Apparatus and Method for Controlling Supply of Barrier Gas in a Compressor Module

An apparatus for controlling the supply of barrier gas to a compressor module comprising a pressure housing (3) containing an electric motor (1) which via a shaft (8) is drivenly connected to a compressor (2). The shaft (8) is supported by magnetic bearings (11, 21) and at least one axial seal (31) around the shaft (8) which divides the pressure housing into a first compartment (10) comprising the motor (1) and a second compartment (20) comprising the compressor (2), the second compartment (20) comprising an inlet (4) and an outlet (5) for fluid that is to be compressed. A line (7) is fluid-connected to the first compartment (10) for the supply of barrier gas from a reservoir (6), and there is a flow restriction means (71) in the line between the reservoir (6) and the first compartment (10). The apparatus facilitates a method for controlling the supply of barrier gas on the basis of required barrier gas velocity (vg) through the seal (31; 32) and the pressure (ps) in the second compartment (20). An overpressure (pa) can be set in the reservoir (6) and a controlled supply of barrier gas to at least the first (10) compartment is effected with the aid of the flow restriction (71; 71') in the feed pipe (7; T).
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Apparatus and method for controlling supply of barrier gas in a compressor module

The invention relates to compressor modules for compressing hydrocarbon gases in a wellstream, and more specifically it relates to a compressor module comprising a pressure housing, a compressor and a motor separated by a sealing element, as disclosed in more detail in the preamble of independent claims 1, 2 and 8. The invention is especially suitable for use in subsea compressor modules.

A compressor module is, in its most basic form, a unit in which a compressor and a motor are connected via a shaft and placed in a common pressure shell. Between the motor and the compressor there is a seal to prevent contamination of the motor. However, there are problems in keeping the gas-filled, electric motor as dry as necessary in order to avoid corrosion and other problems related to the precipitation of hydrocarbon condensates and water in liquid form inside the motor. It is especially important to avoid the presence of water in liquid form together with an H₂S or CO₂ content which may generate acids and, as a result, accelerated corrosion. These problems are addressed in Norwegian Patents NO 172075 and NO 173197, and also in Norwegian Patent Application 20015199. It is also important to prevent particles from penetrating into and building up inside the motor and magnetic bearings.

It is most common to utilise standard oil-lubricated bearings or the like in compressors and their motors. The inventor has explored the possibilities for reliable use of a high-speed motor and magnetic bearings in integrated compressor modules which have a common pressure shell, by providing a way of protecting the electric motor and magnetic bearings in the compressor module. Magnetic bearings in such compressor modules will have several advantages, particularly during operation. Magnetic bearings are more reliable and less expensive to operate. Of particular importance is the fact that the use of magnetic bearings eliminates lubricating oil and thus potential problems that may occur through: dilution of the lubricating oil by the hydrocarbon gases with which it is in contact, accumulation of hydrocarbon condensates or water in the lubricating oil, or degradation of the lubricating oil over time due to its specific use in compressor modules. The problem encountered when utilising non-encapsulated magnetic bearings in a compressor module is in many respects similar to the problems associated with the use of electromotors: both require a completely dry and inert atmosphere with as low a particle content as practically possible in order to function reliably and problem-free over time. Encapsulated magnetic bearings also exist or are in the process of being
developed. It is claimed that these bearings are capable of operating in the untreated wellstream of hydrocarbon gas. However, there are reasons to believe that also in the case of these types of magnetic bearings it is advantage for their long-term functionality and reliability that they should be installed and operated in a dry and particle-free atmosphere.

There is therefore a need for a system and a method for ensuring a completely or nearly completely dry environment for the electric motor and for the magnetic bearings in an integrated compressor module, and which also prevent particles (e.g., from the wellstream flowing through the compressor) from moving from the compressor side into the motor and/or magnetic bearings.

Accordingly, an apparatus is provided for controlling the supply of barrier gas to a compressor module comprising a pressure housing containing an electric motor which via a shaft is drivably connected to a compressor, wherein the shaft is supported by magnetic bearings and an axial seal around the shaft divides the pressure housing into a first compartment comprising the motor and a second compartment comprising the compressor, the second compartment comprising an inlet and an outlet for fluid that is to be compressed, characterised by a line fluid-connected to the first compartment for supply of barrier gas from a reservoir, and a flow restriction means in the line between the reservoir and the first compartment.

