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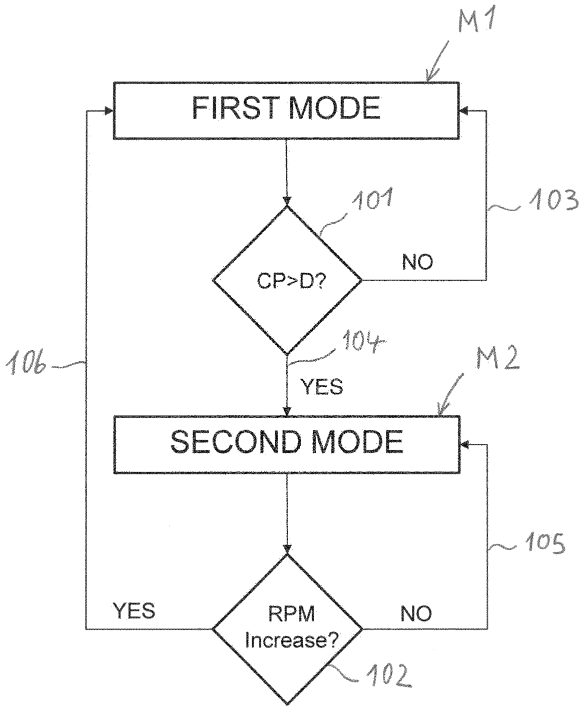
(54) **MULTISTAGE PUMP SYSTEM AND PUMPING ARRANGEMENT**

(57) A multistage pump system for conveying a fluid is proposed, comprising a multistage pump (1) and a control unit (20),
wherein the multistage pump comprises a pump unit (3) arranged in a pump housing (30), and a drive unit (4) arranged in a drive housing (40),
wherein the control unit (20) is configured for controlling the drive unit (4),
wherein the pump housing (30) comprises a pump inlet (21) for receiving the fluid with a suction pressure, and a pump outlet (22) for discharging the fluid with a discharge pressure,
wherein the pump unit (3) comprises a pump shaft (5) for rotating about an axial direction (A), and a plurality of impellers (31, 32, 33) for conveying the fluid from the pump inlet (21) to the pump outlet (32), with each impeller (31, 32, 33) being mounted to the pump shaft (5) in a torque proof manner,
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 the control unit (20) is configured to drive the electric motor (41) at different rotational speeds.

The control unit (20) is configured to operate the electric motor (41) in a first mode (M1), in which the control unit (20) operates the electric motor (41) at a constant rotational speed, and in a second mode (M2), in which the control unit (20) operates the electric motor (41) at a constant engine load, wherein the control unit (20) is further configured to switch from the first mode (M1) to the second mode (M2), when a control parameter (CP) that is indicative of a rate of change of the engine load exceeds a presettable threshold value (D). Furthermore a pumping arrangement is proposed.

Fig. 4

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## Description

[0001] The invention relates to a multistage pump system for conveying a fluid and to a pumping arrangement comprising a first and a second multistage pumps according to the preamble of the independent claim of the respective category.

[0002] Multistage pump systems for conveying a fluid are used in many different industries, in particular for applications where a high pressure shall be generated. A multistage pump system comprises, e.g. a multistage pump and a control unit. The 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 process fluid, in most cases 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 pumps 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 pumps is challenging, in particular because these pumps 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 pump.

[0004] In particular in deep-water 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 available 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 used 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 fluids.

[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 which is higher than 200 kg/m$^3$, e.g. approximately 400 kg/m$^3$ or even up to 500 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. Thus, there is a need for a subsea pump that can reinject such a compressible fluid in a subterranean region.

**[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.

**[0019]** It is a known technology to separate the dense gas, which is e.g. a mixture of $CO_2$, $CH_4$ and light hydrocarbons from the multiphase fluid with a separator device, e.g. a subsea separator. The separator separates the multiphase fluid into a heavier liquid enriched phase, which is delivered to a topside location, and into the lighter $CO_2$ and $CH_4$ enriched phase, which is supplied to a multistage pump for reinjection into a subterranean region. Because of instabilities in the separator a carry-over of liquid into the lighter phase can occur. This carry-over of liquid can result in a liquid slug, e.g. in a water slug, propagating into the multistage pump.

**[0020]** The dense gas of the lighter phase leaving the separator has for example a density of around 400kg/m3. The liquid slugs such as water slugs can have densities of 1000kg/m$^3$. The multistage pump for conveying the lighter phase, here the dense gas, is usually operated with an electric motor. The multistage pump system also comprises the control unit for operating the electric motor. The control unit can comprise or can be configured, for example, as a variable frequency drive (VFD), which is configured to operate the drive unit - and therewith the pump unit - at a given and fixed frequency, i.e. with a constant rotational speed. For subsea applications the control unit can be deployed at the sea ground, for example near the multistage pump. Alternatively, the control unit can be located at a topside location, e.g. on a FPSO unit, or on a platform, or in an onshore facility. Usually, the control unit is connected to the multistage pump or the drive unit, respectively, by a power cable both for a subsea arrangement and for a topside location of the control unit.

**[0021]** When a liquid slug reaches and propagates into the multistage pump, the VFD keeps on controlling the pump shaft with the given rotational speed, i.e. the VFD maintains the frequency of the rotation. Because of the considerably higher density of the liquid forming the liquid slug, a significant spike in the power required to drive the pump shaft will occur. This significant spike occurs in the drive current for driving the electric motor as well as in the torque required for rotating the pump shaft. Furthermore, the torque or current spike will also result in a considerable pressure spike at the pump outlet of the multistage pump. At a constant rotational speed of the pump shaft the pressure rise at the pump outlet caused by the liquid slug is at least approximately proportional to the density increase in the fluid caused by the liquid slug.

**[0022]** These current or torque spikes are very demanding for the multistage pump. In particular at higher rotational speeds, for example at 6000 rpm, the torque or current spikes, respectively, may be as strong that they are outside the allowed operational range of the motor of the multistage pump. Therefore, the liquid slugs may result in a shutdown of the multistage pump. This is a detrimental effect requiring a solution.

**[0023]** Another problem caused by a liquid slug occurs, when two or more multistage pumps are connected in series for conveying the lighter phase. Such a pumping arrangement is disclosed for example in EP 3 771 828 A1. For many applications, in particular regarding the injection of a compressible fluid in a subterranean region, a significant pressure rise is required. Therefore, the pumping arrangement comprising two multistage pumps arranged in series is sometimes preferred for such applications rather than a single multistage pump.

**[0024]** Operating two multistage pumps in series has proven to be a good concept, however the aforementioned liquid slugs propagating through the first multistage pump are also detrimental for the second multistage pump, because 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.

**[0025]** In case there is a liquid slug passing through the first multistage pump, the first multistage pump will generate a considerable pressure spike in the discharge pressure at its pump outlet. This results in a pressure spike at the pump inlet of the second multistage pump. The pressure spike at the pump inlet of the second multistage pump may result in a reverse pressure across the mechanical seals, which seal the pump shaft of the second multistage pump. A reverse pressure means that the pressure prevailing at the process side of the respective mechanical seal is considerably larger than the pressure prevailing at the other (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.

**[0026]** 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.

**[0027]** Starting from this state of the art, it is therefore an object of the invention to propose a multistage pump system with a multistage pump for conveying a fluid, which is better protected against the negative impacts of liquid slugs and more generally of a sudden increase in the density of the fluid occurring at the pump inlet of the pump. The multistage pump shall also be suited for subsea applications and for deployment on the sea ground. In particular, the multistage pump 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 pumping arrangement comprising such a multistage pump system.

**[0028]** The subject matter of the invention satisfying these objects is characterized by the features of the respective independent claim.

**[0029]** Thus, according to the invention, a multistage pump system for conveying a fluid is proposed, comprising a multistage pump and a control unit,

wherein the multistage pump comprises a pump unit arranged in a pump housing, a drive unit arranged in a drive housing,
wherein the control unit is configured for controlling the drive unit,
wherein the pump housing comprises a pump inlet for receiving the fluid with a suction pressure, and a pump outlet for discharging the fluid with a discharge pressure,
wherein the pump unit comprises a pump shaft for rotating about an axial direction, and a plurality of impellers for conveying the fluid from the pump inlet to the pump outlet, with each impeller being mounted to the pump shaft in a torque proof manner,
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 the control unit is configured to drive the electric motor at different rotational speeds. The control unit is configured to operate the electric motor in a first mode, in which the control unit operates the electric motor at a constant rotational speed, and in a second mode, in which the control unit operates the electric motor at a constant engine load, wherein the control unit is further configured to switch from the first mode to the second mode, when a control parameter that is indicative of a rate of change of the engine load exceeds a presettable first threshold value.

