Low Energy Consumption of High-Speed Centrifuges

A sharp reduction of the energy consumption of a high-speed centrifuge was obtained following several design changes, while still maintaining separation capacity. This reduction is necessary for making some applications commercially interesting, e.g., large-volume flow rates in the growing biopharma industry. The reduction was achieved by modifying flow paths of the rotor, i.e., reducing the outlet radius to minimize angular momentum losses and lowering the pressure drop of the internal flow. Further, removing air outside the rotor reduced aerodynamic losses, and using a direct drive motor, losses encountered with gear or belt drives were eliminated. This way, an average energy reduction of 50% was obtained.

Keywords: Disc-stack centrifuge, Energy reduction, High-speed centrifuge, Separator

1 Disc-Stack Separation

The present work is the outcome from a larger study (PRODIAS – Processing Diluted Aqueous Systems), motivated by the process industry striving for long-term sustainability and efficiency in order to ensure competitiveness [1]. Conventional centrifugal separation is particularly useful for cell removal from a fermentation broth at large volume flow rates as it magnifies the G-force of static settlers by factors of thousands, resulting in an equipment of small footprint and low investment cost relative to its capacity. These machines rely on mechanical separation of phases of different density and are made effective by (i) increasing the G-force due to fast rotation of a rotor bowl, in which separation takes place, as well as (ii) reducing the settling distance and enlarging the settling area by using a conical disc stack inside the rotating bowl. However, along with higher separation efficiency by means of increasing rotational velocity, follows also higher energy consumption, due to internal pressure drop, losses of angular momentum of the feed passing through the bowl, and external losses like windage from the exterior of the rotor and mechanical drive losses in gears and bearings.

High-speed centrifuges have been used in industry for more than a 130 years (thus excluding hand-driven centrifuges, or those driven by rope or a small steam turbine, later by belt and worm gear) and their features are well described in the literature [2–5], similar to the related decanter centrifuges [6]. However, the issue of its energy consumption has attracted surprisingly little attention, despite the machines requiring considerable amount of energy per flow rate. Further to this somewhat surprising state, some other branches of engineering, like pumps and motors, have been more exposed to reduction demands, and have been more public with reductions gained. Now, however, along with increased demands for being energy-efficient in every step of an industrial process, and with growing global trends/opinions that energy use is not only a cost, but also a long-term environmental issue, depending on the energy source, an increased customer need in low-energy high-speed centrifugation is obvious.

One early important step reducing power consumption was the introduction of paring-disc outlet, as opposed to having open discharge outlets, which took place on larger scale in the late 1920s [7]. The reason at the time, though, was from a defoaming point of view rather than from an energy consumption point of view. This decade also presented the changeover to electric motor drive centrifuges. Since then, the general mechanical design features in terms of specific power to drive the rotor have remained virtually the same although bearings, gear drive, belt drive etc. have gradually become better and better over the years. In parallel, separation performance of the disc stack has also improved which has brought down the specific power consumption \( \sigma \), i.e., the power consumption per flow rate to obtain a certain cleanliness of the output flow.

In summary, power consumption of high-speed centrifuges has been a fairly neglected area but is now becoming increasingly more important.

This study addresses the pertinent energy consumption contributors and aims at reducing the overall separator energy consumption by 50% at a certain flow rate, compared to traditional centrifugal high-speed separators. Currently, centrifugal separators operate approximately at a ratio \( \sigma \) of 1 kWh m\(^{-3}\) of power to feed flow rate [8] or even 1.4 [9], and looking at our
own Alfa Laval centrifuge range [10], overall agreement with this for open machines is noted although there is a lower range down to say 0.6–0.7 kWh m\(^{-3}\) for some cases with lower separation demand. The PRODIAS objective is to achieve operation at 0.5 kWh m\(^{-3}\) or less, without reducing separation efficiency.

