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In-line Centrifugal Separation of Dispersed Phases

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A new device—the rotational particle separator (RPS)—is compared to the cyclone for the removal of ultrafine particles, such as cryogenically condensed contaminant droplets from natural gas. The comparison focusses on identifying for each configuration the smallest size of particle which has a 50% chance of being removed from the gas stream. Whereas a cyclone can only separate 1 μm droplets at very low-throughputs on the order of 1 MMscfd, at the same energy consumption and device volume, the rotational particle separator removes droplets of that size at throughputs of 300 MMscfd. The advantage of the rotational particle separator, therefore, lies in its compact separation potential for full-scale industrial gas well throughputs. *AIChE J,* 53: 374–380, 2007
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Introduction

Separation of liquid dispersions from another fluid is one of the most important unit operations in the oil and gas business.[1,2] The dispersions are either oil/water or liquid/gas mixtures. The latter category has until now mainly been concerned with removal of water droplets and light hydrocarbon liquids from gas streams. A new area is where ultra fine CO_2 rich mists are created by expansion. This is a novel method which removes natural gas contaminants by condensation and enables access to gas reserves, where the CO_2 or H_2S content is too high to be removed by traditional amine methods.[3]

Cyclones are standard for liquid/gas separation in hydrocarbon processing plants.[4,5] These cyclones are used for water and condensate removal but have *not* been applied for removing condensed contaminants, such as CO_2 or H_2S. This is because cyclones can only handle condensing droplet sizes[6–8] above 15 μm, and large extended highly cooled droplet growth pipes would be required for removing contaminants.

When condensing contaminants from natural gas, the droplet size of the dispersed contaminant is small—typically on the order of 1 μm. It is well known in laboratory chemical applications that microcyclones *can* separate such small droplets, but then the flow is very small and orders of magnitude less than the flow in gas well applications.[6,7] Droplet size and process intensification are, thus, the drivers for development of a new centrifugal technology.

In this study we examine a single device which by contrast can be placed in a pressurised pipeline rather than requiring a large-pressure vessel which would be required for banks of multiple cyclones. It is, thus, much smaller than a cyclone bank, and as we will show can separate smaller droplets. This device developed from a study of the gas centrifuge[9–12] with the addition of high-pressure condensation.[13,14] The objective of this

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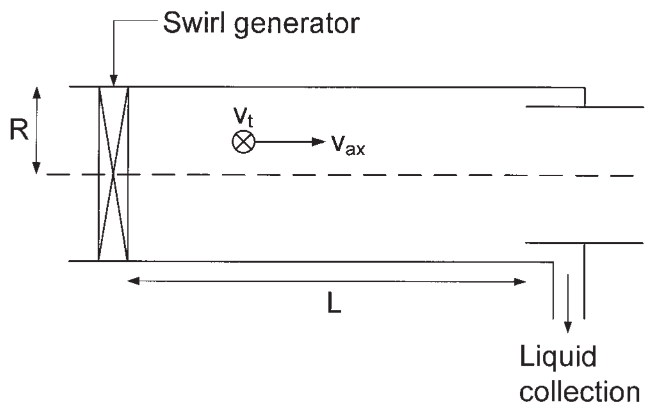

Figure 1. Axial cyclone.

study is quantification of the improvement in the rotational particle separator (RPS) with respect to a cyclone.[15,16] The principle is always to get the maximum gravitational effect but design for a small compact volume. At equal separator volume the RPS can remove smaller particles (ca. 1 μm), and for removing larger particles, the separator volume is a lot smaller than for cyclones.

We identify independent process parameters in which the performance of both inline centrifugal separators can be expressed. These parameters are the residence time τ which defines the separator size and thus the capital cost, and the specific energy consumption ε defining operating cost. These are compared for the same throughput duties defined by feed flow Q. The Axial cyclone section evaluates these parameters for the axial cyclone, and this is also done for the RPS in the Rotational Particle Separator section. The two devices are compared and discussed in the Discussion section.

Axial Cyclone

The axial cyclone, also known as vortex tube or uniflow cyclone, consists of a stationary cylindrical pipe which contains at the entrance stationary vanes or blades: Figure 1. Fluid which enters the pipe and passes through these blades attains a swirling motion. Dispersed phase entrained in the fluid acquires this swirling motion as well. Having a density which is higher than the density of the carrier fluid, the dispersed phase will be subjected to a centrifugal force which causes it to move radially toward the cylindrical wall. It leaves the device via outlets so situated at the end of the pipe constituting the axial cyclone.