**[0030]** The multistage pump comprises the pump unit and the drive unit for driving the pump shaft with the plurality of impellers for rotating about the axial direction. The control unit for the electric motor of the drive unit is configured to drive the electric motor at different rotational speeds so that the pump shaft may rotate with different rotation frequencies. For example, the control unit for the drive unit may be configured as a variable frequency drive (VFD) as it is known in the art. According to the invention, the control unit is configured to operate the electric motor in two different modes, namely in a first mode, where the electric motor is controlled to rotate at a constant rotational speed, and in a second mode, where the electric motor is controlled to operate at a constant engine load, e.g. a constant torque. A control parameter, which is indicative of the rate of a change of the engine load, is observed. If the control parameter exceeds a presettable threshold value the control unit switches from the first mode to the second mode.

**[0031]** In regular operation the control unit operates the electric motor in the first mode, i.e. the electric motor is controlled to rotate with a constant frequency. During operation of the multistage pump the control parameter is observed. When a sudden increase in the density of the fluid occurs at the pump inlet of the pump, for example because of a liquid slug, the control unit initially controls the electric motor to maintain the given rotational speed. Due to the increase in the density of the fluid this requires a considerably larger engine load, e.g. a considerably larger torque. As a consequence the control parameter immediately increase strongly and passes the threshold value. This causes the control unit to change from the first mode to the second mode. Now, the electric motor is controlled to operate at a constant engine load, which in turn causes the rotational speed of the electric motor and therewith the rotational speed of the pump shaft to decrease. Due to the decrease in the rotational speed of the pump shaft and the impellers, there will be no remarkable pressure spike at the pump outlet of the multistage pump.

**[0032]** Thus, by switching from the first mode (constant rotational speed) to the second mode (constant engine load) the occurrence of a pressure spike at the pump outlet is reliably prevented. Furthermore, since the electric motor is operated in the second mode at a constant engine load, it is reliably prevented, that a liquid slug or a similar event occurring at the pump inlet results in strong torque or current spikes, respectively. Thus, there is no risk that a liquid slug causes such a strong increase in the torque or the current that the allowed operational range of the multistage pump is left. A tripping of the multistage pump due to a liquid slug is reliably avoided.

**[0033]** Furthermore, the liquid slug at the pump inlet of the multistage pump does no longer cause a pressure spike in the discharge pressure at the pump outlet of the multistage pump. This is in particular advantageous in view of a pumping arrangement where the pump inlet of a second multistage pump is connected to the pump outlet of the multistage pump. Due to the avoidance of remarkable pressure spikes in the discharge pressure of the multistage pump there are also no remarkable pressure spikes in the suction pressure of the second multistage pump. Thus, the aforementioned problems regarding the mechanical seals of the second multistage pump can be solved by the multistage pump according to the invention when used as the first multistage pump in a pumping arrangement comprising a first and a second multistage pump connected in series.

**[0034]** Preferably, the control unit is configured to switch from the second mode to the first mode, when the rotational speed increases during operation in the second mode. Thus, it is preferred that the control unit switches back from the second mode to the first mode, as soon as the liquid slug or a similar event has passed the multistage pump and left through the pump outlet. When the liquid slug left the multistage pump during operation in the second mode, the pump shaft with the impellers will accelerate, because the electric motor is operated with a constant engine load and the density of the fluid acted on by the impellers decreases. Since the density of the fluid decreases a constant engine load of the electric motor will result in an increase of the rotational speed of the pump shaft and the impellers. The rotational speed will increase towards the value it had prior to the switching from the first to the second mode. Upon detection of said increase in the rotational speed the control unit will switch back to the first mode and operate the electric motor at the desired constant rotational speed.

**[0035]** In a preferred embodiment the control parameter indicates the temporal change of a drive current that is supplied to the electric motor for driving the electric motor. Said drive current is directly proportional to the torque generated by the electric motor and the torque is a measure for the engine load at which the electric motor is operated. Preferably the control parameter is the first time derivative of the drive current or proportional to the first time derivative of the drive current.

**[0036]** According to a preferred configuration the multistage pump is configured for conveying a compressible fluid being in the dense phase at the pump outlet.

**[0037]** 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.

**[0038]** 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.

**[0039]** 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.

**[0040]** Preferably the multistage pump is configured as an injection pump for injecting the compressible fluid being in the dense state 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 or the multiphase fluid 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 multistage pump and reinjected into a subterranean region, e.g. the oil reservoir.

**[0041]** 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.

**[0042]** In particular for subsea applications where the multistage pump is deployed below the water surface, it is preferred that the pump housing and the motor housing form a common housing, in which the pump unit and the drive unit are arranged. The common housing with the drive unit and the pump 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.

**[0043]** According to a preferred embodiment, the multistage pump is configured as a vertical pump with the pump shaft extending in the direction of gravity, and herein the drive unit is arranged on top of the pump unit.

**[0044]** Furthermore, it is a preferred design that the multistage pump is configured as a single phase pump for conveying a single phase fluid.

**[0045]** 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.

**[0046]** 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%.

**[0047]** The primary purpose of e single-phase pump is to pump a single phase fluid.

**[0048]** According to a preferred embodiment at least the multistage pump is configured for installation on a sea ground. The control unit can be configured for installation on the sea ground, too, so that the control unit can be located near the multistage pump on the sea ground. Alternatively, the control unit can be configured for a topside location, for example for a deployment on a FPSO unit or on a platform or at an onshore location.

**[0049]** In addition, according to the invention a pumping arrangement is proposed comprising a multistage pump system having a first multistage pump and further comprising a second multistage pump, wherein the multistage pump system is configured according to the invention, and wherein the first multistage pump and the second multistage pump are arranged in series.

**[0050]** Preferably, the pump outlet of the first multistage pump is connected to the pump inlet of the second multistage pump, so that the discharge pressure of the first multistage pump at least approximately equals the suction pressure of the second multistage pump.

**[0051]** In particular, for such applications where a large pressure is required at the discharge side, it might be advantageous to connect at least two multistage pumps in series to a subsea pumping arrangement, rather than adding additional stages to a single multiphase pump. The at least two multistage pumps are arranged in series. In some embodiments the pump outlet of the first multistage pump is directly connected to the pump inlet of the second multistage pump, e.g. by a piping. In other embodiments one or more additional device(s) is/are arranged between the pump outlet of the first multistage pump and the pump inlet of the second multistage pump, for example a cooling device and/or a buffer device.

**[0052]** Since at least the multistage pump system with the first multistage pump is configured according to the invention, it is possible to avoid pressure spikes at its pump outlet, i.e. at the discharge side of the first multistage pump, as they might be caused in conventional multistage pumps by liquid slugs. Thus, also the pump inlet of the second multistage pump is prevented from receiving such pressure spikes, which is a considerable advantage in particular with respect to the mechanical seals of the second multistage pump, where a reverse pressure across the respective seal may be prevented.

**[0053]** According to a preferred embodiment, each multistage pump is configured for conveying a compressible fluid, being in the dense phase at the pump outlet of the second multistage pump.

**[0054]** According to a preferred embodiment the pumping arrangement is configured for installation on a sea ground.

**[0055]** 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.

**[0056]** 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.

**[0057]** Another application of the multistage pump system according to the invention or the pumping arrangement according to the invention is the transport of $CO_2$ in the dense phase or natural gas in the dense phase or a mixture of both in the dense phase. There are several reasons why transport in a dense phase is preferred over transport in a gaseous phase. The volume flow for the same mass flow is smaller, which means that smaller pipelines can be used and that there is a smaller pressure drop in the pipelines. Another advantage is that supercritical $CO_2$, i.e. dense phase $CO_2$ in the dense phase or natural gas in the dense phase have the capabilities of dissolving a certain amount of water. Unlike in the gaseous phase there is no risk that the entrained water drops out and accumulates in the low points of the pipeline, which is also referred to as liquid hold up. This water accumulation in low points causes significant problems in winter time, since gas hydrates can form and accumulate at these points and gradually close off the pipeline.

**[0058]** The efficient and effective transport of $CO_2$ over long distances is gaining more and more importance given the current developments in CCS (carbon capture and storage) and CCSU (carbon capture, storage and utilization).

**[0059]** The injection of $CO_2$ into oil reservoirs is also used as an enhanced oil recovery method. For onshore oil production, this often requires extensive infrastructure to transport the $CO_2$ from the source to the injection wells.

**[0060]** Another application, where $CO_2$ in the dense phase has to be conveyed, is the use of $CO_2$ as a supercritical solvent in the food processing industry e.g. for the extraction of solutes.