In one embodiment of the invention, an apparatus is provided for controlling the supply of barrier gas to a compressor module comprising a pressure housing containing an electric motor which via a shaft is drivably connected to a compressor, wherein the shaft is supported by first magnetic bearings and at least one second magnetic bearing on the opposite side of the compressor in relation to the first magnetic bearings, and an axial seal around said shaft divides the pressure housing into a first compartment comprising the motor and a second compartment comprising the compressor, the second compartment comprising an inlet and an outlet for fluid that is to be compressed, and wherein a second axial seal is fitted around the shaft between the compressor and the said second magnetic bearing thereby forming a third compartment, characterised by a line fluid-connected to the first compartment and a line fluid-connected to the third compartment, the lines being arranged for supply of barrier gas from a reservoir, and respective flow restriction means in the lines between the reservoir and the first and the third compartment respectively.
Furthermore, a method is provided for controlling the supply of barrier gas to a compressor module, characterised by, on the basis of required barrier gas velocity through the seal and the pressure in the second compartment, setting an overpressure in the reservoir, and effecting a controlled supply of barrier gas to at least the first compartment by means of the flow restriction in the feed pipe.

An embodiment of the invention will now be described with reference to the attached drawings, wherein like parts have been given like reference numerals.

Figure 1 is a schematic diagram of the apparatus according to the invention.

Figure 2 is a schematic diagram of one embodiment of the barrier gas control device.

Figure 3 is a schematic diagram of an alternative embodiment of the barrier gas control device.

Figure 4 is a schematic diagram showing the gas velocity and the mass transport through the axial seal.

Figure 5 illustrates the barrier gas control device in a basic form.

Figure 6 illustrates a mechanical barrier gas control device that keeps the mass flow constant when the pressure upstream of the controller, \( p_\text{s} \), is constant and the temperature is constant, as the temperature will be when the controller is surrounded by seawater.

The compressor module according to the invention comprises an electric motor 1 and a compressor 2 joined together via a shaft 8 and arranged in a common pressure shell 3. Figure 1 illustrates schematically that a stream of essentially hydrocarbon gas, but which may also contain liquids and particles (hereafter referred to as fluid) is conducted from a reservoir (not shown), preferably via a liquid separator and in a line 17, into the compressor module via an inlet 4. On the upstream side (inlet side) of the compressor 2, the fluid has a suction pressure designated \( p_\text{s} \), which in a wellstream compression system will fall slowly over time as the reservoir pressure falls. On the downstream side of the compressor, the fluid has a discharge pressure designated \( p_\text{d} \). When the compressor is in operation, \( p_\text{d} \) is greater than \( p_\text{s} \). The fluid exits the compressor module via the outlet 5. In cases where the compressor is not in operation and is inactive and
the inlet 4 is closed by means of the valve 41 and the outlet is closed by means of the valve 51, the pressure inside the compressor module compartments will equalise and become identical. If the bypass valve 61 is closed, the pressure upstream 51 may build up to become greater than the pressure downstream 61. The maximum limiting case for this pressure build-up, due to leakage through the valves at the wellhead, is the shut-down pressure for the wells (wellhead shut-in pressure, WSHP). When the compressor is not in operation and the bypass valve 61 is open, the pressure upstream 51 and downstream 61 will be equal to the settle-out pressure. If production is then shut down over a longish period of time, and the valves at the gas receiving terminal are closed, the pressure in the transport pipe past the compressor module may build up to WSHP. In both the said cases of an inactive compressor, the pressure inside the compressor module may therefore, due to leakage in either valve 51 or 61 or both, build up to WSHP. To counter the ingress of fluid into the compressor module and the consequences this may have as regards condensation, corrosion and particle build-up when it is inactive, it is therefore advantageous to up-adjust the pressure inside the compressor module to slightly above the pressure in pipes outside the valves 51 and 61, and optionally to slightly above WHSP.

An axial seal 31 is arranged between the compressor 2 and the motor 1 and divides the pressure shell into a first compartment 10 and a second compartment 20. As is generally known, the purpose of the axial seal 31 is to prevent particles, moisture and other contamination from coming into contact with the motor and other electrical equipment. However, the axial seal is not completely tight, hi the pursuit for a maximally dry and clean atmosphere in the first compartment 10, it is essential that the pressure in the first compartment 10 (where the motor is located) is higher than the pressure in the second compartment 20 (where the compressor is located) i.e., that inert gas flows from compartment 10 through the seal and into 20. Figure 1 indicates this by showing that the pressure in the first compartment 10 (and in the optional third compartment, due to the balance piston) is $p_s + \Delta p$.