The dispersed phase is represented by spherical particles of diameter d_p and density ρ_p. The velocity by which the particles move radially can be calculated from a balance between the centrifugal force and the fluid force, which is exerted on the particle in case of motion relative to the surrounding carrier fluid. For particles with diameters ranging from about 0.5 micron to 25 micron, the fluid force can be described by Stokes flow.[17] For smaller and larger particles Cunningham and Reynolds' number corrections have to be introduced,[18] respectively,—however, at a diameter of 1 μm, the effect is only ca. 10%—omitting it is a more conservative approach. The radial migration velocity of a particle can then be described as

$$v_{\mathrm{TC}} = \frac{(\rho_p - \rho_f) d_p^2 v_t^2}{18\mu r} \tag{1}$$

where ρ_f is density of carrier fluid and μ is dynamic viscosity, v_t tangential velocity and r radial position.

Detailed investigations of swirling flows in pipes[19] have shown that the tangential velocity v_t changes its radial shape with axial distance from the point where the swirl is initiated. While initially the radial profile may be more like that of a free vortex, with axial distance tangential velocity profiles evolve more toward that of solid-body rotation. At the same time the strength of the swirling motion decays as a result of wall friction. In this analysis we shall assume that v_t is constant with respect to r. This is a compromise between solid body rotation and the free vortex. The axial decay is described by an exponential function in accordance with experimental observations[19,20]

$$v_t = v_{t0}\ \exp\left(-\frac{z}{R}\beta\right) \tag{2}$$

where z is axial distance, R radius of cyclone wall and β an empirical factor which for conditions existing in cyclones is about 0.05. Furthermore, it is assumed that the axial velocity of the carrier fluid in the cyclone is constant with respect to radius $v_a = v_{a0}$. Again this is a reasonable compromise for the various profiles encountered in measurements. Tangential and axial velocity of dispersed profiles are to first-order equal to those of the carrier fluid. The axial position of a particle is, thus, given by $dz/dt = v_{a0}$, which upon integration yields $z = v_{a0t}$, where it is assumed that at time $t = 0$ the particle is at the entrance of the cyclone, that is, $z = 0$. While a particle follows to first-order the gas in the axial and tangential directions, it will migrate in the radial direction according to Eq. 1. Noting that its radial position r is given by $(d/dt)r = v_m$, implementing Eq. 1 and Eq. 2, and using $z = v_{a0}t$ to eliminate z, gives

$$r\frac{dr}{dt} = \frac{(\rho_p - \rho_f) d_p^2 v_{t0}^2}{18\mu}\ \exp\left(-\beta\frac{v_{a0}}{R}t\right) \tag{3}$$

Upon integrating one obtains

$$r^2(t) = \frac{2(\rho_p - \rho_f) d_p^2 v_{t0}^2}{18\mu}\left(\frac{R}{\beta v_{a0}}\left(-\exp\left(-\beta\frac{v_{a0}}{R}t\right) + 1\right)\right) + r^2(0) \tag{4}$$

where $r(0)$ is the radial position of the particle at the entrance of the cyclone. The axial velocity of carrier fluid and particles is taken constant with respect to radius. It implies that 50 % of particles which enter the cyclone will be present in the area $r > R/\sqrt{2}$, 50% in the area $r < R/\sqrt{2}$. The diameter of the particles which has a probability of 50% of reaching the collecting wall in a cyclone of length L can therefore be calculated from (Eq. 8) by taking for $r(t) = R, r(0) = R/\sqrt{2}$ and $t = L/v_{a0}$. It results in

$$d_{p50} = \left\{\frac{18\mu R}{4(\rho_p - \rho_g) v_{t0}^2}\ \frac{\beta v_{a0}}{1 - \exp\left(-\frac{\beta L}{R}\right)}\right\}^{1/2} \tag{5}$$