### 2 Means for Reducing Power Consumption

The overall power consumption can be described by:

\[
P_{\text{tot}} = P_{\text{ang}} + P_{\text{win}} + P_{\text{mech}} + P_{\text{el}} + P_{\text{pump}}
\]  

(1)

where the respective contributors represent losses of angular momentum, windage from the rotor, mechanical drive friction, electrical losses in the separator motor, and internal pressure drop overcome by the pump and its motor.

The mechanical and hydraulic part of this power reduction project involves:

(a) Minimizing the losses of angular momentum.

(b) Reducing internal flow pressure (power) losses.

(c) Minimizing the mechanical drive losses and windage losses.

#### 2.1 Minimizing Losses of Angular Momentum

The first part (a) is achieved by having both inlet and outlet close to the center of the rotational axis, and so using as close as possible a center-to-center approach for the liquid passing through the rotor from inlet to outlet. This way the flow which after the inlet is brought into rotation in the separation zone (the disc stack) can return the rotational energy to the system and so minimize the losses that rotational flow would cause if the outlet was positioned off set of the axis and rotational energy would not be used in any positive way. The angular momentum of a liquid flowing out of a rotational system at radius \(r\) is:

\[
\dot{L} = \rho Q \omega r^2
\]  

(2)

From this one can derive the associated power loss of the rotating liquid leaving the system:

\[
P_a = M \omega = \dot{L} \omega = \rho Q \omega^2 r_{\text{out}}^2
\]  

(3)

where \(\rho\) is the liquid density, \(\omega\) is the angular velocity assumed fully spun up equaling the rotor speed, \(Q\) denotes the flow rate, and \(r_{\text{out}}\) is the outlet radius.

It is thereby evident that keeping the outlet diameter small is vital in order to minimize power losses from rotational liquid leaving the system.

The traditional centrifuge outlet design involves having the outlet coaxially outside the centrally positioned inlet, and using a paring disc to extract the rotational flow at a finite radius. Such a system, therefore, generates substantial power losses of angular momentum, despite the paring disc’s ability to convert some rotational energy into hydrostatic pressure, but only at an efficiency of some 25 %. Thereby at least 75 % of the rotational energy is lost to dissipation, and this even assumes that the hydrostatic pressure generated is 100 % useful, and not choked in a valve further downstream.

The newly developed system is based on having the inlet at the base and the outlet at the top of the centrifuge, allowing both passages to be close to the rotational center, which minimize losses of angular momentum to a minimum according to Eq. (3). Further, the use of a hermetic inlet and outlet is a necessity for allowing flow to be guided through the rotational axis with minimum losses. For developing purposes, a full-scale clarifier separator was designed and equipped with such inlets/outlets, as depicted in Figs. 2 and 3.
Experimental conditions: Alfa Laval 18-size centrifuge; clarifier setup, i.e., only one liquid outlet; process media: water $20^\circ\text{C}$.

The largest reduction of the overall power consumption originates from reducing losses of angular momentum. This was made possible by using a hermetically sealed central outlet and inlet. Consequently, the inlet is at the base of the rotor and the outlet at the top of the rotor.

The outlet was designed with a small enough radius to minimize losses of angular momentum, and yet not too small so as to allow large flow rates and still low pressure drop. Thereby an outlet diameter of 80 mm could be used, in contrast to a standard paring-disc outlet which for a flow rate of 160 m$^3$h$^{-1}$ may have a diameter of 150 mm, thus generating angular momentum losses of 74 kW (Eq. (3)) with an open machine compared to 21 kW for the current study. This already indicates a dramatic reduction of angular momentum though the pressure drop losses for a fair comparison have been not yet added. In a later section one will see that the reduction in angular momentum losses is about 40 kW at 160 m$^3$h$^{-1}$, or 0.25 kW m$^{-3}$, going from the open machine to the current hermetically sealed with central outlet. As outlined before, the paring disc itself can convert some rotational energy to pressure, but using a minimum-sized paring disc at this flow rate gives no regain, i.e., the paring disc efficiency decreases to zero.