**[0061]** Further advantageous measures and embodiments of the invention will become apparent from the dependent claims.

**[0062]** 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 an embodiment of a multistage pump system according to the invention,

Fig. 4: a flow chart illustrating the operation of the multistage pump,

Fig. 5: diagrams with different parameters during a liquid slug, and

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

**[0063]** Fig. 3 shows a schematic cross-sectional view of an embodiment of a multistage pump system according to the invention, which is designated in its entity with reference numeral 100. The multistage pump system 100 comprises a multistage pump, which is designated in its entity with reference numeral 1 and a control unit 20. The multistage pump 1 comprises a pump unit 3 arranged in a pump housing 30 and a drive unit 4 arranged in a drive housing 40. The control unit is 20 is configured for controlling the drive unit 4.

**[0064]** The pump housing 30 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.

**[0065]** By way of example, in the following description reference is made to an important application, for which the multistage pump 1 is configured as a subsea multistage pump 1 configured for an installation on a sea ground. It has to be noted, that the multistage pump 1 according to the invention is not restricted to subsea applications or to a deployment on the sea ground. The multistage pump 1 according to the invention can also be configured for an installation at topside locations on or above the water surface. For example, the multistage pump 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).

**[0066]** The control unit 20 can be configured for a deployment on the sea ground or for a deployment at a topside location, e.g. on a platform or on a FPSO or ashore. If the multistage pump 1 is arranged at a subsea location, the control unit 20 can be arranged at the subsea location, too or at a topside location, e.g. on a platform or on a FPSO or ashore.

**[0067]** Furthermore, in the example referred to hereinafter, the multistage pump 1 is designed for conveying a compressible fluid having a specific gravity of at most 0.9, preferably between 0.2 and 0.8. The multistage pump 1 is configured as a centrifugal pump. In the embodiment shown in Fig. 1 the pump housing 30 and the drive housing 40 form a common housing 2, in which the pump unit 3 and the drive unit 4 are arranged. The common housing 2 is designed as a pressure housing, which is able to withstand the pressure generated by the multistage pump 1 as well as the pressure exerted on the multistage pump 1 by the environment. The common housing 2 may comprise several housing parts, e.g. the pump housing 30, bearing housing (s), seal housing (s) and the drive housing 40, which are connected to each other to form the common housing 2 surrounding the pump unit 3 and the drive unit 4. It is also possible that a separate pump housing 30 and a separate motor housing 40 are inserted into the common housing 2. The common housing 2 is configured as a hermetically sealed pressure housing preventing any leakage to the external environment.

**[0068]** In particular, the multistage pump 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 multistage pump 1. The points E1, E2, E3 indicate the particular state (pressure, temperature) of the fluid at the pump inlet 21 of the 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 multistage pump 1 for the same application or design of the multistage pump 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 multistage pump 1 and in the dense phase at the pump outlet 22 of the multistage pump 1. 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 multistage pump 1 and at the pump outlet 22 of the multistage pump 1.

**[0069]** In the following description reference is made by way of example to the important application that the multistage pump 1 is designed and adapted for being used as a subsea injection pump 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 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. However, the content of liquid(s) should 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 (not shown) 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. The lighter phase, which contains a considerable amount of $CO_2$, is fed to the multistage pump 1 and injected into a subterranean region of the oil field. 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 pump 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 multistage pump 1 is in particular 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]** The common housing 2 of the multistage pump 1 comprises the pump inlet 21, through which the fluid enters the multistage pump 1, and the pump outlet 22 for discharging the fluid with an increased pressure as compared to the pressure of the fluid at the pump inlet 21. Typically, the pump outlet 22 is connected to a pipe (not shown) for delivering the pressurized fluid to a well, in which the fluid is injected. The pressure of the fluid at the pump outlet 22 is referred to as 'high pressure' or 'discharge pressure', whereas the pressure of the fluid at the pump inlet 21 is referred to as 'low pressure' or 'suction pressure'. A typical value for the difference between the high pressure and the low pressure is for example 100 to 200 bar (10 - 20 MPa).

**[0073]** The pump unit 3 further 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.

**[0074]** 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 1 is an eight stage pump having the first stage impeller 31, the last stage impeller 32 and six intermediate stage impellers 33, which are all arranged in series on the pump shaft 5. Of course, the number of eight stages is only exemplary. In other embodiments the multistage pump 1 may comprise more than eight stages, e.g. ten or twelve stages, or less than eight stages for example four or two stages.

**[0075]** 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 at the low pressure side. 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 the high pressure side of the pump 1.

**[0076]** 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 low pressure to the high pressure.

**[0077]** 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.

**[0078]** The multistage pump 1 is configured as a vertical pump 1, 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.

**[0079]** 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.

**[0080]** 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 and Fig. 6 show the pump 1 in the usual operating position.

**[0081]** 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.

**[0082]** As can be seen in Fig. 3 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 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 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.

**[0083]** 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. 3 by the dashed arrows without reference numeral, the fluid enters the multistage pump 1 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.

**[0084]** For many applications the back-to-back arrangement 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.

**[0085]** For further reducing the overall axial thrust acting on the pump shaft 5 the pump 1 may further comprise a balance drum 7 and/or a center bush 35. This will be explained in more detail hereinafter.

**[0086]** Basically, the pump unit 3 of the multistage pumps 1 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 1 is configured with radial or semi-axial impellers, wherein at least two impellers of the same multistage pump have different specific speeds. 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.

**[0087]** The multistage pump 1 of the multistage pump system 100 according to the present invention can also be configured with helico-axial impellers, i.e. the multistage pump 1 can be configured as a helico-axial pump.

**[0088]** The multistage pump 1 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, 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. 1, 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 1 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 pump 1. The balancing device may comprise a balance drum 7 (also referred to as throttle bush) and/or a center bush 35. The first embodiment of the multistage pump 1 comprises the balance drum 7 and the center bush 35 for at least partially balancing the axial thrust that is generated by the impellers 31, 32, 33.

**[0093]** The balance drum 7 is fixedly connected to the pump shaft 5 in a torque proof manner. The 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 balance drum 7 defines a front side 71 and a back side 72. The front side 71 is the side facing the pump unit 3 and the impellers 33. In the first embodiment the front side 71 is facing the intermediate stage impeller 33 of the fifth stage. The back side 72 is the side facing the axial pump bearing 55 and the drive unit 4. The balance drum 7 is surrounded by a stationary part 26, so that a relief passage 73 is formed between the radially outer surface of the balance drum 7 and the stationary part 26. The stationary part 26 is configured to be stationary with respect to the common housing 2. The relief passage 73 forms an annular gap between the outer surface of the balance drum 7 and the stationary part 26 and extends from the front side 71 to the back side 72.

**[0094]** A balance line 9 is provided for recirculating the fluid from the back side 72 of the balance drum 7 to the low pressure side at the pump inlet 21. In particular, the balance line 9 connects the back side 72 with the low pressure side of the pump 1, where the low pressure, i.e. the pressure at the pump inlet 21 prevails. Thus, a part of the pressurized fluid passes from the front side 71 through the relief passage 73 to the back side 72, enters the balance line 9 and is recirculated to the low pressure side of the pump 1. The balance line 9 constitutes a flow connection between the back side 72 and the low pressure side at the pump inlet 21. The balance line 9 may be arranged - as shown in Fig. 3 - outside the common housing 2. In other embodiments the balance line 9 may be designed as internal line completely extending within the common housing 2.

**[0095]** Due to the balance line 9 the pressure prevailing at the back side 72 is essentially the same - apart from a minor pressure drop caused by the balance line 9 - as the low pressure prevailing at the pump inlet 21.

**[0096]** The axial surface of the balance drum 7 facing the front side 71 is exposed to an intermediate pressure between the low pressure and the high pressure. In the embodiment shown in Fig. 3 said intermediate pressure is the suction pressure of the fifth stage prevailing at the outlet of the crossover line 34 during operation of the pump 1. Of course, due to smaller pressure losses the pressure prevailing at the axial surface of the balance drum 7 facing the front side 71 may be somewhat smaller than said intermediate pressure. However, the considerably larger pressure drop takes place over the balance drum 7. At the back side 72 it is essentially the low pressure that prevails during operation of the Thus, the pressure drop over the balance drum 7 is essentially the difference between the intermediate pressure and the low pressure.

**[0097]** The pressure drop over the 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 impellers 33 of the second, third and fourth stage.

**[0098]** As a further balancing device for reducing the overall axial thrust acting on the pump shaft 5, a center bush 35 is arranged between the first set of impellers 31, 33 and the second set of impellers 33, 32. 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 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 second stationary part 36 being stationary with respect to the common housing 2. A annular balancing passage 37 is formed between the outer surface of the center bush 35 and the second stationary part 36.