The shaft 8 is journalled by magnetic bearings 11, 21 as shown in Figure 1. If the magnetic bearing 21 is of the encapsulated type as described above, the axial seal 32 is superfluous. The axial seal 32 is thus optional if the magnetic bearing is encapsulated.

Figure 1 shows a feed line 7 for gas, connected at one end thereof to the first compartment 10 and at the other end to a gas reservoir 6 which contains gas at a pressure $p_a$ and a temperature $T_a$. The gas reservoir 6 may (optionally) be fluid-connected to another supply via a supply line 9, e.g. in a so-called umbilical. Thus, by
means of the feed line 7, gas may be delivered to the first compartment 10 from the reservoir 6. This gas will be a dry hydrocarbon gas or a dry inert gas 8 (extraneous gas) which is also clean of particles. This gas may come from a separate gas supply or dried hydrocarbon gas may be used. Regardless of the source, the gas supplied to the compressor module via the line 7 (and the line 7', which will be described below) will hereafter be referred to as "barrier gas".

hi view of the friction loss and thus the heat generation in the motor which must be cooled off (e.g.) by heat exchange with the surrounding seawater in a heat exchanger (not shown in the figures), it is advantageous that the barrier gas supplied should have as low a molecular weight as possible, and thus as low a density as possible at a given pressure and temperature, because the friction decreases with decreasing gas density. To obtain minimum friction and thus minimum loss and maximum efficiency, the use of methane (CH₄; molecular weight 16) is therefore more favourable than nitrogen (N₂; molecular weight 28). Hydrogen (H₂; molecular weight 2) is most favourable as regards reduced loss. The density of hydrogen will be about only10% of a typical natural gas mixture (molecular weight 20). The density of hydrogen at 100 bars will therefore only correspond to the density of natural gas at 20 bars, with corresponding reduction of the friction loss in the motor, cooler requirement and increased efficiency of the motor.

It is usual to refer to the side of the compressor where the motor is located as the "drive end" (DE) of the compressor module, and to the other side as the "non-drive end" (NDE) of the compressor module. A person of skill in the art will understand that in a practical embodiment, the compressor's DE magnetic bearings may be located in a clutch case between the motor and the compressor, and the seal will in that case be between the clutch case and the compressor. The clutch case and the motor nevertheless form an essentially common compartment with a common, clean atmosphere filled by the barrier gas. If the magnetic bearings are of the encapsulated type, and the operating conditions and/or the composition of the wellstream gas (the fluid) are such that these bearings can be allowed to operate directly in the wellstream gas, it is only the motor that requires protection, and the compressor's DE magnetic bearings can therefore be located on the compressor side of the seal.

As mentioned above, an optional axial seal 32 may divide the second compartment 20 into a further, third compartment 30 in which the compressor module's NDE magnetic bearings maybe located. An optional line 7' or 7" for the supply of barrier gas from
the reservoir 6 to this third compartment is shown in Figure 1. If the magnetic bearings are of the encapsulated type, and the operating conditions and/or the composition of the wellstream gas (the fluid) are such that these bearings can be allowed to operate directly in the fluid, it is only the motor that requires protection, hi this case the seal 32 is optional.

To prevent contaminants from penetrating into the second compartment 20 (which contains the compressor 2) and past (i.e., through) the axial seals 31, 32 and into the first (or the third) compartment, the linear velocity of the barrier gas through the axial seals must be greater than or equal to and oppositely directed from the velocity of the fluid (and particles therein) in the second compartment 20.

As regards gaseous contaminants in the fluid (H₂S, CO₂, water vapour and hydrocarbon vapour molecules), there will be a certain, and very small rate of diffusion, in the range of less than 1 mm/s, which moves these components in the opposite direction of the barrier gas stream in the seal. An extremely small supply (mass flow per time unit) of barrier gas into the seal will therefore be sufficient to keep these contaminants out of the "clean" compartments, e.g., the first and the third compartment.

A critical case will be possible ingress of particles and possibly droplets that fall from the wellstream towards the seal 32 (for NDE magnetic bearings), e.g., if the compressor module is arranged vertically. Such particles may have a vertical velocity of fall in excess of 0.1 m/s.