Moreover, the centrifuge rotor was designed using a suitably sized and enlarged inlet, providing smooth passages and a very large inlet pipe, in order to minimize flow pressure drop (b) and associated power losses of the centrifuge and pipe feed pump. Note that the use of hermetic inlets/outlets connects the centrifuge flow system with the outside hydraulic system as the pressure drop of the centrifuge is now supported by the external feed pump. Therefore, this power consumer is included in the analysis.

As for the third item, i.e., mechanical drive losses (c), an electric direct drive frame was used eliminating gear or belt drive losses. In addition, the frame was equipped with a vacuum system extracting air outside the rotor, thus sharply reducing windage (air friction) losses from the rotor (c).

The following graphs indicate the power reduction obtained against the initial target set in the PRODIAS project (< 0.5 kWh m$^{-3}$). It should be noted that current industry standard is in the range of 1 kWh m$^{-3}$, hence 0.5 kWh m$^{-3}$ is a challenging and ambitious target.

Fig. 4 shows the power consumption levels reached implementing the full scope of power reduction items; inlet and outlet at the rotational axis, low pressure drop (hermetic, large inlet pipe, etc.), effective electric direct drive, vacuum imposed outside the rotor reducing air friction.

Evidently, the power consumption is sharply reduced compared to industry standard, as seen on the specific energy curve ranging from 0.3 to 0.5 depending on the flow rate (right-hand side axis). Note that the specific energy ratio is defined such that the smaller the ratio, the more efficient the centrifuge and the less power needed to drive it. The result is satisfactory from a PRODIAS target point of view, as the initial target set was 0.5 or below, hence, the main goal of the centrifuge PRODIAS program is hereby achieved.

It is particularly satisfactory to fulfil the target this way, as this analysis includes also the external pumping power, i.e., the pump feeding the inlet with a suitable flow rate, which is normally not included in the industry standard measurements. Further, the speed of the centrifuge is high, 5200 rpm, which is equivalent to a rotor peripheral speed of 200 m s$^{-1}$. High speed
assures high separation efficiency enabling running a larger flow rate through the process.

The graph shows also the power consumption of the centrifuge (blue) compared to centrifuge + feed pump (gray), and it is seen that the pressure drop generated in the smaller flow rate range (<50 m³h⁻¹) is almost negligible whereas in the large range (>150 m³h⁻¹) it is the dominating power consumption item. The separator power consumption, which is the sum of all drive train contributors (motor, bearings, windage) is only increasing moderately with flow rate, from 13 kW at idling, i.e., zero flow rate, to 24 kW at 200 m³h⁻¹. Looking at the windage losses individually, which was made possible switching on and off the vacuum pump, one can estimate the contribution from this power consumption reduction item more clearly, as demonstrated in Fig. 5, and in Fig. 6 plotting also specific energy consumption. The test was carried out allowing either ambient pressure outside the rotor (1.103 kPa) or absolute pressure reduced sharply. This way it was found that the reduction in air friction losses outside the rotor was about 13 kW, independently of the process flow rate.

It should be noted that the vacuum applied is not absolutely zero pressure, for practical reasons, instead 68 % vacuum was applied, and so the absolute pressure outside the rotor in Fig. 5 was (1–0.68) × 101.3 kPa = 0.32 kPa. In some other runs the pressure was down to 74 % vacuum (0.26 bar).

### 2.2 Minimizing Mechanical Drive Losses

This part consists of the items reducing power by mechanical means. Two items were implemented that significantly reduce the power consumption in this category:

- (1) Direct drive motor.
- (2) Lowering the aerodynamic drag of the rotating bowl.