**[0099]** The function of the center bush 35 and the balancing passage 37 is in principle the same as the function of the 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, and at the other axial surface facing the intermediate impeller 33 of the fourth stage a lower pressure prevails, which is essentially the same as the intermediate pressure 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 impeller 33 of the fourth stage.

**[0100]** The pressure drop over the center bush 35 essentially equals the difference between the high pressure and the intermediate pressure. Said pressure drop over the center bush 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 impellers 33 of the fifth, sixth and seventh stage and the last stage impeller 32.

**[0101]** The 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.

**[0102]** 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.

**[0103]** Of course, it is also possible, to exchange the position of the second radial drive bearing 44 and the axial drive bearing 45.

**[0104]** 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.

**[0105]** 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.

**[0106]** 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 motor 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.

**[0107]** 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.

**[0108]** The control unit 20 is connected to the electric motor 41 as indicated in Fig. 3 by the dashed signal line S to control the electric motor 41. The control unit 20 is configured to drive the electric motor 41 at different rotational speeds. The control unit 20 is preferably designed as a variable frequency drive (VFD), so that the speed of the electric 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.

**[0109]** 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.

**[0110]** The multistage pump 1 further comprises two sealing units 50 for sealing the pump shaft 5 against a leakage of the fluid along the pump shaft 5. By the sealing units 50 the fluid is prevented from entering the drive unit 4 as well as the pump bearings 53, 54, 55. One of the sealing units 50 is arranged between the balance drum 7 and the axial pump bearing 55 and the other sealing unit 50 is arranged between the first stage impeller 31 and the second radial pump bearing 54. Preferably each sealing unit 50 comprises a mechanical seal. Mechanical seals are well-known in the art in many different embodiments and therefore require no detailed explanation. In principle, a mechanical seal is a seal for a rotating shaft and comprises a rotor fixed to the pump shaft 5 and rotating with the pump shaft 5, as well as a stationary stator fixed with respect to the common 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 of the pump 1.

**[0111]** For the lubrication and the cooling of the sealing units 50 and the pump bearings 53, 54, 55 as well as for the cooling of the drive unit 4 a barrier fluid system (not shown) is provided. Barrier fluid systems 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 drive unit 4, the pump bearings 53, 54, 55 and the sealing units 50. 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 circuit is controlled such that the pressure of the barrier fluid is at least as high as but preferably higher than a reference pressure of the process fluid, here the compressible fluid. According to a preferred configuration, the pressure of the barrier fluid in the circuit is higher than the low pressure at the pump inlet 21.

**[0112]** By this measure there is always a leakage flow of barrier fluid through the sealing units 50 into the pump unit 3. Therefore any leakage flow of the fluid from the pump unit 3 through the sealing units 50 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.

**[0113]** In other embodiments the multistage pump 1 is designed as a process fluid lubricated pump, which does not require a separate barrier fluid that is different from the process fluid, here the compressible fluid. In such embodiments the multistage pump 1 is preferably designed as a seal-less pump, i.e. without the two sealing units 50. The seal-less multistage pump 1 has no mechanical seals.

**[0114]** The term "process fluid lubricated pump" refers to pumps, where the process fluid that is conveyed by the pump 1 is used for the lubrication and the cooling of components of the pump, e.g. bearing units. It is known to use the process fluid as such or a fluid that is produced from the process fluid, e.g. by phase separation or phase enrichment. A process fluid lubricated pump does not require a specific barrier fluid different from the process fluid to avoid leakage of the process fluid e.g. into the drive unit 4, because the process fluid or the fluid produced from the process fluid is deliberately allowed to enter the drive unit 4 and is used for cooling and lubricating components of the pump 1 such as the pump bearings 53, 54 and 55. In addition, a process fluid lubricated pump 1 does not require a lubricant different from the process fluid for the lubrication of the pump components.

**[0115]** In other embodiments (not shown) a further balance drum may be arranged below the lower end of the pump unit 3, namely between the pump unit 3 and the non-drive end 52 of the pump shaft 5, more precisely between the lower end of the pump unit 3 and the second radial pump bearing 54. In still other embodiments a balance drum is provided only at the lower end of the pump 1, between the pump unit 3 and the second radial pump bearing 54 at the non-drive end 52 of the pump shaft 5 and no balance drum is provided above the pump unit 3 near the drive end 51 of the pump shaft 5. In all these embodiments the center bush 35, the second stationary part 36 and the balancing passage 37 in between are optional features, i.e. the multistage pump 1 may be designed with or without these features.

**[0116]** Referring now to Fig. 4 the operation of the multistage pump 1 and in particular the operation of the electric motor 41 by means of the control unit 20 will be described in more detail.

**[0117]** The control unit 20 is configured to operate the electric motor 41 in two different modes, namely in a first mode M1, in which the control unit 20 operates the electric motor 41 at a constant rotational speed, i.e. with a constant frequency, and in a second mode M2, in which the control unit 20 operates the electric motor 41 at a constant engine load of the electric motor 41. Operating the electric motor 41 at a constant engine load means to control the electric motor 41 such that the power which is delivered from the electric motor 41 to the drive shaft 42 and therewith to the pump shaft 5 is maintained at a constant value. To maintain the engine load of the electric motor 41 at a constant value, it is possible to keep the power which is supplied to the electric motor 41 constant. It is also possible to keep the torque generated by the electric motor 41 constant, or to keep the drive current that is supplied to the electric motor 41 for rotating the drive shaft 42 at a constant value. All three possibilities result in at least approximately a constant engine load of the electric motor 41.

**[0118]** During regular operation the control unit 20 operates the electric motor 41 in the first mode M1, i.e. at a constant engine load, e.g. with a constant power input to or power output from the electric motor 41. During operation a control parameter CP is continuously monitored, wherein the control parameter CP is indicative of a rate of change of the engine load. As long as the electric motor 41 is operated at a constant engine load and the density of the fluid at the pump inlet 21 does not considerably change, the control parameter CP remains at least approximately close to zero, because there is no substantial change in the engine load. According to a preferred embodiment the control parameter CP is the temporal change of the drive current that is supplied to the electric motor 41 or proportional to this temporal change. Thus, the control parameter CP is preferably the first time derivative of the drive current or proportional to this time derivative

**[0119]** In case a remarkable increase in the density of the fluid at the pump inlet 21 will occur and the multistage pump 1 is operated in the first mode M1, the control unit 20 initially tries to keep the rotational speed at the given constant value. Such a sudden increase of the density of the fluid at the pump inlet 21 may e.g. be caused by a liquid slug such as a water slug propagating to the pump inlet 21. A liquid slug may occur e.g. due to an instability in the separation device separating the crude oil in a heavier phase having a higher density and a lighter phase having a lower density, wherein the lighter phase is supplied to the multistage pump 1. The lighter phase has for example a density of up to 400 $kg/m^3$, whereas the fluid in the liquid slug has a density of for example 1000 $kg/m^3$ . The sudden increase of the density of the fluid at the pump inlet would cause a considerable increase in the power required to maintain the rotational speed of the pump shaft 5. Consequently, at the very beginning, when the liquid slug enters the multistage pump 1 the control parameter CP will drastically increase. Just as an example, when the multistage pump 1 operates at a constant frequency of 6000 rpm, the occurrence of a liquid slug at the pump inlet 21 of the multistage pump 1 may cause an increase of the drive current of twenty amperes in less than 0.1 second. This enormous increase is reflected in the control parameter CP. If the control parameter CP exceeds a presettable threshold value D, see step 101 in Fig. 4, the control unit 20 immediately changes from the first mode M1 to the second mode M2. Thus, in step 101 it is checked, whether the control parameter CP is larger than the presettable threshold value D. If the control parameter CP does not exceed the threshold value D, the control unit 20 continues to operate the electric motor 41 in the first mode M1 as indicated by the arrow 103 in Fig. 4. If the control parameter CP exceeds the threshold value D, the control unit 20 switches to the second mode M2 as indicated by the arrow 104 in Fig. 4 and operates the electric motor 41 at a constant engine load. To operate the electric motor 41 at a constant engine load, it is for example possible to keep the drive current supplied to the electric motor 41 or the torque generated by the electric motor 41 at a constant value.