Energy consumption occurs through the pressure drop the fluid undergoes when flowing through the cyclone. One can assume that swirl induced at the entrance (and associated radial pressure buildup) is eventually lost: the irreversible pressure loss can be taken equal to ρv_{t0}^2. The total energy loss can be calculated by integrating over all radial positions

$$\dot{E} = \int_0^{2\pi} \int_0^R \rho v_t^2 v_{ax} r dr d\theta \tag{6}$$

For v_{t0} and v_a constant with respect to r, the result is $\dot{E} = \rho v_{t0}^2 Q$: where Q is volume flow through the cyclone $Q = v_a \pi R^2$. Energy consumption per unit mass flow $\varepsilon = \dot{E}/(\rho_f Q)$ then reads as $\varepsilon = v_{t0}^2$. The pressure drop over the cyclone relates to ε as $\Delta p = \varepsilon \rho_f$.

The amount of swirl imposed at the inlet of the cyclone is limited. For large values of the ratio of swirl velocity to axial velocity a reverse flow in the center of the cyclone will appear, which leads to mixing of separated material. To avoid this, the swirl velocity is limited to a magnitude which is twice the axial velocity: $v_{t0} = 2v_{a0}$.

The residence time of the fluid in the cyclone is given by $\tau = L/v_{a0}$.

It is now possible to express d_{p50} given by Eq. 5 as a function of the volume flow of carrier fluid Q, which is to be filtered

$$d_{p50}^2 = \frac{9\mu}{(\rho_p - \rho_f)\pi} \frac{Q}{\tau \varepsilon^{3/2}} \frac{\gamma}{1 - e^{-\gamma}} \tag{7}$$

where

$$\gamma = \beta \left(\frac{\pi}{8}\right)^{1/2} \frac{\tau \varepsilon^{3/4}}{Q^{1/2}} \tag{8}$$

In the above relation two important design parameters occur: the residence time τ which is a measure for the size of the device, and, thereby, for the investment costs, and the energy consumption ε which determines to a large extent the operating costs. Using these parameters it is possible to compare the separative performance of the cyclone with that of the rotational particle separator analyzed in the subsequent section.

Rotational Particle Separator

The inline version of the rotational particle separator (RPS) is an axial cyclone within which a rotating separation element is built.[20,21] (Figure 2). The rotating element consists of a multitude of axially oriented channels of diameter of about 1 to 2 mm. The element is freely mounted in bearings, and rotates as a result of the torque impelled by the swirling flow entering the element. As the rotating element is entirely contained within a cylindrical stationary pipe the device is also suitable for operating under high-pressure. Because of the small radial dimensions of the channels, particles can be separated whose sizes are much smaller (about a factor 5) than those of the cyclone, all this at equal external dimensions and energy consumption.

The separation process taking place in the channels of the RPS is equal to that in the cyclone. The radial migration velocity of a particle can be expressed by Eq. 1, where in this case r is the radial position of the channel. Channel heights are very small: therefore, the radial path of a particle in the channel can be calculated disregarding variations of velocities with r. The diameter of the particle which is collected with 50% probability can be calculated by considering the radial position of the channel R_{50} at which a cut of 50% of the throughput occurs. The filter

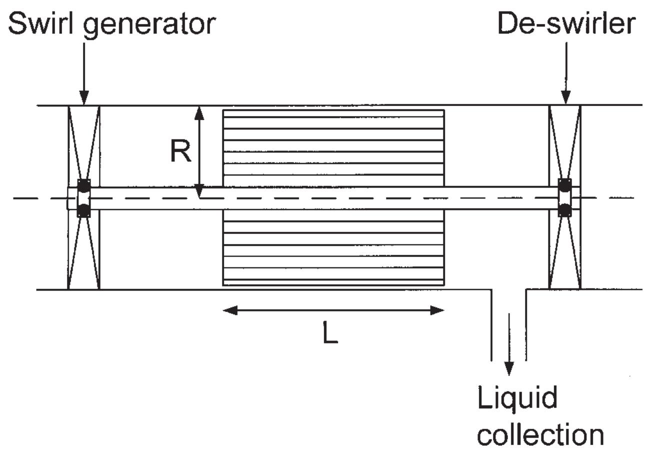

Figure 2. Rotational particle separator.

element has an inner radius of δR, in which case

$$R_{50} = \frac{R}{\sqrt{2}}(1 + \delta^2)^{1/2}$$

Now d_{p50} can be calculated by considering the path of the particle which enters a channel located at $r = R_{50}$. Tangential velocity equals Ωr where Ω is the angular speed of the rotating element; axial velocity v_a is constant with respect to r so that $v_a = Q\left(\pi R^2\left(1-\delta^2\right)\right)^{-1}$. One then obtains

$$d_{p50}^2 = \frac{9\sqrt{2}\mu d_c}{(\rho_p - \rho_f)(1+\delta^2)^{1/2}(1-\delta^2)} \frac{Q}{\pi \Omega^2 L R^3} \tag{9}$$

where d_c is the aperture size of the channel.