Using direct drive motors will result in a lower power consumption corresponding to the power needed to drive an alternative drive providing the rotor shaft with the high revolutionary speed required. There are two widely used drives, i.e., belt drive and worm gear drive, where the former would have typically minimum 2 % losses of the delivered power and the latter somewhat more 5–10 %. It may be argued that the direct motor spins faster and so it should have larger aerodynamic losses, but these are small and part of an anyhow present high-speed shaft. Further, the belt drives under consideration are less than ideal having a considerable curvature on the small pulley and generating aerodynamic drag as well as they spin quite fast and, therefore, it may be even on the conservative side to say that the loss on a belt drive compared to a direct drive is 3 %. But also the motor itself involves a purposely designed induction motor for direct drive, and so an effectiveness of 95 % was achieved for the motor and the variable frequency drive (vfd) combined, which is an improvement from say 92 % for standard external motors with vfd.

![Centrifuge Power Consumption](image1)

**Figure 5.** Centrifuge power consumption with (dashed lines) and without (solid lines) air friction outside the rotor. Gray: centrifuge power consumption; blue: centrifuge power consumption.

![Centrifuge Power Consumption](image2)

**Figure 6.** Centrifuge power consumption with (dashed lines) and without (solid lines) air friction outside the rotor. Blue: centrifuge power consumption; orange: specific energy consumption (kWh m⁻³).
Moving on to the item studied in more detail in the present experiments, lowering the absolute pressure outside the rotor, one may present this as follows. Aerodynamic features generate considerable drag outside the rotor, since the peripheral speed of the rotor is in the excess of 200 m s\(^{-1}\).

### 2.3 Minimizing Aerodynamic Drag

Before implementing the vacuum technology reducing the aerodynamic drag, an analysis was carried out to explore the potential of such efforts. The aerodynamic drag of rotating bodies of variously shaped bodies are known theoretically and can be estimated analytically.

An area element of the cone (\(dA\)), as in Fig. 7 a, will sense a drag force (\(dF\)) which is proportional to the product of the dynamic pressure and a friction coefficient:

\[
dF = c_f \frac{\rho v^2}{2} dA
\]

where

\[
dA = 2\pi r ds = 2\pi r \frac{dr}{\cos \theta}
\]

such that the frictional torque for a turbulent flow case can be expressed as:

\[
dM = c_f \frac{\rho \omega^2 \pi}{2 \cos \theta} \frac{dr}{r} = c_f \rho \omega^2 \pi r^3 \frac{dr}{\cos \theta}
\]

\[
M = c_f \frac{\rho \omega^2 \pi}{2 \cos \theta} (R_1^3 - R_2^3)
\]

(7)

for which, as customary, it is assumed an average \(c_f\) over the surface can be used. With this one can derive the friction torque for, e.g., a flat disc (\(\cos \theta = 1\)).

\[
M = c_f \frac{\rho \omega^2 \pi}{S} (R_1^3 - R_2^3)
\]

(8)

which readily gives the power loss due to aerodynamic friction as:

\[
P_{\text{disc}} = c_f \frac{\rho \omega^3 \pi}{5} (R_1^3 - R_2^3)
\]

(9)

and similarly from Eq. (7) for a cylinder (\(\cos \theta = H/(R_1-R_2) = 0, R_1-R_2 \rightarrow 0\), l’Hospital rule):

\[
P_{\text{cyl}} = c_f \rho \omega^3 \pi H (R_1^3 - R_2^3)
\]

(10)

The coefficient of friction follows approximately estimations found in the literature on general flat-plate turbulent boundary layers, often quoted as \(c_f = 0.074 Re_\text{c}^{1/5}\), and more specifically empirical relationships can be found for rotating bodies in, e.g., [11], featuring turbulent Taylor-Couette flow [12] where the friction coefficient is, e.g., \(c_f = 0.051 G^{1/10} Re_\text{c}^{-1/5}\) involving a gap ratio \(s/R\) and \(Re_\text{c}\) is the rotational Reynolds number \(\omega R^2/\nu\) (disc) and for a cylinder \(c_f = 0.027^{-1/5}\).