**[0120]** As long as the liquid slug propagates through the pump unit 3, the electric motor 41 is operated in the second mode M2. When the liquid slug leaves the multistage pump 1 through the pump outlet 22, the electric motor 41, which is still operated in the second mode M2, starts to accelerate, because as soon as the liquid slug leaves the multistage pump 1, the density of the fluid in the multistage pump 1 returns to the initial value, namely the density of the lighter phase. This acceleration causes an increase in the rotational speed, i.e. the RPM value increases. This is detected in step 102 in Fig. 4. As long as there is no remarkable increase in the rotational speed of the electric motor 41 the control unit 20 continues to operate the electric drive 41 in the second mode M2 as indicated by arrow 105 in Fig. 4. When an increase in the rotational speed is detected, the control unit 20 switches from the second mode M2 to the first mode M1 as indicated by arrow 106. Of course, it is also possible, but not necessary, to define a further threshold value, namely a threshold value for the rotational speed of the drive shaft 42 or the pump shaft 5, respectively, and to switch from the second mode M2 back to the first mode M1 as soon as the actual rotational speed passes the threshold value for the rotational speed. Furthermore, it is possible to switch to the first mode M1 as soon as during the second mode M2 the rotational speed reaches the initial value, i.e. the value prior to the increase of the control parameter CP. It is also possible to switch to the first mode M1 as soon as during the second mode M2 the rotational speed reaches a value which is somewhat lower, e.g. a hundred rpm lower, than the initial value.

**[0121]** For a better understanding, Fig. 5 shows different diagrams with different parameters during a liquid slug. On the left side diagrams are shown illustrating the temporal course of the different parameters, when the control unit 20 would not change to the second mode M2 in case of a liquid slug propagating to the multistage pump 1. On the right side diagrams are shown illustrating the temporal course of the same parameters, when the control unit 20 changes to the second mode M2 in case of a liquid slug propagating to the multistage pump 1.

**[0122]** On the horizontal axis of each diagram the time T is plotted and on the vertical axis the different parameter C1, C2, C3, C4, C5 are plotted. At the time t1 the liquid slug enters the multistage pump 1, and at the time t2 the liquid slug leaves the multistage pump 1.

**[0123]** The parameter C1 indicates the density of the fluid at the pump inlet 21 of the multistage pump. Parameter C1 follows the same curve on the left side and the right side. At the time t1, when the liquid slug enters the multistage pump, the parameter C1 increases to the value of the liquid forming the liquid slug. At the time t2, when the liquid slug leaves the multistage pump the parameter C1 decreases to the value of the lighter phase.

**[0124]** Reference is now made to the left side of Fig. 5, showing an operation in the first mode M1 during the liquid slug.

**[0125]** The parameter C2 indicates the rotational speed of the electric motor and therewith the rotational speed of the pump shaft 5. The parameter C2 is essentially constant during the passage of the liquid slug through the multistage pump 1.

**[0126]** The parameter C3 indicates the discharge pressure of the fluid at the pump outlet 22. Since the multistage pump is operated at a constant rotational speed, the liquid slug causes a pressure spike in the discharge pressure, which is due to the increase in the density of the fluid.

**[0127]** The parameter C4 indicates the power of the electric motor. Qualitatively, the curve for the torque generated by the electric motor and the curve for the drive current look the same as the curve for parameter C4. C4 also indicates the engine load of the electric motor 41. It can be seen that the liquid slug causes a spike in the parameter C4.

**[0128]** The parameter C5 indicates the flow delivered by the multistage pump 1. It can be seen that the liquid slug also causes a moderate increase of the parameter C5.

**[0129]** Reference is now made to the right side of Fig. 5, showing an operation in the second mode M2 during the liquid slug.

**[0130]** Since the electric drive 41 is operated at a constant engine load during the liquid slug, the parameter C2 indicating the rotational speed of the electric motor 41 and therewith the rotational speed of the pump shaft 5 decreases during the passage of the liquid slug through the multistage pump 1.

**[0131]** Both the parameter C3 indicating the discharge pressure and the parameter C4 indicating the power or the engine load of the electric motor, respectively, are constant in time during the passage of the liquid slug through the multistage pump 1.

**[0132]** Furthermore, also the parameter C5 indicating the flow delivered by the multistage pump 1 is constant in time during the passage of the liquid slug through the multistage pump.

**[0133]** Thus, by operating the multistage pump 1 in the second mode M2 during the passage of the liquid slug, the spikes in the torque, in the drive current and in the discharge pressure can be reliably prevented.

**[0134]** The embodiment of the multistage pump 1 illustrated in Fig. 3 comprises a back-to-back arrangement of the impellers 31, 32, 33. In other embodiments the multistage pump 1 of the multistage pump system 100 according to the invention can be 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, e.g. downwards in the axial direction A. The flow of the fluid from the pump inlet 21 (suction pressure) towards the pump outlet 22 (discharge pressure) is always directed in the same direction, namely either in upward direction or in downward direction, and does not change as in the back-to-back arrangement (Fig. 3).

**[0135]** In an inline arrangement the balance drum 7 is e.g. arranged at the upper end of the pump unit 3 adjacent to the last stage impeller, namely between the last stage impeller and the drive end of the pump shaft. The front side of the balance drum is in fluid communication with the pump outlet. The balance line is provided for recirculating the fluid from the back side of the balance drum to the suction pressure side at the pump inlet. Due to the balance line the pressure prevailing at the back side is essentially the same - apart from a minor pressure drop caused by the balance line - as the suction pressure prevailing at the pump inlet.

**[0136]** The axial surface of the balance drum facing the front side is exposed to the discharge pressure prevailing at the pump outlet. Thus, the pressure drop over the balance drum essentially equals the pressure difference between the discharge pressure at the pump outlet and the suction pressure at the pump inlet.

**[0137]** Of course, due to smaller pressure losses the pressure prevailing at the axial surface of the balance drum facing the front side may be somewhat smaller than the discharge pressure. However, the considerably larger pressure drop takes place over the balance drum.

**[0138]** The pressure drop over the balance drum results in a force that is directed in the axial direction A and opposite to the axial thrust generated by the plurality of impellers therewith at least partially balancing the axial thrust.

**[0139]** Both for an inline arrangement and for a back-to-back arrangement (Fig. 3) the number of individual impellers 31, 32, 33 forming the first set of impellers 31, 33 and the number of individual impellers forming the second set of impellers 33, 32 may be different or may be the same. It depends on the respective application, whether the first set and the second set have the same number of impellers or whether the first set of impellers has a different number of impellers than the second set of impellers.

**[0140]** Fig. 6 shows a schematic cross-sectional view of an embodiment of a pumping arrangement according to the invention, which is designated in its entity with reference numeral 200.

**[0141]** The pumping arrangement 200 may be configured for installation on a sea ground and comprises at least a multistage pump system 100 and a second multistage pump 1b, wherein the multistage pump system 100 comprises a first multistage pump 1a. At least the multistage pump system 100 is configured according to the invention. The second multistage pump 1b may be incorporated in a multistage pump system which is configured according to the invention, i.e. a control unit can be provided for the second multistage pump 1b, too, wherein the control unit is configured in an analogous manner as the control unit 20 for the first multistage pump 1a. The second multistage pump 1b and/or the control unit for the second multistage pump 1b can also be configured in a different manner, e.g. in any manner that is known from the state of the art.

**[0142]** The first multistage pump 1a may be configured for example as explained with respect to the embodiment of the multistage pump system 100 according to the invention and illustrated in Fig. 3 or as a multistage pump 1 having an inline design of the impellers. The first multistage pump 1a and the second multistage pump 1b may be configured in an identical manner or they may be configured in different manners. Just as an example, the first multistage pump 1a may be configured according to the embodiment shown in Fig. 3 and the second multistage pump 1b may be configured with an inline arrangement of the impellers.

**[0143]** In the embodiment illustrated in Fig. 6 both the first multistage pump 1a and the second multistage pump 1b are configured with eight stages and an back-to-back arrangement of the impellers 31, 32, 33.

**[0144]** As can be seen in Fig. 6 the first multistage pump 1a and the second multistage pump 1b are arranged in series.

**[0145]** The multistage pump system 100 with the first multistage pump 1a comprises the control unit 20 for operating the electric motor 41 in the first mode M1 or in the second mode M2. Also for the second multistage pump 1b a control unit 20 can be provided, which is configured in the same manner, i.e. for operating the electric motor 41 of the second multistage pump 1b in the first mode M1 or in the second mode M2. Alternatively for the second multistage pump 1b a motor control unit can be provided as it is known from the state of the art.

**[0146]** When the pumping arrangement 200 is configured as a subsea pumping arrangement 200 it 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 1a, 1b in series rather than adding additional stages to a single multistage pump 1.