The energy consumption of the RPS consists of two parts. First there is the energy loss because of rotation. Downstream of the filter element a stationary de-swirler is installed (Figure 2), which recovers about half of the energy associated with swirl. Therefore, the irreversible pressure drop along each streamline passing through the rotating filter element is taken equal to half of that of the cyclone: 1/2 ρv_t^2 where $v_t = \Omega r$.

The second source of energy consumption is pressure drop in the small-sized channels of the RPS. Total energy consumption per unit volume flow, or irreversible pressure drop, can be expressed as $\Delta p_{\mathrm{RPS}} = \Delta p_t + \Delta p_{ch}$ where $\Delta p_t = \frac{1}{4}\rho(\Omega R)^2 (1 + \delta^2)$, is irreversible pressure drop because of swirl evaluated over the whole filter element according to Eq. 6, taking $v_t = \Omega r$ and $v_a = v_{a0}$. Irreversible pressure drop over the channels can be described according to $\Delta p_{ch} = \rho v_a^2 f L/(2d_c)$, where f is the friction factor. Here we disregard the extra pressure losses due to entrance effects, as well as blockage of channels—in practice these amount to <10% of the channel pressure drop.[20] We have shown that as liquid builds up on the channel walls, shear stress exerted on the liquid is large enough to tear the liquid stream into large separable droplets downstream of the rotational particle separator.[3] (Axle filters and bearing systems are specifically designed to withstand this shear stress.) We show below that the channel pressure drop itself is still only 1/7 of the total pressure drop even when partial restriction occurs due to liquid formation in the channels. The total pressure drop has its origin predominantly in the swirl term, and, thus, *not* in any liquid blockage. The correction due to entrance and possible blockage is, therefore, extremely small

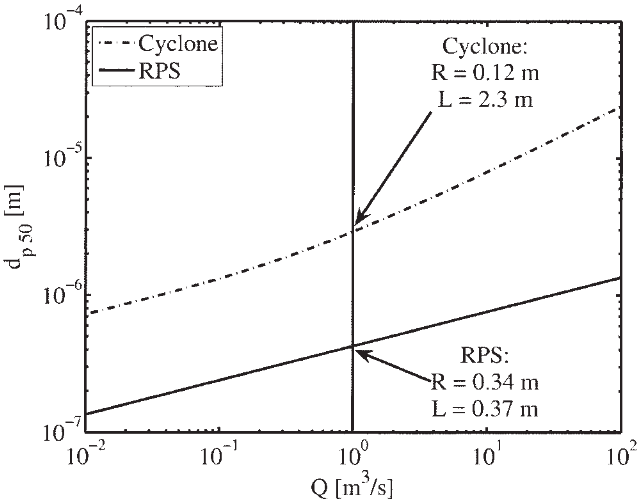

Figure 3. Separation performance d_{p50} as a function of volume flow.

System is water droplets in air. $P = 1$ bar (although the result hardly changes with pressure); $T = 20°C$. Residence time $\tau = 0.1$s; specific energy consumption 2 kJ/kg—equivalent pressure drop $\Delta p = 2500$ Pa (0.025 bar).

and we, therefore, restrict our channel losses to the friction factor, which depends on Reynolds number of the flow. For $Re < 10^5$ one can take the Blasius formula[18] $f = 0.316\, Re^{-0.25}$.