A centrifugal bowl (Fig. 7 b) is from an aerodynamic perspective essentially a vertical cylinder with a horizontal disc at either end, thus disregarding the cone shape of some parts, but adjusting the factor experimentally from a real centrifuge bowl. This was explored in more detail in an earlier analysis dividing the rotor exterior into smaller sub-elements, consisting of either cylinders, discs or cones, and when adding them together, a fairly accurate power drag can be estimated and compared with experiments [13]. This is because the elements with largest radius dominate the contribution, like elements 1,2 (cyl) and 3,8 (discs) in Fig. 7 b. For the sake of brevity, this result is used here and two discs (Eq. (9)) are added to one cylinder (Eq. (10)) which reads:

\[
P_{\text{disc}} = c_f \frac{\rho \omega^3 \pi}{5} (R_1^3 - R_2^3)
\]

Figure 7. (a) Rotating cone, (b) sketch of a real VNPX710 centrifuge bowl exterior (blue) inside casing.
\[ P_{\text{rotor}} = 2P_{\text{disc}} + P_{\text{cyl}} = \frac{2\pi \rho a^3 R^5 \left( c_f \text{ disc} + \frac{5H_{\text{out}}}{2R} \right)}{5} \]

Note that when rewriting for a combined friction coefficient for the whole rotor this way, the resulting friction coefficient deviates substantially from that of a single disc and cylinder, as it becomes already analytically 3–4 times larger than that of an individual element as seen from the expression within brackets, as \( H/R \) is mostly within 0.9–1.0 for various sizes and applications of centrifuge rotors.

Although the above modeling involves some approximations, agreement with experiments was found to be reasonable, however 10% low. The slight deviation may be attributed to effects of the rotor periphery not accounted for, such as discharge ports in the rotor periphery or air friction losses due to the complex geometry of stationary frame parts, secondary vortex motion [14], or additional air pumping tendencies.

The coefficient found when adding all bowl exterior sub-elements [13] using the literature value of the friction coefficient was 0.0075, but following real centrifuge bowl experiments, it is adjusted to be:

\[ c_f \text{ rotor} = 0.0082 \]  

(12)

The windage power losses may hence be estimated for the ambient air pressure in this case to be:

\[ P_{\text{rotor}} = \frac{2\pi \times 1.189 \times \left( \frac{2\pi \times 5200}{60} \right)^3 \times 0.391^5 \times 0.0082}{5} = 18 \text{ kW} \]  

(13)

Now it was explored how reducing the absolute pressure can lower the aerodynamic losses of the rotor. From Eq. (11) it is implied that power will drop linearly with reduced pressure, as the ratio of pressure to density is constant for an ideal gas like air at constant temperature \((p/\rho = RT)\), and similarly the friction coefficient will not vary with absolute pressure since its only fluid property dependence is kinematic viscosity, which is only dependent on temperature.

In Fig. 8, aerodynamic power consumption is plotted vs. absolute air pressure.

Looking first at the theoretically derived power consumption, the linear relationship is seen with absolute pressure, reaching 18 kW at ambient pressure. Similarly, as indicated by the orange dot, when using experimental data from switching on/off the vacuum and adding the remaining drag contribution \((P(68\%))\), the prediction is an overall aerodynamic drag of 20.3 kW, which is somewhat close to the theory of 18.0 kW.

Although there is another 5 kW to be gained from lowering the pressure yet further from 0.32 bar to 0, this was not carried out, as there are practical limitations with having too low a pressure for the overall process operation, viz. handling of the intermittent discharge and the associated operating water.

2.4 Lowering Internal Flow Pressure Drop

Also presented here is the pressure drop for the process media through the centrifuge, measured using pressure sensors in the piping system before and after the centrifuge. Reducing internal flow pressure is important to the overall power consumption, as it sets the power consumption of the external feed pump power, which will be included in the overall power consumption for the centrifuge cleaning process in order to make a fair comparison with current industry standard. This is necessary as the presented low-energy method is based on using hermetic inlets and outlets for minimizing rotational flow losses, as outlined in previous chapters, and thereby the flow pressure drop and excess pressure at the outlet is generated primarily by the external feed pump upstream of the centrifuge.