**[0147]** The pump outlet 22 of the first multistage pump 1a is connected to the pump inlet 21 of the second multistage pump 1b by means of a piping 112. In some embodiments the pump outlet 22 of the first multistage pump 1a is directly connected to the pump inlet 21 of the second multistage pump 1b without any additional device in between. In other embodiments - as it is shown in Fig. 6 - one or more additional device(s) 113, 114 is/are arranged between the pump outlet 22 of the first multistage pump 1a and the pump inlet 21 of the second multistage pump 1b. In the embodiment illustrated in Fig. 6 the outlet 22 of the first multistage pump 1a is connected to a cooling device 113. From the cooling device 113 the pressurized fluid is guided to a buffer 114. The outlet of the buffer 114 is connected to the inlet 21 of the second multistage pump 1b. The outlet 22 of the second multistage pump 1b is connected to a well (not shown) leading to a subterranean region, in which the fluid, e.g. the carbon dioxide, is injected.

## Claims

1. A multistage pump system for conveying a fluid, comprising a multistage pump (1) and a control unit (20),

   wherein the multistage pump comprises a pump unit (3) arranged in a pump housing (30), and a drive unit (4) arranged in a drive housing (40),
   wherein the control unit (20) is configured for controlling the drive unit (4),
   wherein the pump housing (30) comprises a pump inlet (21) for receiving the fluid with a suction pressure, and a pump outlet (22) for discharging the fluid with a discharge pressure,
   wherein the pump unit (3) comprises a pump shaft (5) for rotating about an axial direction (A), and a plurality of impellers (31, 32, 33) for conveying the fluid from the pump inlet (21) to the pump outlet (32), with each impeller (31, 32, 33) being mounted to the pump shaft (5) in a torque proof manner,
   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 the control unit (20) is configured to drive the electric motor (41) at different rotational speeds,

   **characterized in that**
   the control unit (20) is configured to operate the electric motor (41) in a first mode (M1), in which the control unit (20) operates the electric motor (41) at a constant rotational speed, and in a second mode (M2), in which the control unit (20) operates the electric motor (41) at a constant engine load, wherein the control unit (20) is further configured to switch from the first mode (M1) to the second mode (M2), when a control parameter (CP) that is indicative of a rate of change of the engine load exceeds a presettable threshold value (D).

2. A multistage pump system in accordance with claim 1, wherein the control unit (20) is configured to switch from the second mode (M2) to the first mode (M1), when the rotational speed increases during operation in the second mode (M2).

3. A multistage pump system in accordance with anyone of the preceding claims, wherein the control parameter (CP) indicates the temporal change of a drive current that is supplied to the electric motor (41) for driving the electric motor (41).

4. A multistage pump system in accordance with anyone of the preceding claims, which is configured for conveying a compressible fluid being in the dense phase at the pump outlet (22).

5. A multistage pump system in accordance with claim 4, wherein the multistage pump is configured as an injection pump for injecting the compressible fluid being in the dense state into a subterranean region.

6. A multistage pump system in accordance with anyone of the preceding claims, wherein the pump housing (30) and the motor housing (40) form a common housing (2), in which the pump unit (3) and the drive

unit (4) are arranged.

**7.** A multistage pump system in accordance with anyone of the preceding claims, wherein the multistage pump 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).

**8.** A multistage pump system in accordance with anyone of the preceding claims, wherein the multistage pump is configured as a single phase pump for conveying a single phase fluid.

**9.** A multistage pump system in accordance with anyone of the preceding claims, wherein at least the multistage pump is configured for installation on a sea ground.

**10.** A pumping arrangement comprising a multistage pump system having a first multistage pump (1a) and further comprising a second multistage pump (1b), wherein the multistage pump system (1a) is configured according to anyone of the preceding claims, and wherein the first multistage pump (1a) and the second multistage pump (1b) are arranged in series.

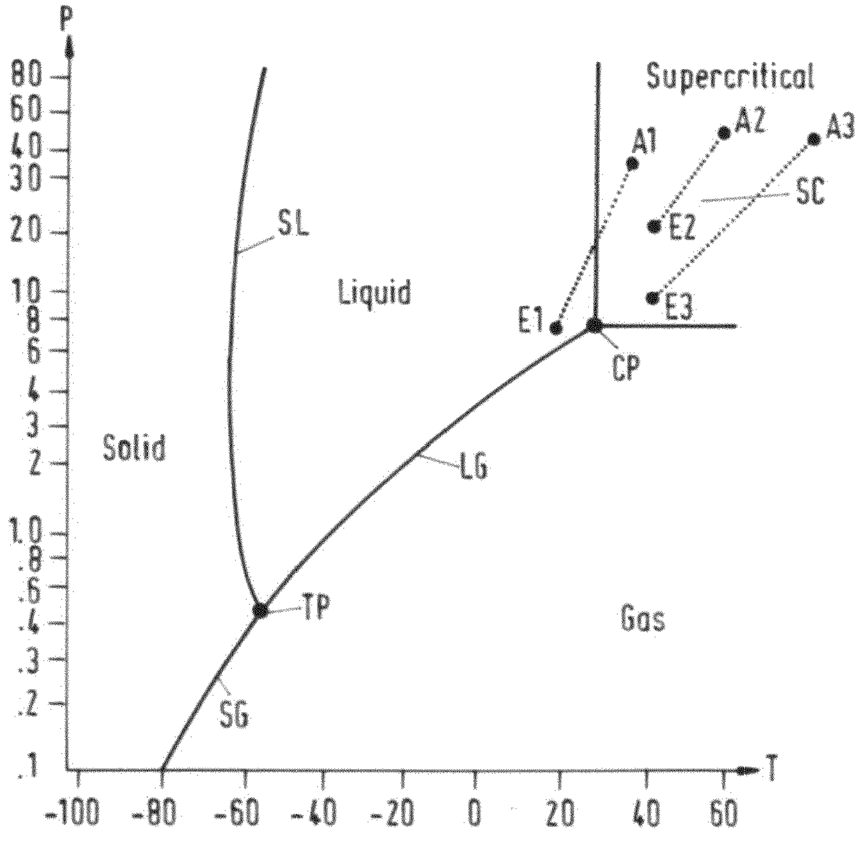
**11.** A pumping arrangement in accordance with claim 10, wherein the pump outlet (22) of the first multistage pump (1a) is connected to the pump inlet (21) of the second multistage pump (1b), so that the discharge pressure of the first multistage pump (1a) at least approximately equals the suction pressure of the second multistage pump (1b).

**12.** A pumping arrangement in accordance with anyone of claims 10-11, wherein each multistage pump (1a, 1b) is configured for conveying a compressible fluid, being in the dense phase at the pump outlet (22) of the second multistage pump (1b).

**13.** A pumping arrangement in accordance with anyone of claims 10-12, configured for installation on a sea ground.

**14.** 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 (1b) into a subterranean region.

Fig. 1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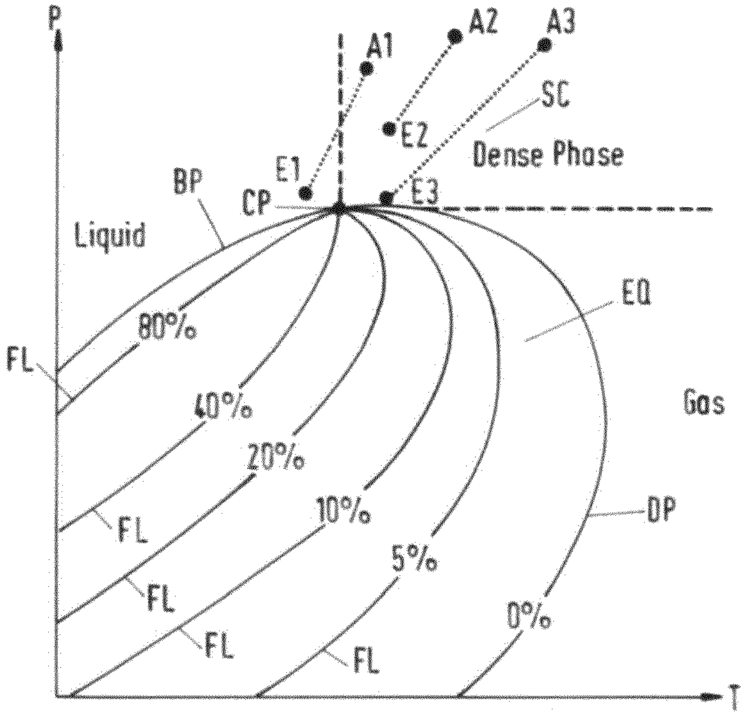
.1
T
-100
-80
-60
-40
-20
0
20
40
60
Supercritical
A1
A2
A3
SC
E1
E2
E3
CP
SL
Liquid
Solid
LG
TP
Gas
SG