With the presented formulas we can transform Eq. 9, depending on L, R, v_{a0} and Ω, to a new equation in which d_{p50} depends on Δp_t, Δp_c, τ, and Q

$$d_{p50}^2 = \frac{9\mu d_c^{5/6}(1+\delta^2)^{1/2} f^{1/6}}{2^{5/3}(\rho_p - \rho_g)\pi^{1/2}(1-\delta^2)^{1/2}} \frac{\rho^{7/6}}{(\Delta p_{ch})^{1/6}\Delta p_t} Q^{1/2}\tau^{5/6} \tag{10}$$

It is seen that d_{p50} becomes smaller (better separation performance) with increasing dissipations Δp_t and Δp_{ch}. The best situation is one where the ratio of both dissipations $x = \Delta p_{ch}/\Delta p_t$, has a value such that the total dissipation is a maximum and the value d_{p50} is a minimum as well (maximum separation performance). To calculate this ratio it is noted that according to Eq. 10 $d_{p50}^2 \sim ((\Delta p_{ch})^{1/6}\ \Delta p_t)^{-1} = ((\Delta p_{RPS})^{7/6}(x)^{1/6}(1-x))^{-1}$, which reaches a minimum for $x = 1/7$, so that $\Delta p_{ch} = 1/7\ \Delta p_{RPS}$ and $\Delta p_t = 6/7\ \Delta p_{RPS}$.

It is noted that the minimum value of d_{p50} is only slightly sensitive to deviations of x from its optimum value of 1/7. For example a deviation of x of 20% from its optimum value leads to an increase of d_{p50} of 0.2%.

The expression for d_{p50} now becomes

$$d_{p50}^2 = \frac{9\mu d_c^{5/6} f^{1/6}}{2^{5/3}(\rho_p - \rho_g)\pi^{1/2}} \left(\frac{1+\delta^2}{1-\delta^2}\right)^{1/2} \varepsilon^{-7/6} Q^{1/2} \tau^{-5/6} \tag{11}$$

where $\varepsilon = \Delta p_{RPS}/\rho_f$ and

$$f = \left(0.316\left(\frac{2}{7}\right)^{-1/12} \rho_f^{-3/12} d_c^{-1/3} \mu^{1/4} \varepsilon^{-1/12} \tau^{1/12}\right)^{12/11} \tag{12}$$

From Eq. 11, it is seen that d_{p50} of the RPS becomes smaller with d_c the channel aperture size. In practice, however, d_c is limited by fabrication and operational requirements to about 1 mm. It is further seen that d_{p50} decreases with δ. The dependence is weak; in practice the inner radius of the filter element is taken about 1/2 to 1/3 of the outer radius. For given values of d_c and δ, Eq. 11 specifies d_{p50} of the RPS vs. flow Q with residence time τ, and specific energy consumption ε as parameters.

Discussion

In our foregoing analysis we identified that performance for rotational separations can be expressed in terms of three variables which determine capital and operating costs. These were the flow rate (Q), residence time (τ) and specific energy consumption (ε). In the following discussion we will examine the effect of varying each of these three variables in turn *while keeping the other two constant*, on the separative performance as measured by the median size of particle, which can be removed from the flow. These three cases refer to noncompressible liquids dispersed in a gas. Finally we will conclude with the case of a compressible liquid dispersed in a gas which is the relevant case for removal of contaminants from natural gas. In all cases, in order to derive the scaling and sizing information appropriate to the two separators concerned, the reader should refer to the dimensioning formulas listed in the Appendix.

Equations 7 and 11 can be used to compare the separative performance of the cyclone and RPS as a function of volume flow Q at equal values of specific energy consumption ε and residence time τ (= building volume). The result for separation of water droplets ($\rho \approx 1000$ kg/m^3) from air at ambient conditions (P = 1 bar, T = 20° C, and $\rho = 1.2$ kg/m^3) is shown in Figure 3, which displays d_{p50} for both units as a function of flow in actual cubic meters per second. Residence time and specific energy consumption are set at values commonly found in practice of $\tau = 0.1$ s and $\varepsilon = 2$ kJ/kg, which corresponds to $\Delta P \sim 2{,}500$ Pa