Figure 4 illustrates how a minimum required barrier gas velocity through an axial seal to prevent particles from falling down through a vertically arranged seal can be calculated. The barrier gas flow (m) is illustrated in Figures 1 and 4. The velocity of fall is calculated using Stoke's law:

\[ V_p = \frac{D_p^2 (p_p - p)g}{18\mu} \]

In the formula and the figures, the symbols have the following meanings:

- \( V_p \): velocity of fall for a particle into the seal, m/s
- \( V_g \): barrier gas velocity through the seal, m/s
- \( D_p \): particle diameter, m
- \( p_p \): particle density, kg/m³
- \( p \): gas density, kg/m³
g: acceleration of gravity, m/s²
µ: dynamic viscosity, kg/m·s

Assuming that the barrier gas is methane (CH₄), the temperature is 20°C and the particle density is 2500 kg/m³, it is found that:
- Velocity of fall for a particle of a diameter of 50 μm: 0.25 m/s
- Velocity of fall for a particle of a diameter of 25 μm: 0.06 m/s

Owing to the position of the compressor module in the wellstream, normally after a liquid separator, very few of the particles entering the compressor will be greater than 25 μm, and almost none will be greater than 50 μm. Theoretically, an upward barrier gas velocity greater than 0.25 m/s through a vertical seal will therefore be sufficient to prevent particles from entering the seal. To be on the safe side, a substantially greater barrier gas velocity can be chosen to keep contaminants out. If, for example, it is decided to supply a volume stream of barrier gas such that a velocity of 1.25 m/s is obtained in the seal, this will give a safety factor of 5. In practice, the final determination of the supply of barrier gas will, in addition to calculations as indicated above, be based on practical experience. The final choice of barrier gas velocity and other necessary dimensioning will be based on a balance of safety against influx of contaminants through the seal under stationary and transient conditions, and to avoid unduly large dimensioning of the barrier gas supply system. It is particularly important to keep the supply pipe (e.g., the line 9) for barrier gas within a favourable diameter, for example, less than 25 mm, and also avoid the need for more than one feed pipe for one compressor station.

The choice of a barrier gas velocity of between 1 m/s and 5 m/s, depending upon the actual operating conditions, will normally result in protection against influx and build-up of harmful levels of contaminating gases, condensed water, hydrocarbon condensate and particles in the motor and magnetic bearings, whilst avoiding an unfavourably large barrier gas supply system. It is the chosen barrier gas velocity through the seal, based on an understanding of the mechanisms which may lead to the ingress of contaminated fluid through the seals, which forms the basis for the dimensioning of the supply and feed system and the control of the supply of barrier gas.

From the formula for Stake's law, it is apparent that the velocity of fall for a particle is more or less dependent upon pressure and temperature. This is the case because the gas density, which is a function of pressure and temperature, is very small (typically in the range of 40-100 kg/m³) in relation to the particle density (typically 2500 kg/m³). The
gas density may therefore be neglected in the calculation. Furthermore, the dynamic viscosity varies little in the relevant pressure range (typically 50-100 bars) and temperature range (typically 50-80°C). This means that the mass flow supply of barrier gas over time (and in step with falling wellstream pressure) can be reduced proportionally with the reduction in the pressure on the inlet side of the compressor, \( p_s \), without any increase in the danger of an ingress of particles, because the flow rate of barrier gas through the seal will then be maintained constant.

As a rule, the compressor module will be positioned horizontally, and in that case particles/droplets have no velocity of fall in the direction of the seal. The indicated (above) barrier gas velocities in the seal will in such cases provide extensive protection against contamination of the motor and seals under stationary operating conditions ("steady state"). Such a chosen barrier gas velocity will also impede particles that might be thrown against the seals by the rotating impellers and the shaft, because their momentum is quickly taken from them by the counter-flowing barrier gas and by the actual physical barrier formed by the seal. Even under transient conditions, for example, the starting and the stopping of the compressor module or on the shut-down or start-up of wells, there will be substantial protection against contamination. Even if gaseous contaminants were briefly to penetrate into the motor under transient conditions, they would be so diluted by the clean barrier gas in the motor that they would not reach harmful levels. The same applies to the build-up of particles.

Also in the cases where the compressor module is horizontal, it is possible - as in the case of vertical modules (described above) - to argue that the supply of barrier gas can be reduced when the pressure on the inlet side of the compressor drops, because the gas flow rate through the seal will be maintained with less supplied mass flow.

In Figure 4 it is shown that the seal 31, 32 forms an annulus around the shaft 8. This annulus may typically have an aperture of about 0.3 mm. To show the extent of the need for barrier gas, there has been chosen a barrier gas velocity of 2.5 m/s, a shaft diameter of 150 mm and a gas pressure of 100 bars. It is further assumed that the barrier gas is methane gas. Calculations then show that the need for supply (in other words, mass flow, \( m \)) of barrier gas for a seal will be in the range of 0.0068 kg/s. For a compressor module with two seals, where both DE and NDE magnetic seals are in a protective and "clean" atmosphere, the need for barrier gas will be about 0.02 kg/s. Such a low consumption of barrier gas will result in a favourable dimensioning of the
supply system for barrier gas, including pipes in the umbilical, even with several compressor modules in a subsea compressor station.

