ABSTRACT

Based on known Rotational Particle Separator design principles and fluid flow relations a natural gas – liquid separator is designed. A particular feature of this design is that the filter element is freely mounted in bearings and rotates, without the need of a motor, by introducing a swirl in the fluid flowing towards the filter element. The design is particularly suited for operation under high pressures as the rotating filter element is fully contained within a cylindrical pipe. The shaft does not pin through the external wall, so no sealing is required.

Key Words: rotational particle separator, offshore industry, separation.

NOMENCLATURE

<table>
<thead>
<tr>
<th>Symbol</th>
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<td>maximum stable droplet diameter</td>
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<td>dp</td>
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<td>particle or phase diameter</td>
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<td>dp,100%</td>
<td>[m]</td>
<td>diameter of particle or phase collected with 100% probability</td>
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<td>[m]</td>
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The rotational particle separator (RPS) is a novel and patented technique, initially intended for separating solid and/or liquid particles of 0.1 µm and larger from gases, Brouwers [1, 2]. The core component is the rotating filter element (Figure 1), which consists of a multitude of axially oriented channels, which rotate as a whole around a common axis. Particles or droplets flowing in the fluid in a laminar motion are centrifuged to the outer walls of each individual channel while the purified fluid leaves at the exit.

![Figure 1. The rotating filter element](image)

Practical designs of the RPS available in the market include equipment for purifying gases of industrial processes and portable air cleaners for domestic appliances [3, 4]. New developments are made in the area of the offshore industry. As the amount of water extracted from mature oil or gas wells is increasing, water management is becoming an important issue in the production of oil and gas.

Traditionally settling tanks are used to separate the water from the oil or gas. They are however large and often limit the maximum production rate of an offshore platform. For the clean up of separated water prior to discharge frequently high-g devices like hydro cyclones are used. To cater for increasing water cuts in mature fields and to develop new marginal oil fields there is a need for compact equipment that integrates the oil dehydration and water de-oiling steps in a single unit. The rotational particle separator can be used to reduce the size of phase separation on space and weight constraint platforms and possibly can even allow for down-hole separation. A prototype of an oil – water separator based on the RPS principle is already designed and tested [5]. Hydrodynamic performance experiments showed that both angular speed and pressure drop are in accordance with design values. Unsatisfactory separation results were obtained for a specific oil-water mixture. The primary reason was the interfacial tension of the mixture, which was low in comparison with that of mixtures encountered in practice. The results obtained for air bubbles in water were in line with expectations. In the current paper a design is presented for purifying natural gas from wells.

2. DESIGN CONSIDERATIONS

A convenient design of a rotating particle separator for the offshore industry is given in Figure 2. The contaminated fluid (gas/liquid) enters a static swirl generator where it is brought in a rotational motion. Large contaminants are swept out of the flow in the pre-separator. The filter element is brought into rotation by the impulse of the fluid. Due to the centrifugal force, the droplets in the fluid are driven to the wall of the channels where a film is formed. This film breaks at the end of the filter element and larger droplets are created which can easily be separated in the post-separator. If the dispersed phase (oil) is the lighter medium, as in an oil-water mixture, it is swept to the core of the separator. In the case of a liquid-gas mixture, the liquid moves towards the wall. In the current design the outer wall of the filter element is extended into the post separation room in order to reduce the shear stresses on the liquid film at the outer wall of the post-separator. In this way re-entrainment of liquid droplets in the gas flow is avoided. An
important feature of the presented design is that the filter element is freely mounted and does not have a motor to rotate it. So there is no driving shaft pointing through a hole in the wall of the outer casing. Such a hole could be a possible source of fluid leakage at high pressure towards the environment. The housing of the presented design is a thick walled tube which can sustain high pressures and which is aligned to the pipe transporting the fluid.