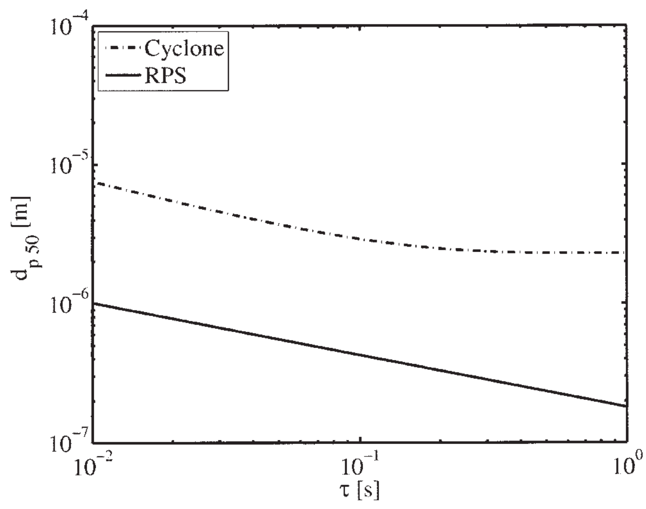

Figure 4. Separation performance d_{p50} as a function of residence time.

System is water droplets in air. $P = 1$ bar (although the result hardly changes with pressure). Volume throughput is 1 m^3/s (equivalent to 100 MMscfd at 30 bar); specific energy consumption 2 kJ/kg—equivalent pressure drop $\Delta p = 2{,}500$ Pa (0.025 bar) at 1 bar absolute pressure.

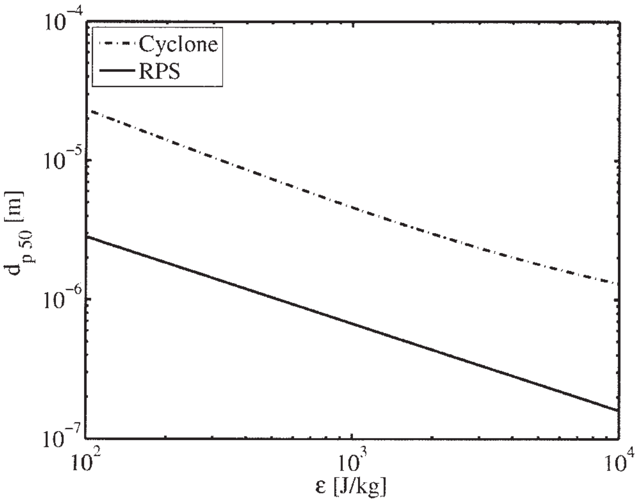

Figure 5. Separation performance d_{p50} as a function of specific energy consumption.

System is water droplets in air. p = 1 bar (although the result hardly changes with pressure); volume throughput is 1 m^3/s (equivalent to 100 MMscfd at 30 bar); residence time τ = 0.1s; equivalent pressure drop of ε = 2 kJ/kg: Δp = 2,500 Pa (0.025 bar) at 1 bar absolute pressure.

at $P = 1$ bar, and $T = 20°C$. The RPS channel diameter and inner/outer radius ratio are taken equal to the standard production values of $d_c = 1.5$ mm, and $\delta = 0.5$. As can be seen, the smallest particles collected by the RPS are an order of magnitude smaller than those collected by the axial cyclone. To resolve such small particles, a conventional cyclone would require extremely fast supersonic peripheral gas velocities. This is associated with very high energy consumption—however, note that we have specifically excluded this regime: we have restricted the specific energy consumption and limited the swirl number to 2—this ensures that tangential velocities stay an order of magnitude smaller than acoustic velocities. In fact, the RPS can take a throughput 10,000 times higher than a cyclone when it comes to collecting 1 μm sized droplets.

Figure 4 and 5 show the results for varying values of residence time and energy consumption, respectively. Throughput is fixed here at a moderate value of 1 m^3/s. The results for the RPS are in both cases again an order of magnitude better than that of the axial cyclone. Both units show a large increase in performance for higher-residence time and energy consumption.

Although Figure 3–5 are plotted using the properties of air at ambient conditions, they are approximately valid for all gases at a large pressure range, since the properties of the carrier fluid are of minor influence as follows from Eq. 7 and 11. The influence of pressure is seen in both cases to be inversely proportional to the difference between the particle and gas-phase densities, that is, ρ_p–ρ_g. For a carrier gas pressure of 100 bar the density difference term in these equations still has approximately 90% of the value it has for a carrier gas at ambient pressure, that is, kg/m^3 instead of 1,000 kg/m^3. Using Eqs. 7 and 11, this leads to an increase in d_{p50} of about 6%, which is only marginal. When the carrier fluid is a gas, then this inverse density term is at best a second-order correction for the range of pressure conditions normally encountered in practice—the justification for this is the range of surface facility inlet manifold pressures in gas processing plants—these are typically in the range 70–130 bar. Thus, the performance of the separator at 1 m^3/s can correspond to either a gas flow at 1 bar or at 100 bar equivalent to 100 normal m^3/s. (The latter corresponds in imperial production units to 300 MMscfd—a good sized gas well.)

