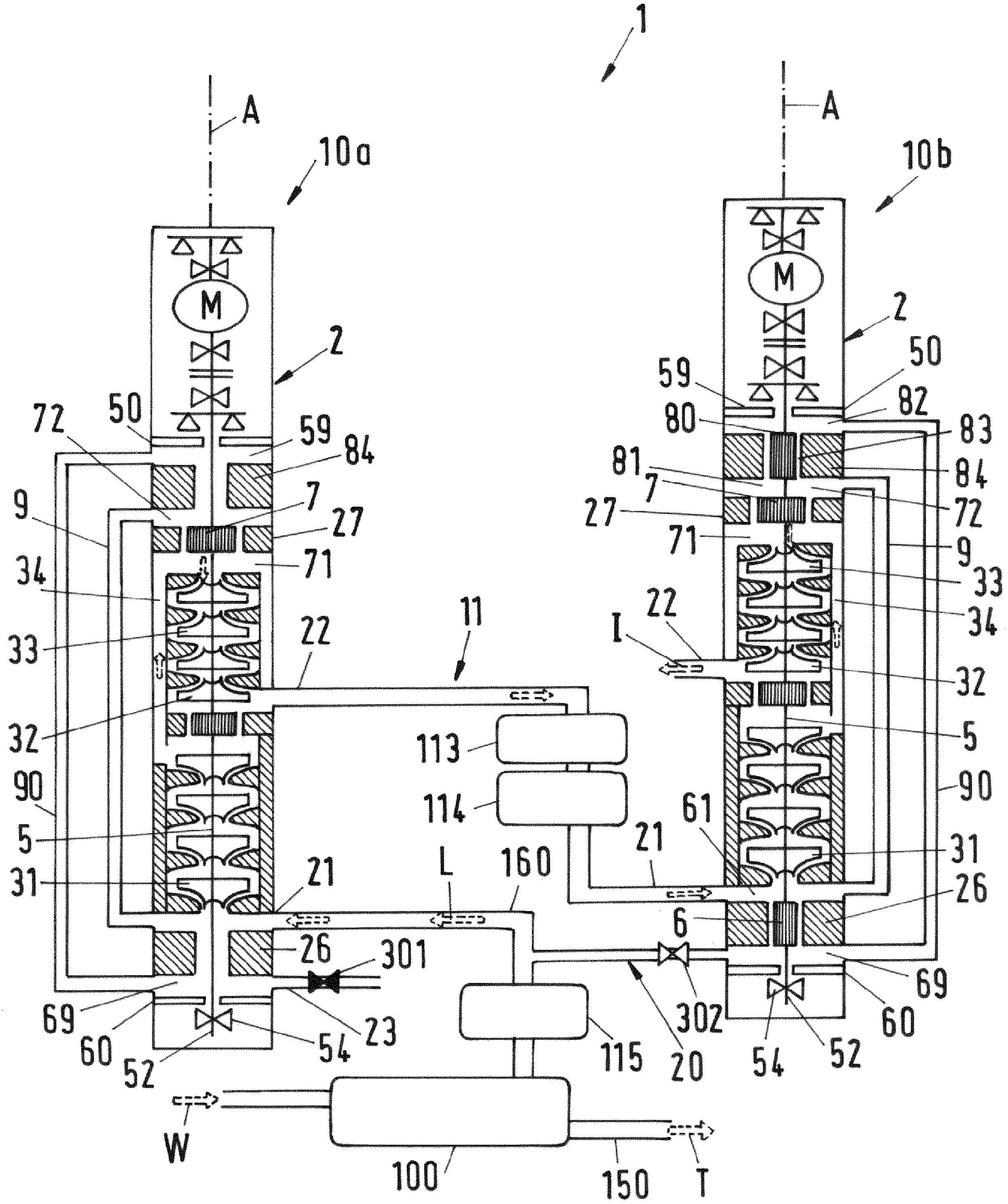
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Fig.7

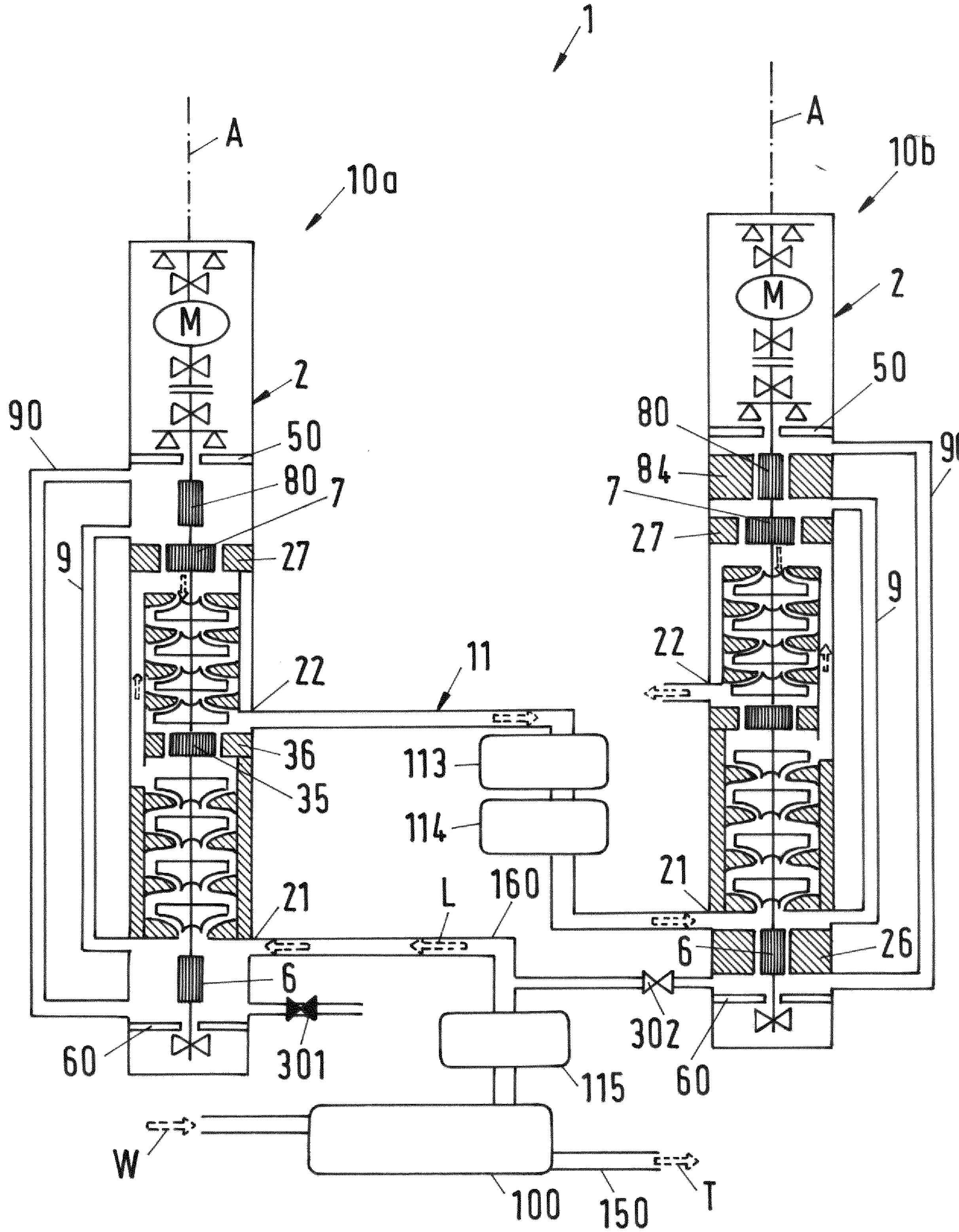
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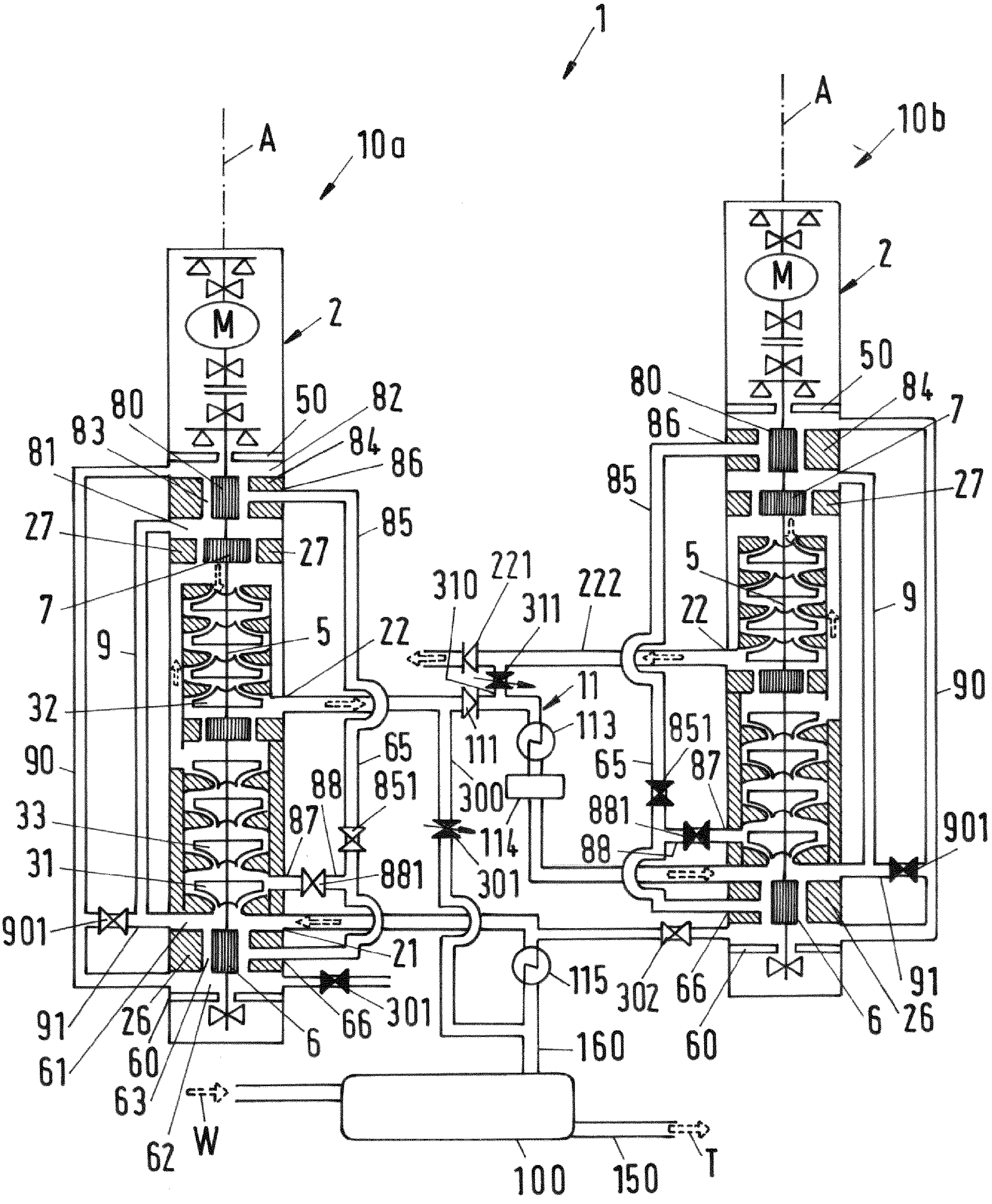
Fig.8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21
66
301
6
310
221
311
222
111
11
113
300
114
301