Fig. 2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CP
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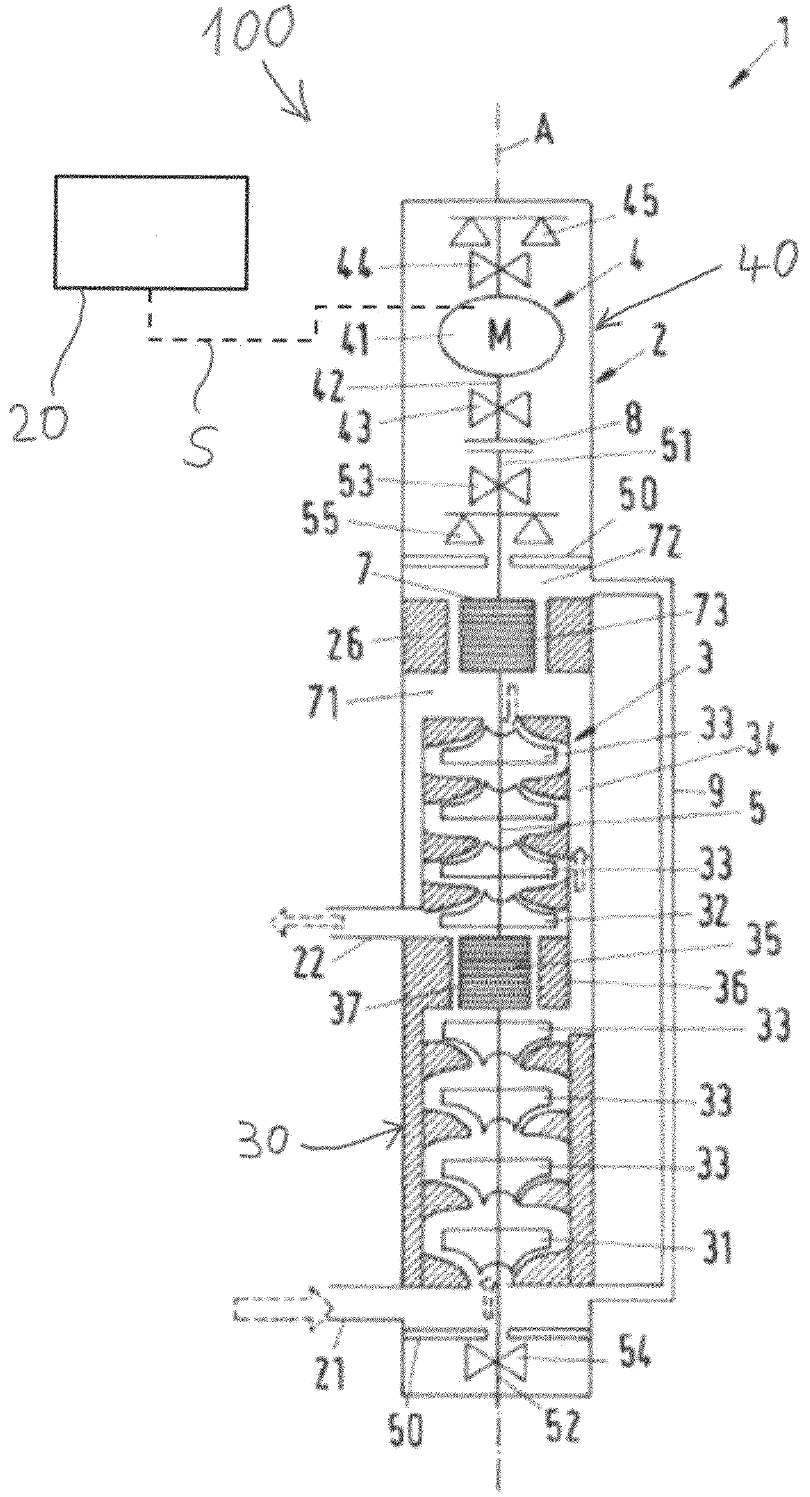
0%
FL
T

Fig. 3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71
33
34
9
5
33
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22
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37
33
33
30
33
31
21
54
50
52
20
S