To obtain protection of the motor and magnetic bearings when the compressor is in operation, the invention is thus based on utilising, in combination, the knowledge, of

- necessary barrier gas velocity, \( v_g \), through the seal; and
- knowledge of (measurement of) the pressure inside the compressor compartment facing the seals, \( p_s \), (normally equal to the suction pressure of the compressor) and of how this pressure falls over time.

Given this knowledge, the following steps can be taken:

- a certain desired overpressure, \( p_a \), can be set in an accumulator 6 which can be installed inside the barrier gas supply pipe 9, or the volume in the supply pipe may constitute an adequate accumulator;
- a controlled supply of barrier gas can be supplied to the compressor module (the first and optionally the third compartment) by means of a flow restriction 71, 71' in the feed pipe 7, 7', 7" between the accumulator 6 and the inlet to the first (optionally also the third) compartment which contains the motor and/or magnetic bearings.

  o The flow restriction is calibrated and/or controlled so that it gives a volume flow of barrier gas into the first (optionally also the third) compartment which at least provides the determined minimum barrier gas velocity through the seal.

  o The flow restriction may alternatively be calibrated and/or controlled so that it gives a fixed mass flow, i.e., a volume flow that increases as a function of falling suction pressure, \( p_s \), of barrier gas in the first (optionally also the third compartment) which at least gives the determined minimum barrier gas velocity through the seal. Such a control is in practice very easy to obtain in a subsea system where \( T_a \) will be constant (equal to the surrounding seawater temperature), by keeping \( p_a \) constant over time irrespective of the value of \( p_s \). A flow meter between the accumulator at \( p_a \) and the flow restriction may then be controlled on a fixed signal, for example, fixed pressure drop through a metering orifice. Mass flow and velocity through the supply pipe 9 will then be constant over time.
Because the velocity $v_g$ of the barrier gas through the seal is low, the pressure drop across the seal will also be small. The pressure inside the motor compartment and the bearing compartment (in other words, the first and third compartments) will only have a very small overpressure, $\Delta p$, (roughly in the range of 0.02 to 0.3 bar) in relation to the suction pressure, $p_s$, of the compressor, and this fact is important for choosing the right value for the pressure in the accumulator, $p_a$, which, for example, can be set at 5 to 50 bars or more above $p_s$. The pressure drop across the seal is therefore neglected in relation to the pressure drop across the flow restriction, and it is therefore in practice the flow restriction alone that determines the flow of barrier gas.

The above-mentioned accumulator 6 does not necessarily need to be a separate tank, but can in certain cases consist of the volume of compressed gas in the pipe ahead of the flow restriction 71, 71'. In many cases it will, however, be advantageous to have a suitably dimensioned pressure tank as an accumulator near the flow restriction, as shown in Figure 1.

The flow restriction 71, 71' may be located in immediate proximity to the feed pipe 7, 7', 7" inlet into the compartment for the motor and seal, or it may be located many kilometres away. For subsea compressors, this may mean that under water and very close to the compressor module there is a flow restriction and an accumulator tank, and that the barrier gas is supplied through a supply pipe in the umbilical from a platform or the shore and at a pressure from the gas supply source that results in the desired pressure $p_a$. For transport over long distances, it is advantageous that the flow restriction takes place close to the compressor module so that the transport of barrier gas takes place at maximum pressure to keep the diameter of the supply pipe as small as possible. In other cases, and especially when the distance between the compressor module and the barrier gas supply source is small, the feed of the desired volume flow of barrier gas can take place at relatively low pressure - just sufficient to overcome the friction in the pipe - direct from the barrier gas source, e.g., with a piston compressor which is set to give desired volume flow through the seal without having an accumulator and flow restriction close to the compressor module, or by the supply of desired volume flow from the shore or a platform in that the accumulator tank and mass flow control are located there. The disadvantage of this will be that because of low gas pressure, i.e., expanded gas volume, the diameters of the feed pipe will have to be relatively large, and will thus be cost-increasing for the umbilical.
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The flow restriction 71, 71' is shown in its most basic form in Figure 5, where a reservoir pressure \( p_a \) is controlled in one of several ways which will be described below, to the desired pressure in the first compartment (10), namely \( p_s + \Delta p \).