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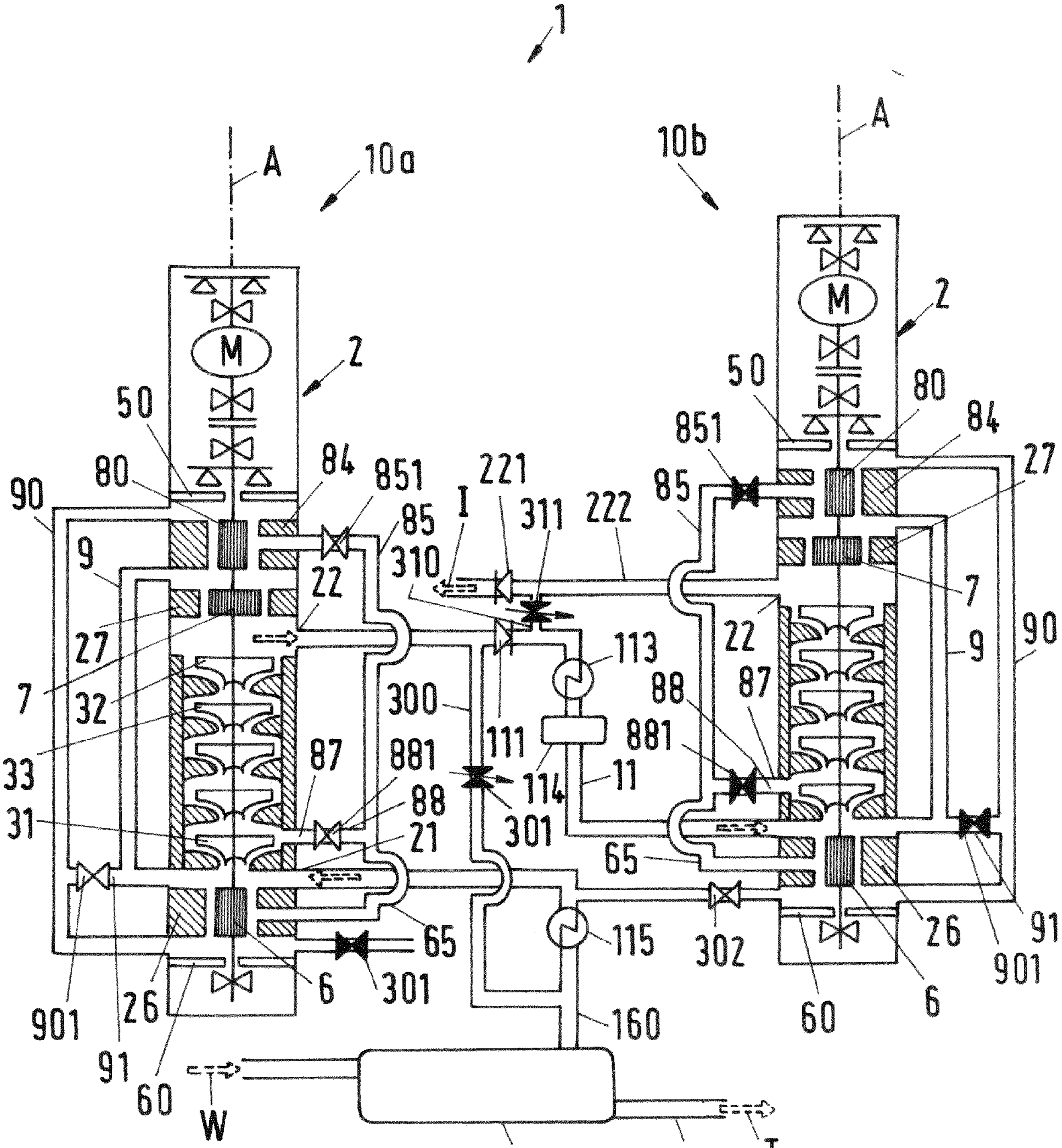
Fig.9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27
90
9
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310
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27
7
32
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31
22
113
300
111
114
11
88
87
881
881
87
88
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301