Fig. 4

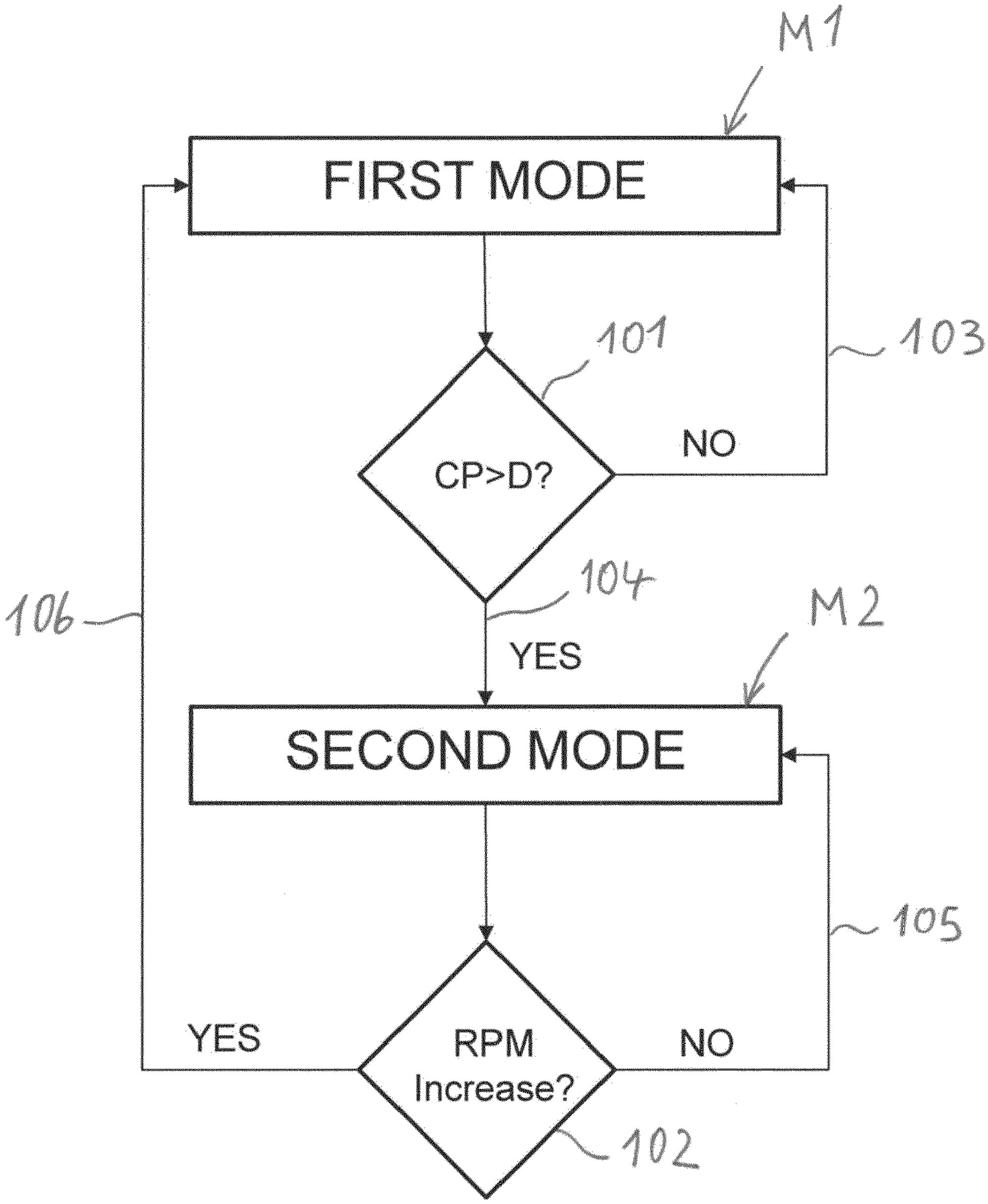

M1
FIRST MODE
101
103
CP>D?
NO
104
YES
106
M2
SECOND MODE
105
YES
RPM
Increase?
NO
102

Fig. 5

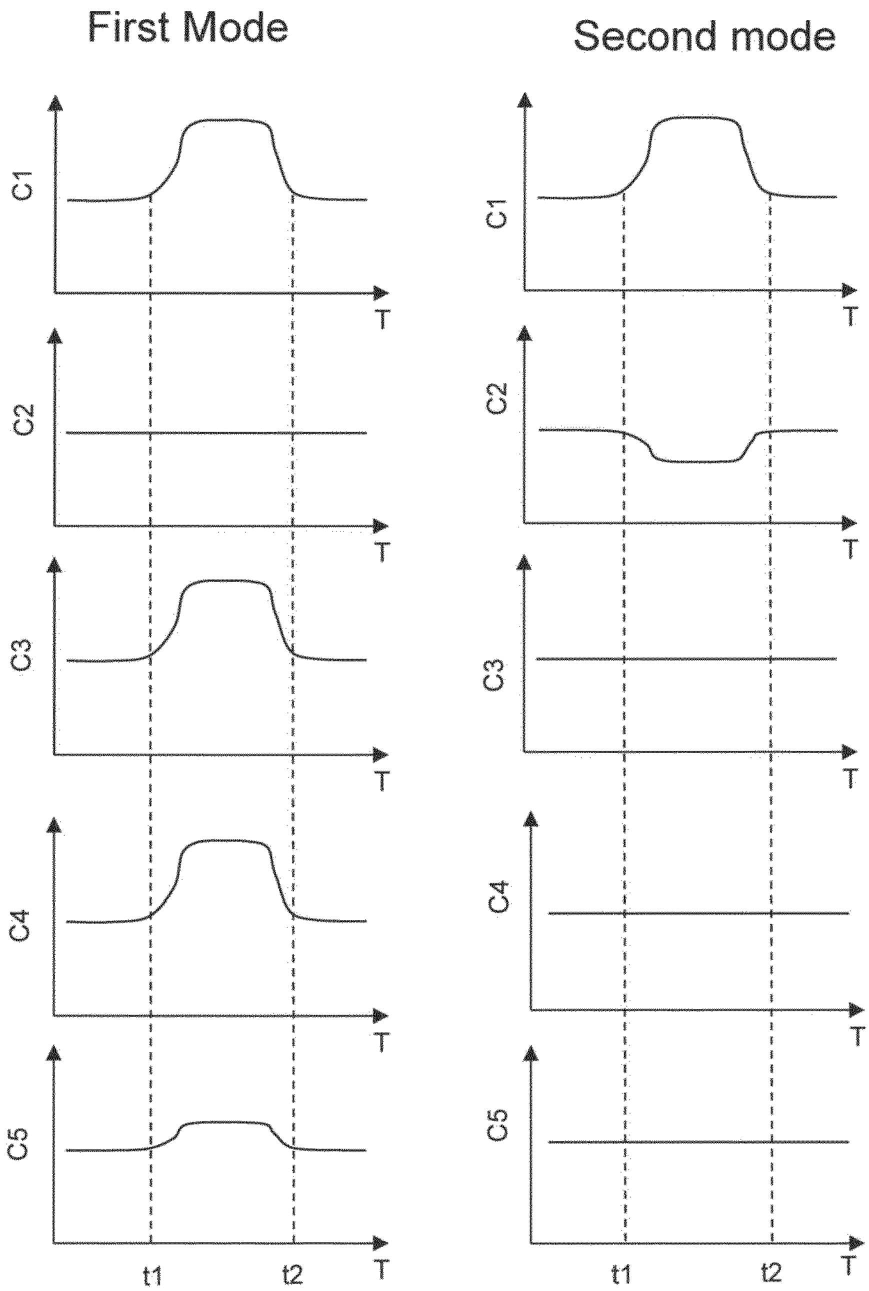
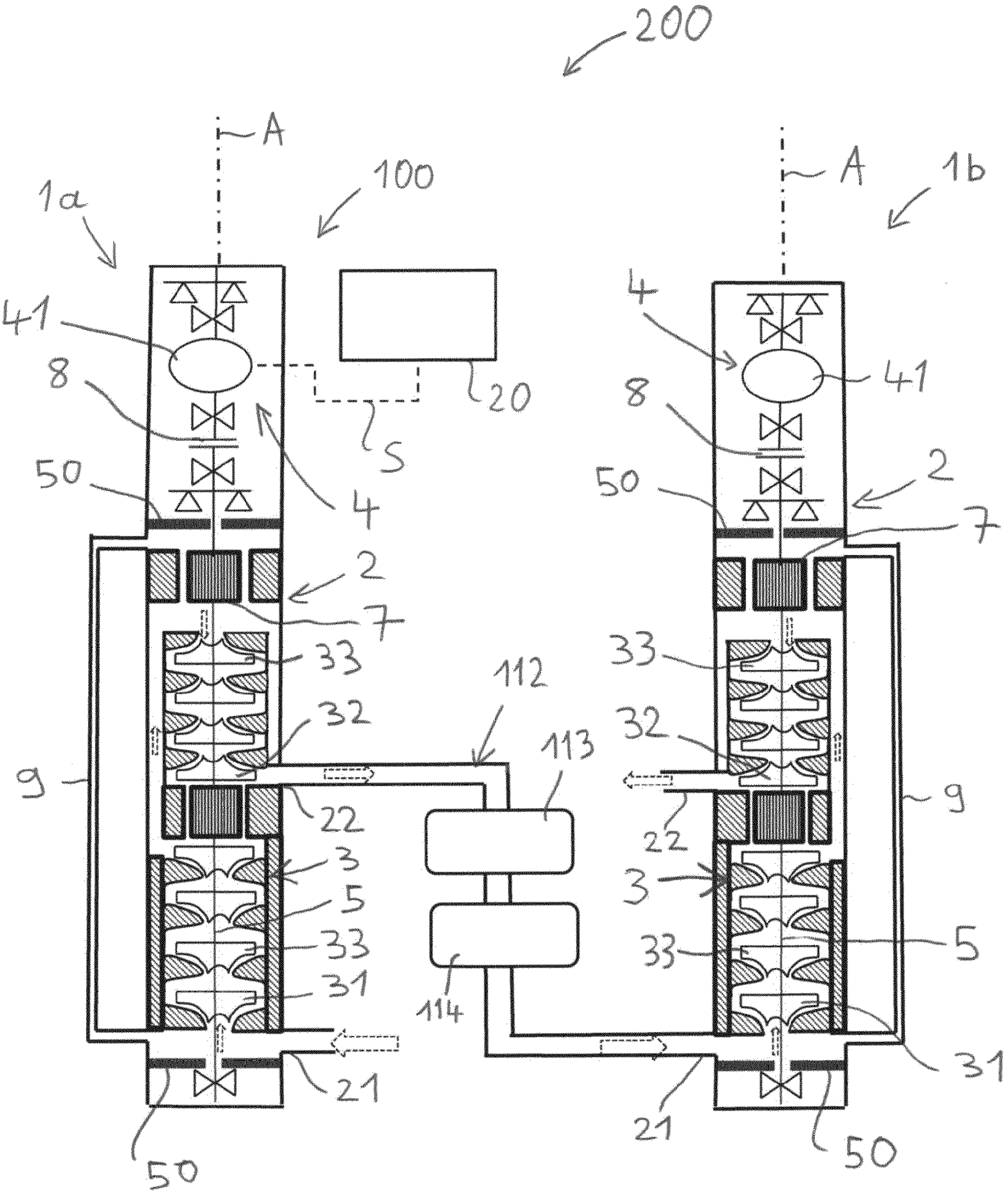


Fig. 6



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Application Number
EP 22 16 7420

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| Category | Citation of document with indication, where appropriate, of relevant passages | Relevant to claim | CLASSIFICATION OF THE APPLICATION (IPC) |
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| X<br>Y | US 2021/108497 A1 (BRENNE LARS [NO]) 15 April 2021 (2021-04-15)<br>* paragraph [0037] - paragraph [0045] *<br>* paragraph [0063] *<br>* paragraph [0079] *<br>* paragraph [0104] - paragraph [0108] *<br>* figures 1, 2 *<br>----- | 1-3,8,9<br>4-7, 10-14 | INV.<br>F04D31/00<br>F04D17/12<br>F04D25/06<br>F04D27/02<br>F04D1/06<br>F04D15/00 |
| Y,D | EP 3 771 828 A1 (SULZER MANAGEMENT AG [CH]) 3 February 2021 (2021-02-03)<br>* paragraph [0049] - paragraph [0050] *<br>* paragraph [0058] - paragraph [0060] *<br>* paragraph [0069] *<br>* paragraph [0126] - paragraph [0128] *<br>* figure 3 *<br>----- | 4-7, 10-14 | |
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The present search report has been drawn up for all claims

1

| Place of search | Date of completion of the search | Examiner |
|---|---|---|
| The Hague | 7 June 2022 | Oliveira, Damien |

CATEGORY OF CITED DOCUMENTS

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## ANNEX TO THE EUROPEAN SEARCH REPORT ON EUROPEAN PATENT APPLICATION NO. EP 22 16 7420

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07-06-2022

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