The only situation where the inverse term *is* important, is when the carrier fluid is a liquid instead of gas at ambient pressure. An example is the application of oil-water separation when decrease of the pressure difference term is of course substantial. This leads to a shift upward, that is, larger d_{p50}, for the results in Figure 3–5 which remain universally applicable. This effect is equally large for both the cyclone and the RPS as follows from Eqs. 7 and 11.

In addition we note that for application to contaminated gas, that condensed contaminant (CO_2 and H_2S) fluid droplets are significantly more compressible than water. This compressibility actually means that it is favorable to carry out the processes at higher-pressure, because any increase in the gas density is more than offset by the significant increase in the fluid density in Eq. 11. This is all the more so for contaminated gas since the condensed waste liquid CO_2 contains typically ca. 10% methane gas which makes the liquid phase even more compressible. The relative advantage of using the RPS system for cleaning gas is seen in Figure 6, where the effect of pressure is seen to be large at the levels of flow associated with gas well production. For this example of industrial relevance the units of flow are the gas production industry norm: MMscfd (millions of standard cubic feet per day —for SI conversion 1 MMscfd = 0.33 norm m^3/s.)

The results for both units are a function of volume flow and *not* of mass flow. This means that an increase in pressure will not only lead to a smaller unit, at constant-residence time, but also to a better separation performance at equal mass flow. For instance, from Figure 3 it can be seen that for air at ambient pressure carrying water droplets the d_{p50} of a well designed RPS, with a throughput of 1 m^3/s is equal to $d_{p50} \sim 0.4$ μm. Increasing the pressure to 10 bar leads, at equal mass flow and

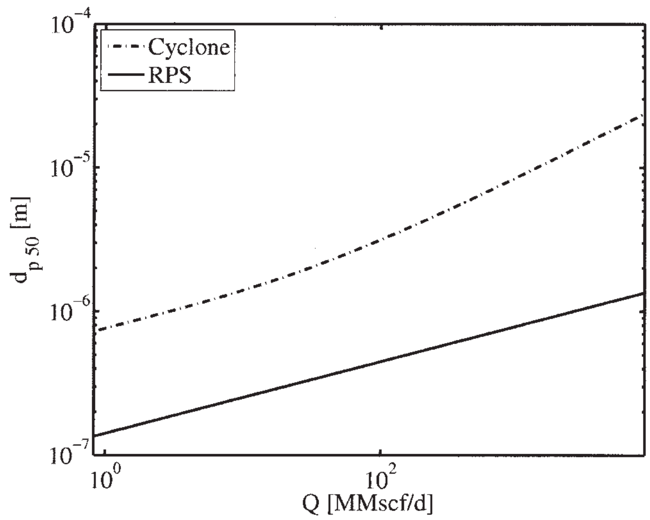

Figure 6. Separation performance d_{p50} as a function of normal (That is, molar) flow for a 50/50 CH_4/ CO_2 separation.

System at p = 27 bar, T = −47°C; residence time τ = 0.1 s; specific energy consumption ε = 2 kJ/kg; pressure drop Δp = 0.7 bar.

for an ideal gas, to a volume flow of 0.1 m^3/s. Figure 3 shows that for this flow the d_{p50} for an RPS reduces to 0.2 μm. At the same time the volume of the RPS has decreased equally with the volume flow, to 0.1 times the original size. If the original separation performance, before the pressure increase, was satisfactory, then one could also, by increasing the pressure, reduce the residence time or energy consumption, without affecting the separation performance. This leads to an even smaller device or to lower operational costs.