An internal outlet pump is still optional, but from a system point of view it is advantageous to use only one pump, as in the current case, i.e., the upstream external feed pump. Note that the power analysis this way is conservative and may underestimate the savings made, as there is a feed pump also for the standard open inlet machines although it needs to operate only to overcome the pressure drop of the upstream piping system. The power consumption from flow pressure drop which the feed pump efficiency \( \eta \) needs to overcome can be expressed as:

Aerodynamic Power Consumption

5200 rpm variable vacuum

Figure 8. Centrifuge aerodynamic power loss from the exterior of the rotor. Green: actual aerodynamic power, experimental (using 68% vacuum). Orange: experimentally derived power consumption at 1 bar; gray: theory Eq. (11).
The turbulent flow regions, i.e., all regions apart from the flow between separator discs, are turbulent and dominate the overall pressure drop. The laminar part in the disc stack would account for some 8% of the overall pressure drop, as seen from in-house calculations verified against experiments. Further, the feed flow (inlet) and even more so the clean phase passages involve low viscosity for the biopharma application in question (aqueous systems) being almost identical to that of water, as it is characterized by the low concentration of solids. Only a potential solids outlet channel would experience higher viscosity and increased pressure drop. The pressure drop for various turbulent\(^1\) flow passages is according to classical expressions with recent corrections [16]:

\[
P_{\text{press.drop}} = \frac{Q \Delta p}{\eta} \quad \text{(14)}
\]

\[
\Delta p = \left( \zeta + \frac{\lambda L}{d} \right) \frac{\rho U^2}{2} = \left( \zeta + \frac{\lambda L}{d} \right) \frac{\rho Q^2}{2A^2} \left\{ \frac{1}{\sqrt{\lambda}} = 1.930 \log \left( Re_d \sqrt{\lambda} - 0.537 \right) \right\} \quad \text{(15)}
\]

where \( \zeta \) is a single loss pressure coefficient for a certain geometry, e.g., a bend. The second term with friction factor \( \lambda \) represents the pressure loss from a length of a pipe and \( U \) is the mean velocity in the pipe. This is an expression based on non-rotating cases and is hence valid for only limited rotation rates, as is the case for the rotating inlet pipe, unlike other parts of the rotor where adjustment for high rotation rates are required. Further, \( A \) is the cross-sectional area for the passing flow. By using hermetic inlet and outlet and carefully design the inlet and outlet passages, the respective single loss coefficients are reduced, since otherwise tight bends and throats or free surface effects generate flow patterns with substantial single losses. Further, increasing the flow cross-sectional area is highly beneficial, especially in the higher flow rate regime. For example, the inlet pipe cross-sectional area was increased by a factor of \( \frac{d_2}{d_1} = 57/44.5 = 1.28 \), thus reducing the inlet pipe pressure drop in Eq. (15) by a factor of \( 1.28^2 = 1.64 \) (64%).

In Fig. 9, it is notable that the centrifuge pressure drop is very low compared to industry standard, as a normal value at 100 m\(^3\)h\(^{-1}\) is otherwise about 6–7 bar. This is primarily due to the smooth central inlet and outlet, suitable internal rotor flow passages, and the use of a large-diameter inlet pipe. The associated change in power consumption is seen on the right-hand side axis, and at 100 m\(^3\)h\(^{-1}\) a difference of about 15 kW in pressure drop power is noted, which equals a reduction of 0.15 kWh m\(^{-3}\) and, if extrapolating, 0.25 kWh m\(^{-3}\) at 160 m\(^3\)h\(^{-1}\). Note that for the hermetic machine this is an explicit term in the power consumption (Eq. (1), pump power), but for an open machine it is a part of the flow losses calculated by Eq. (3).

The test was run up to a very large flow rate, i.e., 200 m\(^3\)h\(^{-1}\) to show the great improvements obtained positioning the inlet/outlet at the center and having a low pressure drop.