Figure 3 shows a variant of the flow restriction 71; 71', where a choke (valve) 79 is controlled on the basis of flow measurement. Figure 2 shows a further variant of the flow restriction 71; 71' which comprises a plurality of chokes 79' in parallel.

Above, the control of the barrier gas flow has been described as a suitable flow restriction 71, 71' installed in the feed pipe 7, 7', 7'' into the compartment or compartments for the motor and magnetic bearings, and this flow restriction restricts between on the one hand a known pressure, \( p_s \), (determined by the reservoir pressure at any given time and the drop in pressure up to the inlet of the compressor) and on the other hand a desired set pressure, \( p_a \), ahead of the flow restriction.

The supply of barrier gas to the "clean" compartments can in practice be controlled in several ways:

- **Constant \( p_s \), constant flow restriction**
  Allow \( p_a \) to remain fixed, and calibrate and set a fixed flow restriction which gives minimum sufficient supply of barrier gas at the start of compression (i.e., at the start-up, in the early phase of the well), and which gives increasing consumption as the reservoir pressure (and thus also \( p_s \) ) drops.
  The volume flow of barrier gas, and thus the mass flow, increases by the square root of the difference \( p_a - p_s \). This very simple control has the drawback that the supply system for barrier gas including the diameter of the supply pipe, must then be dimensioned for such increasing consumption. If, e.g., \( p_s \) falls from 100 bars at the start of compression to 60 bars at the end of compression, and \( p_a \) is maintained at 130 bars, the supply system will have to be overdimensioned by about 50%. But in some cases this may be an acceptable solution in order to obtain simple technical design and operation.

- **Adjust \( p_s \) in step with falling \( p_s \)**
  Down-adjust \( p_a \) as \( p_s \) drops and reduce the supply pressure from the supply source from the shore or a platform correspondingly, so that the supply of barrier gas to the compressor module at any given time gives almost fixed barrier gas velocity through the seal. As shown above, supplied mass flow of barrier gas can be reduced over time with the drop in \( p_s \), so that the velocity of the barrier
gas through the seal will be maintained fixed and with it its ability to shut contaminants out. When $p_a$ is reduced, and thus the pressure level in the umbilical pipe 9 is reduced, there will, if the mass flow is kept constant, be an increase in volume of the gas to be transported that is approximately conversely proportional to the reduction of $p_a$, and the friction in the pipe will increase by the square of the increase in gas volume, that is to say, a quadrupling of the drop in pressure from the barrier gas supply source through the umbilical to the compressor module. In that the supply of the mass flow of barrier gas may also be reduced correspondingly and still maintain the prescribed velocity through the seal, the drop in pressure through the supply pipe will be maintained at an acceptable level over time. A method based on maintaining the prescribed velocity through the seal by downwardly adjusting over time: $(p_a) =$ function $(p_s)$ with fixed flow restriction between the two pressure levels is therefore favourable because such control is very simple as the pressure $p_s$ falls slowly over time. As mentioned above, the supply of barrier gas will be set with a large safety factor influx against ingress of contaminants. Brief transients in temperature and pressure will, as pointed out earlier, not result in harmful ingress even if $(p_a)$ is adjusted downwards.

- **Fixed $p_a$ adjustable flow restriction via valve**
  - $p_a$ is fixed, and the flow restriction 71, 71’ is adjustable, e.g., in the form of one or more valves which can be operated manually (e.g., by a person or using a remote operated vessel (ROV)) so that the flow restriction at certain intervals is increased as $p_s$ drops. The flow restriction may be adjusted by, e.g., measuring the pressure drop across the flow restriction. Alternatively:
  - $p_a$ is fixed, and the flow restriction 71, 71’ is adjustable, e.g. in the form of a hydraulically operated valve which can be controlled manually from a control room so that the flow restriction at certain intervals is increased as $p_s$ drops. The flow restriction may adjusted by, e.g., measuring the pressure drop across the flow restriction.

- **Fixed $p_a$, flow restriction based on measured pressure drop**
  $p_a$ is fixed, and a flow control valve is installed that adjusts the flow on the basis of measured pressure drop across a restriction upstream of the valve. This gives constant mass flow because pressure $(p_a)$ and temperature (for a subsea compressor: seawater temperature) upstream of the valve are approximately
fixed values. The velocity through the seal for the barrier gas will, owing to the
gas expansion, increase as $p_s$ falls. This is only an advantage as regards
protection against the penetration of contaminants through the seal, whilst it
does not represent a disadvantage for the dimensioning of the supply system
which must be dimensioned for the start of the compression state when mass
consumption of barrier gas will be greatest in order to obtain required minimum
velocity through the seal.