**Figure 2.** Basic design of a rotational particle separator for the offshore industry

### 2.1. Separation performance

In the filter element the fluid rotates as a rigid body and flows in an axial direction parallel to the rotation axis, such that a laminar flow exists [6]. As the droplets present in the fluid flow are very small in size, inertia forces acting on them are small. It can be assumed that the particles follow the streamlines of the fluid flow, except for the radial direction where, as a result of centrifugal and buoyancy forces, they move relative to the fluid. The radial particle velocity $v_p$ can be calculated from the equilibrium between centrifugal, buoyancy and drag forces acting on the particle. It follows that this radial velocity is given by

$$v_p = \frac{(\rho_f - \rho_p) d_p^2 \Omega^2 r}{18 \eta_f} \quad (1)$$

where $\rho_p$ is the dispersed phase density, $\rho_f$ the fluid density, $d_p$ the particle diameter, $\Omega$ the angular speed, $r$ the radius of the filter element and $\eta_f$ the dynamic viscosity of the fluid. Whether the particles reach the outer wall of the channel depends on the residence time in the channel and the radial distance the particles must travel. Assuming a uniform flow into the filter element, Brouwers [6] derives an expression for the smallest droplet, which can just reach the outer wall with 100% probability in case of an optimal distribution of the axial flow. The desired velocity profile can be accomplished by an appropriate design of the inlet and outlet configuration of the filter element

$$d_{p,100\%} = \sqrt[27]{\frac{27 \eta_f \phi d_c}{(\rho_f - \rho_p) \Omega^2 L_c \pi (1 - \epsilon)(R_o^3 - R_i^3)}} \quad (2)$$

$\phi$ is the volume of the fluid flow through the filter element, $d_c$ the channel height, $L_c$ the channel length, $\epsilon$ the reduction of the effective cross sectional area of the element due to the wall thickness of the channels, $R_o$ the outer radius of the element and $R_i$ is the inner radius.

The smallest fractions, which are separated in the RPS, move with a radial velocity, which is equal to the ratio of the channel height to the channel length times the axial fluid velocity [6]. In practical applications the ratio between the channel height and channel length is very small, in the order $10^{-2}$. Thus the particles move with a radial velocity, which is only 1% of the axial fluid velocity, $u_f$. So, the process of radial migration of the particles to the channel walls is already disturbed when secondary flows of one percent in magnitude of the axial fluid velocity occur in planes perpendicular to the axial channel axis. In order to avoid these secondary flows, the flow in the channels of the RPS has to be laminar. Consequently the axial Reynolds number has to be below 2000

$$Re_{ax} = \frac{\rho_f u_f d_c}{\eta_f} < 2000 \quad (3)$$

From Schlichting [7] it is known that in the case of a tube rotating around its symmetry axis, laminar flow is only ensured if the rotational Reynolds number, defined as

$$Re_{\Omega} = \frac{\rho_f \Omega d_c^2}{\eta_f} < 2000 \quad (4)$$

remains below 108 for values of $166 < Re_{ax} < 2000$. Brouwers [8] shows that these considerations are also valid for tubes rotating around an axis not coinciding with but parallel to their symmetry axis. A further consideration is that tubes should be parallel within 1% to prevent secondary flow due to Coriolis forces [8].
In the pre-separator coarse particles are removed from the flow. Although the streamlines in the pre-separator partly point to the inner radius, large particles can move, as a result of centrifugal and buoyancy forces, in a radial direction opposite to the fluid stream. Particles, which have a radial velocity that is larger than the average radial velocity of the gas stream are driven towards the outer radius of the pre-separator and can be collected there. The diameter of the particles that can reach the wall can be determined analogous to Eq. (2).

The droplets that can reach the walls of the filter element form a liquid film. At the exit of the channels the film breaks up due to centrifugal forces and/or turbulence. According to Hinze [9] the maximum stable droplet diameter in an isotropic, homogeneous turbulent flow can be expressed by

\[ d_{\text{max}} = C \left( \frac{\rho_f}{\sigma} \right)^{-\frac{3}{5}} \varepsilon^{-\frac{2}{5}} \]  

(5)

if the shear forces on the droplets can be neglected. This is the case if the viscous forces inside the droplet are small compared to other forces. \( C \) is a constant and \( \sigma \) denotes the interfacial tension.