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**Figure 9.** Centrifuge pressure drop using the pressure drop reduction items. Blue: pressure drop over the centrifuge (upstream-downstream); orange: industry standard (B2 PX218 brew), markers: pressure, solid line: power; gray: inlet feed pump power for current hermetic centrifuge.
through the machine, which are less notable and important at a small flow rate. For practical reasons it was not possible to run the test on real process media involving a large batch of a solids-containing liquid system, e.g., fermented cell biomass like yeast or bacteria, and so instead the process was run with clean water. Thereby the separator was set to run in clarification mode, i.e., with only one liquid outlet. This is a real and practical process, corresponding to running the separator in clarification mode and handling the solids phase by intermittent discharges. Note, however, that this is only practical up to a certain solids concentration of the feed, say 0.5% concentration by weight. But, e.g., an undiluted fermentation solution may have a solids content of about 2% by weight, thus one may expect an addition to the power consumption from the solids outlet contribution if the solids feed needs to be continuously discharged.

An estimation of the additional power consumption at continuous discharge can be obtained considering the additional pressure drop experienced, which the feed pump would have to build up and which would have to be choked on the light phase by an external valve. A rough and conservative estimate of the additional pressure drop is 5 bar extra in addition to the pressure drop already presented through the machine (Fig. 9), and thus at 25 m³h⁻¹ there is an extra \( P = \Delta p Q = 5 \times 10^3 \times 25/3600 = 3.5 \text{ kW} \) and at 100 m³h⁻¹ an extra 14 kW. The associated specific power consumption (kWh m⁻³) would thus increase from 0.52 to 0.65 at 25 m³h⁻¹ and from 0.25 to 0.39 at 100 m³h⁻¹.

Returning to Eq. (1) illustrating the contributors to the overall power consumption, an example from the current study may be presented at a flow rate of 160 m³h⁻¹ and 5200 rpm for this large-size machine. The centrifuge power without the inlet feed pump is (Eq. (1) measured value):

\[
P_{\text{cen}} = P_{\text{ang}} + P_{\text{win}} + P_{\text{mech}} + P_{\text{el}} = 20.6 \text{ kW}
\]

such that the overall power consumption is:

\[
P_{\text{tot}} = P_{\text{cen}} + P_{\text{pump}} = 20.6 + \frac{160 \times 540 \times 10^3}{0.88 \times 3600} = 20.6 + 27.4 = 48 \text{ kW}
\]

where the pump efficiency \( \eta = 0.88 \) involves fluid and electric motor losses.

### 3 Power Consumption Contributors – Example at 160 m³h⁻¹

Continuing the above analysis, the respective contributions allow for calculating the actual angular momentum loss at 160 m³h⁻¹:

\[
P_{\text{ang}} = P_{\text{cen}} - P_{\text{win}} - P_{\text{mech}} - P_{\text{el}}
\]

\[
= 20.6 - 5 - 7.4 - 1.1 = 7.1 \text{ kW}
\]

which is strikingly low, compared to the flow losses for the paring disc alternative (74 kW). As outlined before, the paring disc itself can regain some rotational energy, but using a minimum sized paring disc at this flow rate gives no regain, i.e., the paring disc efficiency decreases to zero. Even when adding the feed pump power (27 kW) for the hermetic case for a better comparison of flow losses, a reduction of 74–7–27 = 40 kW of flow losses against the open machine is noted, or in specific energy a drop of 0.25 kWh m⁻³.