- **Fixed $p_a$, flow restriction based on measured pressure drop, and pressure and
temperature in the motor compartment**

  $p_a$ is fixed, and a flow control valve is installed that adjusts the flow on the basis
  of measured pressure drop across a restriction upstream of the valve and the
  measurement of the pressure and the temperature in the motor compartment and
  the magnetic bearing compartment, so that the valve produces a fixed volume
  flow, and thus a fixed desired set barrier gas velocity through the seal. This is a
  possible, but unnecessarily complex control.

When the compressor 2 has been stopped and is not in operation, it will be isolated from
inlet 4 and outlet 5 by respective shut-off valves 41, 51. In such a state, the influx of
barrier gas will stop when in the whole of the compressor module 3 (i.e., first, second
and third compartments) a pressure has built up that is equal to the accumulator pressure
$p_a$. It is important that the pressure inside the compressor module 3 at standstill is
greater than the pressure in the inlet and outlet pipe ahead of the valves so that in the
event of leakage in the valves 41, 51, barrier gas will flow out of the compressor
module, and not liquid and contaminated gas into the module. If during operation of the
compressor $p_a$ is set higher than the highest pressure that can occur in the inlet or outlet
line during standstill, protection against influx during shut-down will automatically be
obtained. This highest pressure that can build up will normally be Wellhead Shut-in
Pressure (WHSP). If during operation $p_a$ is set lower than the pressure that builds up on
the outside of the valves, $p_a$ can be up-adjusted as required, e.g., 1 to 5 bars above the
pressure in the inlet/outlet pipe on feed-in of gas from the supply source (on the shore or
a platform).

Figure 6 shows a mechanical control device for barrier gas which gives constant mass
flow when pressure and temperature upstream of the controller are constant. The
control device includes a valve housing 101 with an inlet 102 which is fluid-connected
to a cavity in the valve housing where a piston 103 is located. As shown in the figure,
there is a fixed opening 104 in the piston, or between the piston and the internal valve housing wall, through which opening gas can flow. The piston 103 is connected to a spring 105 which in turn is fixed in a channel 109 in the valve housing. The piston is also connected to a first pipe 111 that is coaxially arranged with and partly surrounds a second pipe 107. The first pipe 111 is arranged in a separate channel 108 and can (together with the piston 103) slide back and forth over the second pipe 107.

The second pipe 107 is fluid-connected to an opening 110 in the valve housing and a part of the second pipe 107 that is on the downstream side of the piston 103 is provided with one or more openings 106 which thus create fluid connection between said cavity and said opening 110. The opening 110 is, in use, in connection with the control device described above, an outlet for barrier gas into the motor compartment and the compartment for magnetic bearings.

When used as a control device for barrier gas, a barrier gas at a pressure $p_a$ and constant temperature (temperature of surrounding seawater) will be passed into the inlet 102 and into the above-mentioned valve housing cavity. When gas flows through the piston opening 104, a drop in pressure will occur. This drop in pressure will produce a force that will push the piston 103 and the first pipe 111 in the direction of the gas flow (towards the right in Figure 6). The spring 105 will produce a counter-force that will push the piston 103 and the first pipe 111 in the direction of the gas flow (to the right in Figure 6). The spring 105 can be adjusted so that the drop in pressure through the piston opening 104 remains fixed. When the pressure 110 drops, it will result in the flow through the piston opening 104 tending to increase, which will push the piston towards the right. Then, however, the first pipe 111 (which accompanies the piston) will cover a major part of the opening 106 in the second pipe 107, which increases the pressure drop through the opening 106 and keeps the pressure drop through the opening 104 fixed. Conversely, the piston will shift to the left if the outlet pressure 110 increases so that the drop in pressure and thus the flow through 104 remain fixed.

If the flow from the outlet 110 is closed off, for example, on stoppage and shut-down of the compressor module, the piston 103 will be pushed all the way over to the left, which leads to the opening 106 in the second pipe 107 being opened fully. This will lead to the pressure in the whole system, including the compressor module, settling at $p_a$. 
Patent claims

1. An apparatus for controlling the supply of barrier gas to a compressor module comprising a pressure housing (3) containing an electric motor (1) which via a shaft (8) is drivably connected to a compressor (2), wherein the shaft (8) is supported by magnetic bearings (11, 21) and an axial seal (31) around the shaft (8) divides the pressure housing into a first compartment (10) comprising the motor (1) and a second compartment (20) comprising the compressor (2), the second compartment (20) comprising an inlet (4) and an outlet (5) for fluid that is to be compressed, characterised by a line (7) fluid-connected to the first compartment (10) for the supply of barrier gas from a reservoir (6), and a control device (71) in the line between the reservoir (6) and the first compartment (10).