Application of the RPS implies the introduction of rotating equipment. Complications involved with rotation, however, are limited. Peripheral speeds of the rotating body are limited, typically to 50 m/s, which is far below the speeds where structural integrity might become problematic. In fact, the speeds are limited because of constraints on energy consumption (value of ε). Sealing problems of shafts penetrating through stationary walls do not exist either. The RPS is internally driven by the swirl of the gas entering the rotating body. External drive by a motor is not necessary. If deemed necessary, a magnetic externally coupled drive can be used thereby making sealing problems obsolete. In summary, the complexities involved with rotation are rather modest; they are outweighed by the advantages—the possibility to collect micron-seized particles in a device of limited dimension.

Conclusions

1. The separation performance of inline centrifugal separators expressed as the diameter of the particle that has 50% chance of separation, d_{p50}, is a function of three independent process parameters: residence time, specific energy consumption and volume flow. These three parameters can be used as a basis for comparison between different separator configurations.
2. The rotational particle separator (RPS) is able to separate an order of magnitude smaller particles than the axial cyclone is able to, at equal-residence time, specific energy consumption and volume flow.
3. At the same droplet diameter, for example, for $d = 1$ μm in Figure 5, then the energy consumption is an order of magnitude less for the RPS than for a cyclone.
4. If the carrier fluid is a gas then the separation performance of a centrifugal separator as a function of volume flow varies only slightly with pressure. An increase in operating pressure leads at equal mass-flow rate to a smaller volume flow rate, and, thus, to a better separation performance or smaller equipment.

Notation

d_c = diameter of RPS channels, m
d_{eff} = effective diameter of RPS channels, m
d_p = particle diameter, m
d_{p50} = diameter of particle which has 50% chance of being separated, m
$\dot{E}$ = energy consumption rate, J/s
f = friction factor
h = height of triangular RPS channel, m
L = separator length m
$\dot{m}$ = mass flow rate kg/s
p = pressure, Pa
Q = volume flow rate, m^3/s
R = separator radius, m
R_{50} = cut-radius, m
Re = Reynolds number
R_{id} = inner-filter diameter of RPS, m
R_{od} = outer-filter diameter of RPS, m
R_w = wall radius, m
s = tangential position, m
t = time, s
T = temperature, K
v_{ax} = axial velocity m/s
v_{ax0} = axial velocity at t, 0 m/s
v_r = radial velocity, m/s
v_t = tangential velocity, m/s
v_{t0} = tangential velocity at $t = 0$ m/s
x = channel dissipation to total dissipation ratio (RPS)
z = axial position, m
μ = dynamic viscosity, Pa·s
τ = residence time, s
θ = angular position rad
ρ = mass density, kg/m^3
ε = specific energy consumption, J/kg
β = axial velocity decay parameter
δ = inner to outer RPS filter-diameter ratio
Ω = angular velocity rad/s
ρ_f = mass density of the carrier fluid, kg/m^3
ρ_p = mass density of the particles, kg/m^3
Δp_{ch} = pressure drop in RPS channels, Pa
Δp_{cyc} = total-pressure drop of the cyclone, Pa
Δp_{RPS} = total-pressure drop of the RPS, Pa
Δp_t = pressure drop due to RPS swirl generator, Pa

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Appendix: Formulas for Equipment Dimensions

Cyclone

Volume: $V = Q\tau = \pi R^2 L$

Length: $L = 1/2\varepsilon^{1/2}\tau$

Radius: $R = 2^{1/2}\pi^{-1/2}Q^{1/2}\varepsilon^{-1/4}$

Tangential velocity: $v_{t0} = \varepsilon^{1/2}$

Axial velocity: $v_{ax} = \frac{1}{2}v_{t0} = \frac{1}{2}\varepsilon^{1/2}$

RPS

Volume: $V = Q\tau = \pi R^2(1 - \delta^2)L$

Length: $L = \left(\frac{2d_c\tau^2\varepsilon}{7f}\right)^{1/3}$

Radius: $R = \left(\frac{7}{2}\right)^{1/6}\pi^{-1/2}(1 - \delta^2)^{-1/2}f^{1/6}d_c^{-1/6}Q^{1/2}\varepsilon^{-1/6}\tau^{1/6}$

Tangential velocity: $v_{t0} = \left(\frac{24\varepsilon}{7(1+\delta^2)}\right)^{1/2}$

Axial velocity: $v_{ax} = \left(\frac{2d_c\varepsilon}{7f\tau}\right)^{1/3}$

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