The energy dissipation per unit mass \( \varepsilon \) can be estimated by

\[ \varepsilon = \frac{2f u_{\text{mean}}^3}{d_{\text{pipe}}} \]  

(6)

where \( f \) is the friction factor, \( u_{\text{mean}} \) the mean velocity in the pipe and \( d_{\text{pipe}} \) the pipe diameter. It is assumed that the minimum droplet diameter equals half the maximum stable droplet diameter.

Droplets are also generated due to centrifugal forces. The interfacial tension \( \sigma \) gives rise to a surface force that counteracts this deformation process. The droplet size at which break-up occurs can be found by equalling these forces

\[ d_{\text{breakup}} = \sqrt[3]{\frac{6\sigma}{(\rho_f - \rho_p)\Omega^2R_o}} \]  

(7)

The length necessary for the post-separator is found analogous to Eq. (2). The post-separator is an open space receiving flows from individual filter element channels. The angular momentum ensures further phase separation takes place.

### 2.2. Angular momentum

In the swirl generator the angular momentum required to drive the filter element is generated. The angular momentum equals the momentum losses in other parts of the separator. The parts that contribute the most to these losses are the filter element, the pre-separator, the gap between the filter element and the static housing and the bearings

\[ G_{\theta, \text{swirl}} = G_{\theta, \text{fe}} + G_{\theta, \text{gap}} + G_{\theta, \text{decay-pre}} + G_{\theta, \text{bearings}} \]  

(8)

where \( G_{\theta, \text{swirl}} \) is the angular momentum generated in the swirl generator and \( G_{\theta, \text{fe}}, G_{\theta, \text{gap}}, G_{\theta, \text{decay-pre}} \) and \( G_{\theta, \text{bearings}} \) are respectively the loss of angular momentum in the filter element, the gap between the filter element and the static housing, the pre-separator and the bearings.

The general definition of angular momentum is

\[ G_{\theta} = \int_S \rho_f u_f w_f r \ dS \]  

(9)

where \( w_f \) denotes the tangential component of the fluid velocity and \( S \) the cross sectional area. The angular momentum of the flow leaving the filter element is given by [5]

\[ G_{\theta, \text{fe}} = 2\pi \rho_f \Omega (1 - \varepsilon) \int_{R_i}^{R_i} u_f r^3 \ dr \]  

(10)

For a linear axial velocity distribution \( u_f = b r \), with \( b \) a constant, as desired for an optimal performance, it follows that the axial fluid velocity is given by

\[ u_f = b r = \frac{3\phi}{2\pi (1 - \varepsilon) (R_o^3 - R_i^3)} r \]  

(11)

Substitution of Eq. (11) in (10) gives the loss of angular momentum in the filter element

\[ G_{\theta, \text{fe}} = \frac{3\rho_f \Omega \phi}{5} \frac{R_o^5 - R_i^5}{R_o^3 - R_i^3} \]  

(12)

In Li and Tomata [10] a relation is given for the axial decay of angular momentum in a hydraulically smooth tube. With this relation the loss of momentum in the pre-separator can be described as

\[ G_{\theta, \text{decay-pre}} = G_{\theta, \text{swirl}} (1 - 10^{-0.01605(x^*)^{0.8}}) \]  

(13)

where \( x^* \) denotes the ratio between the pipe length and hydraulic diameter of the tube.
The momentum required to turn a cylinder in a static housing is given by [7]

\[ G_{gap} = C_m 0.5 \rho_f \Omega^2 R_o^4 L_c \]  \hspace{1cm} (14)

For a turbulent flow the torque coefficient \( C_m \) is given by \( C_m = 0.019886 Ta^{-0.2} \). The Taylor number \( Ta \) can be written as

\[ Ta = \frac{\rho_f \Omega}{\eta_f} R_o^{1/2} s_g^{3/2} \]  \hspace{1cm} (15)

where \( s_g \) denotes the gap size between the rotating filter element and the static housing.