2. An apparatus for controlling the supply of barrier gas to a compressor module comprising a pressure housing (3) containing an electric motor (1) which via a shaft (8) is drivably connected to a compressor (2), wherein the shaft (8) is supported by first magnetic bearings (11) and at least one second magnetic bearing (21) on the opposite side of the compressor in relation to the first magnetic bearings (11), and an axial seal (31) around said shaft (8) divides the pressure housing into a first compartment (10) comprising the motor (1) and a second compartment (20) comprising the compressor (2), the second compartment (20) comprising an inlet (4) and an outlet (5) for fluid that is to be compressed, and wherein a further axial seal (32) is fitted around the shaft (8) between the compressor (2) and the said second magnetic bearing (32), thereby forming a third compartment (30), characterised by a line (7) fluid-connected to the first compartment (10) and a line (7'; 7'') fluid-connected to the third compartment (30), wherein the lines (7, 7'; 7'') are arranged for the supply of barrier gas from a reservoir (6), and respective control devices (71, 71') in the line between the reservoir (6) and the first compartment (10) and the third (30) compartment, respectively.

3. An apparatus according to claim 1 or 2, characterised in that the barrier gas is any gas mixture that is dry, free of particles (as low a particle content as practically possible), and which is inert with respect to the materials in the motor and the magnetic bearings.
4. An apparatus according to claim 3, characterised in that the barrier gas is hydrogen.

5. An apparatus according to claim 1 or 2, characterised in that the reservoir (6) comprises a supply line (9) connected to another gas source.

6. An apparatus according to claim 1 or 2, characterised in that the control device (71; 71′) is a flow restriction means.

7. An apparatus according to claim 1 or 2, characterised in that the control device (71; 71′) comprises a plurality of chokes (79′) arranged in parallel.

8. A method for controlling the supply of barrier gas to a compressor module as disclosed in claim 1 or 2, characterised by, on the basis of required barrier gas velocity \( v_g \) through the seal (31; 32) and the pressure \( p_s \) in the second compartment (20), setting an overpressure \( p_a \) in the reservoir (6), and effecting a controlled supply of barrier gas to at least the first compartment (10) by means of the flow restriction (71; 71′) in the feed pipe (7; T).

9. A method according to claim 8, characterised in that the flow restriction (71; 71′) is controlled so that it provides a volume flow of barrier gas into at least the first compartment (10), which at least provides the necessary barrier gas velocity \( v_g \) through the seal.

10. A method according to claim 8, characterised in that the choke (71; 71′) is controlled so that it gives a fixed mass flow of barrier gas into at least the first compartment (10), which at least provides the required barrier gas velocity \( v_g \) through the seal.
11. A method according to claims 8 and 10, characterised in that the overpressure (p$_a$) in the reservoir (6) is kept constant, and that the flow restriction is controlled on the basis of pressure drop across a metering orifice.

12. A method according to claim 8, characterised in that the flow restriction is kept constant and that the overpressure (p$_a$) in the reservoir (6) is adjusted in response to a change in the pressure (p$_s$) in the second compartment (20).

13. A method according to claim 8, characterised in that the overpressure (p$_a$) in the reservoir is kept constant and the flow restriction is adjusted on the basis of measured pressure drop across the restriction (71; 71').
# INTERNATIONAL SEARCH REPORT

**International application No.**
PCT/NO2006/000341

## A. CLASSIFICATION OF SUBJECT MATTER

**IPC:** see extra sheet

See extra sheet to IPC for the particular details.

## B. FIELDS SEARCHED

**Minimum documentation searched (classification system followed by classification symbols):**

- **IPC:** E21B, F04D, F16C

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

- **SE,DK,FI,NO classes as above**

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

- **EPQ-INTERNAL, WPI DATA, PAJ**

## C. DOCUMENTS CONSIDERED TO BE RELEVANT

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Further documents are listed in the continuation of Box C.

- **X**: See patent family annex.

### Date of the actual completion of the international search
9 January 2007

### Date of mailing of the international search report
11-01-2007

Name and mailing address of the ISA/ Swedish Patent Office

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International patent classification (IPC)

E21B 43/12 (2006.01)
F04D 29/04 (2006.01)
F16C 39/06 (2006.01)

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