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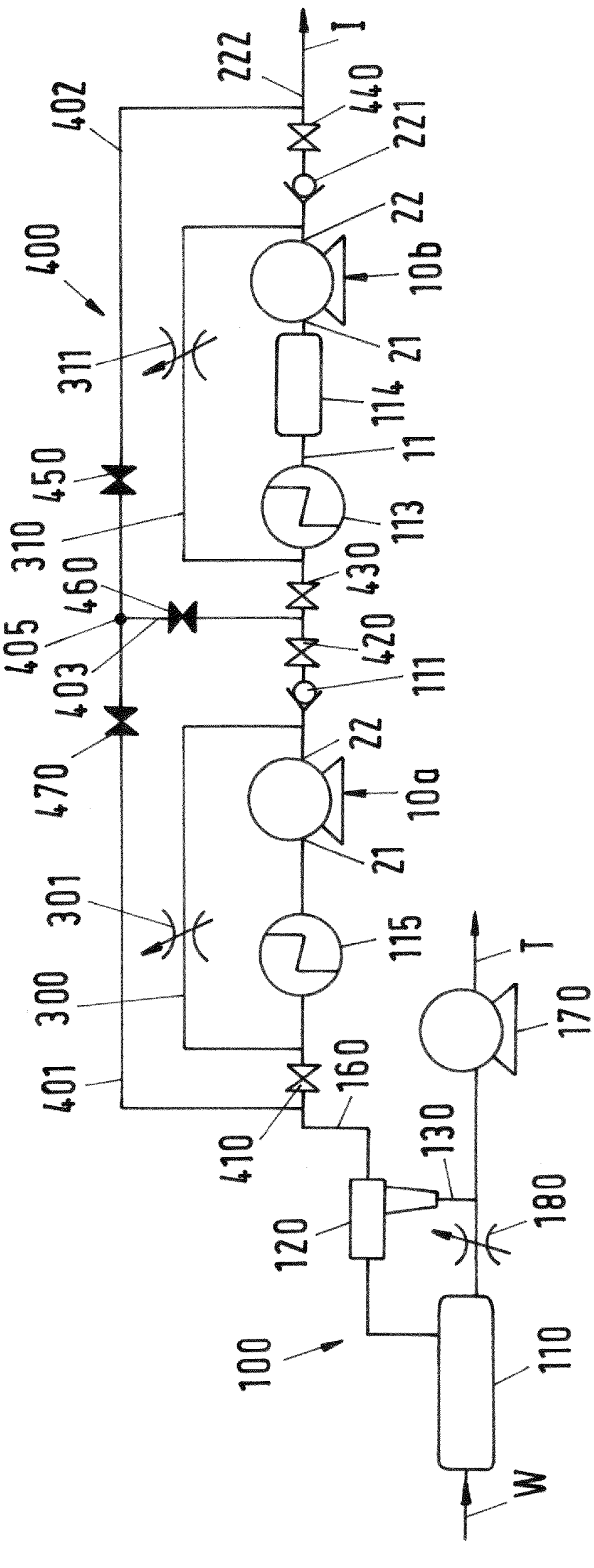
Fig.10



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W
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# EUROPEAN SEARCH REPORT

Application Number
EP 23 16 5190

## 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 | EP 3 771 828 A1 (SULZER MANAGEMENT AG [CH]) 3 February 2021 (2021-02-03)<br>* paragraph [0053] *<br>* paragraph [0125] - paragraph [0127] *<br>* figures 1, 3, 4 *<br>----- | 1-15 | INV.<br>F04D17/12<br>F04D25/06<br>F04D25/16<br>F04D29/051<br>F04D29/12 |
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| Place of search | Date of completion of the search | Examiner |
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| The Hague | 16 August 2023 | Oliveira, Damien |

CATEGORY OF CITED DOCUMENTS

X : particularly relevant if taken alone
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## ANNEX TO THE EUROPEAN SEARCH REPORT ON EUROPEAN PATENT APPLICATION NO.

EP 23 16 5190

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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