The loss of angular momentum in the bearings due to friction is dependent on the type of bearing chosen.

### 2.3. Pressure loss

The pressure drop over the filter element is due to friction at the channel walls and the development of a free vortex in the pre-separator to a solid body rotation in the filter element. The pressure loss due to friction in the channels is given by [7]

\[ \Delta P_{friction, fe} = \left( \frac{f \sigma}{d_c} + 1.16 \right) \frac{1}{2} \rho_f u_f^2 \]  \hspace{1cm} (16)

This equation is corrected for entrance effects. In a laminar flow the friction coefficient \( f \) equals \( 64/Re \). The pressure loss due to friction over the control volume can be calculated by considering conservation of mass over the control volume

\[ \Delta P_{friction} = \frac{\Omega (R_o^3 - R_i^3)}{3(R_o - R_i)} \]  \hspace{1cm} (19)

The pressure drop over the swirl generator is due to friction in the small annulus and the introduction of a tangential velocity component. The flow relations of a fluid with viscous effects in a duct of constant cross-sectional area give the pressure loss due to friction in case a compressible flow is considered [11]. In case of an incompressible flow Eq. (16) gives the pressure drop due to friction. The pressure drop caused by introducing a swirl component in the flow can be estimated by

\[ \Delta P_{swirl} = \frac{1}{2} \rho f w_f^2 = \frac{1}{2} \rho f (u_f \tan \alpha)^2 \]  \hspace{1cm} (20)

where \( \alpha \) is the blade angle of the vanes.

### 2.4. Dimensioning

In the test set-up natural gas will be replaced with SF6. This makes it possible to simulate the properties of natural gas, which has a normal working pressure of 40 bar, at a relatively low pressure of 10 bar. Based on the design principles mentioned above, an SF6-water separator was designed for a flowrate of 2335 m³ h⁻¹. The operating temperature is 25 °C and the maximum working pressure is 10 bar. For these conditions the necessary fluid properties are:

- \( \rho_{water} = 1000 \text{ kg m}^{-3} \)
- \( \rho_{SF6} = 50 \text{ kg m}^{-3} \)
- \( \eta_{SF6} = 1.5 \times 10^{-5} \text{ N s m}^{-2} \)
- \( \sigma_{SF6-water} = 7 \times 10^{-2} \text{ N m}^{-1} \)

The cut-off diameter (\( d_{p,100\%} \)) should at least be the same as is reached with current techniques (settling tanks), about 10 μm. For the prototype the length of the filter element is limited due to production limitations to 0.18 m and the channel height had to be at least 1 mm to prevent blockage. Furthermore the total pressure drop over the filter element is limited to 0.5 bar and as the separator will be used in-line, the outer diameter of the filter element should be kept as small as possible. From calculations it followed that the smallest possible value for the outer diameter, such that the pressure drop due to friction alone is beneath 0.5 bar, is 240 mm. In order to keep the total pressure loss at 0.5 bar (with \( D_o = 240 \text{ mm} \)) the angular speed of the filter...
element is limited to 1670 rpm. With this angular speed a $d_{p,100\%}$ of 3 µm is reached, which is suitable for the current application.

In order to achieve this angular speed and compensate for the losses mentioned in the paragraph 2.2 a tangential velocity of 20 m s$^{-1}$ is required. In order to attain this, the flow is diverted to the outside and forced into an annulus with 50° vanes. After the swirl generator the flow is expanded to reduce the axial speed while maintaining the angular momentum. During this expansion large particles that are not able to follow the streamlines, which point inwards, are flung outwardly and can be separated there. The pre-separator is designed such that particles with a diameter of about 25 µm or larger are separated. To achieve this a pre-separation length of 50 mm is necessary.

At the outlet of the filter element the liquid film breaks up due to turbulence and centrifugal forces. From calculations it appeared that droplet break-up due to centrifugal forces in the post-separator dominates. Droplets with a minimum diameter of about 350 µm are formed (Eq. (5)). From Eq. (2) it follows that a post-separation length of about 1 mm is required. Conservatively, the post-separation length is chosen 50 mm.

As the separator is tested in a salty environment, the separator should be corrosion resistant. Therefore the whole prototype is manufactured of stainless steel.

It should be mentioned that in the current design the stability criteria, as mentioned in paragraph 2.1, are not met. This is due to the fact that the outer diameter of the filter element should be kept as small as possible while at the same time a relatively high angular speed is necessary to reach a small cut-off diameter. The influence of this non laminar flow in the channels of the filter element on the separation performance of the RPS has to be examined.

3. THEORETICAL PERFORMANCE

In order to predict the hydrodynamic and separation performance of the separator at other flow rates a model of the separator is made. With this model the total pressure drop over the separator, the angular speed and cut-off diameter of the separator can be calculated as a function of the flow rate. In the model it is assumed that the post-separator has no influence on the performance of the separator.

3.1. Hydrodynamic performance

The angular speed of the separator is the result of the equilibrium between the produced angular momentum in the swirl generator and losses in the other parts of the separator. This equilibrium is given Eq. (8). The produced momentum in the swirl generator and the losses in the pre-separator are only a function of the flow rate. The other terms in Eq. (8) however also depend on the angular speed. This implies that Eq. (8) can only be solved by an iterative calculation. The result of these calculations is given in Figure 3. For flow rates beneath 70 m$^3$ h$^{-1}$ the filter element does not rotate. This is due to the static bearing friction.

![Figure 3. Hydrodynamic performance natural gas–water separator. Speed (continuous line) and pressure drop (dotted line) as a function of flow rate.](image-url)

The main pressure losses over the separator occur in the filter element and in the swirl generator. In both parts there are pressure losses due to friction (Eq. (16)) Besides these contributions also the pressure losses due to the introduction of angular momentum in the swirl generator (Eq. (20)) and due to the difference in the tangential velocity component prior to and in the filter element (Eq. (17)) must be taken into account. The friction losses in the channels of the filter element contribute the most to the total pressure drop. In Fig. 3 the total pressure drop over the separator versus flow rate is depicted. It can be seen that the total pressure drop at the design condition is about 0.8 bar.
3.2. Separation performance

A characteristic parameter describing the separation performance of the separator is the cut-off diameter $d_{p100\%}$ of the filter element. This cut-off diameter can be calculated by Eq. (2). The cut-off diameter depends on the flow rate as well as on the angular speed. The relation between these two quantities is already calculated in paragraph 3.2. With this relation the cut-off diameter can be calculated and the result is given in Figure 4. It can be seen that for flow rates smaller than the design flow rate the separation performance deteriorates. For the flow rates beneath 70 m$^3$ h$^{-1}$ the filter element does not rotate and thus no particles are separated.

![Figure 4. Separation performance natural gas – water separator](image)

4. FUTURE WORK

In order to validate the design calculations a prototype of the natural-gas water separator is currently built. This prototype will be tested at CDS Engineering, an innovative orientated company, operating as a process designer and equipment supplier of 2-phase and 3-phase separation equipment for the offshore industry. Also field tests in cooperation with the NAM will be carried out. Besides this experimental work, the behaviour of the gas and liquid flow in the RPS will be analysed theoretically. It concerns among other things the description of the behaviour of the liquid film in the filter element. Due to the high gas flow velocities in the channels of the filter element, high shear forces are exerted on the liquid film. This may cause re-entrainment of liquid into the gas flow, which deteriorates the separation performance of the RPS.

ACKNOWLEDGMENTS

This project is supported by the Dutch E.E.T. programme, nr. EETK99138.

REFERENCES