A further reason to this sharp reduction is that the average outlet radius is actually substantially smaller than the pipe outlet radius (40 mm). This is not surprising, as flow is fed back somewhat evenly distributed from zero radius to the outer

### 3.2 Mechanical Losses

The bearings (5 pcs), rotary sealings (3 pcs) and operating water hydro seal for the discharge system would account for mechanical losses, which can be estimated from the measurements using the idling power, i.e., Fig. 4 at \( P_{\text{cen}} \) at \( Q = 0 \), and subtracting the remaining aerodynamic losses from above and using an electrical motor efficiency of 0.95. This way the mechanical losses are:

\[
P_{\text{mech}} = (12.8 \text{ kW} - 5 \text{ kW}) \times 0.95 = 7.4 \text{ kW}
\]

This is a fairly high contribution of mechanical losses, which is due to the experimental machine being equipped with large-size bearings due the large inlet required for lowering the inlet pressure drop, and full scope of seals (3 pcs) for allowing three-phase application, and a major part consisting also of the operating water paring disc which is used for the solids phase intermittent discharge function.

### 3.3 Electrical Losses

The electrical losses of the centrifuge motor and vfd is:

\[
P_{\text{el}} = P_{\text{cen}}(1 - \eta_{\text{cen}})/\eta_{\text{cen}} = 20.6 \times \frac{1 - 0.95}{0.95} = 1.1 \text{ kW}
\]

### 3.4 Angular Momentum Losses

According to the previous section on this topic, the windage loss from the rotor is in the range of \( P_{\text{w}} = 5 \text{ kW} \) when applying an absolute pressure of 0.32 bar and 18 kW at atmospheric pressure. Thus, there is a reduction of 18–5 = 13 kW, i.e., 0.08 kWh m⁻³, as seen from both theory as well as experiments switching on/off the vacuum pump.

The bearings (5 pcs), rotary sealings (3 pcs) and operating water hydro seal for the discharge system would account for mechanical losses, which can be estimated from the measurements using the idling power, i.e., Fig. 4 at \( P_{\text{cen}} \) at \( Q = 0 \), and subtracting the remaining aerodynamic losses from above and using an electrical motor efficiency of 0.95. This way the mechanical losses are:

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The electrical losses of the centrifuge motor and vfd is:

\[
P_{\text{el}} = P_{\text{cen}}(1 - \eta_{\text{cen}})/\eta_{\text{cen}} = 20.6 \times \frac{1 - 0.95}{0.95} = 1.1 \text{ kW}
\]

Continuing the above analysis, the respective contributions allow for calculating the actual angular momentum loss at 160 m³h⁻¹:

\[
P_{\text{ang}} = P_{\text{cen}} - P_{\text{win}} - P_{\text{mech}} - P_{\text{el}}
\]

\[
= 20.6 - 5 - 7.4 - 1.1 = 7.1 \text{ kW}
\]
radius of the outlet pipe. The average outlet radius can be calculated from the power loss above by (Eq. (3));

\[ r_{out} = \sqrt{\frac{P_{ang}}{\rho Q^2 w_f}} = 23 \text{ mm} \] (21)

showing yet another advantage of the hermetic outlet by distributing the flow effectively inwards and not only limiting the flow to a layer at the outer wall (40 mm), and so limiting losses of both angular momentum and pipe pressure drop. Also, due to rotational effects, the velocity distribution will be more biased to the pipe center than for a non-rotating case [17].

4 Conclusions

The work attempts to substantially reduce the power consumption of high-speed separators (centrifuges), and so simplifying for machines to operate at larger flow rate. For some applications, the operating cost may be a limiting factor, and in this respect the achievements provide great potential for such applications to be run with high-speed centrifuges. The reduction seen was obtained analyzing the power consumption contributors in detail and choosing appropriate design so as to minimize power consumption. Primarily the following contributors were subject for optimization:

- Reducing rotational liquid losses at the outlet.
- Reducing the air friction around the rotor by lowering the absolute pressure (vacuum).
- Reducing the pressure drop of the internal flow passages, and so reducing the feed pump power consumption.

The project has succeeded and surpassed the target of minimizing the power consumption of a large-scale centrifuge such that its power consumption to flow rate ratio is less than 0.5 kWh m\(^{-3}\). Even better values down to 0.25 were achieved under certain process conditions.

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Symbols used

- \(A\) [m\(^2\)] area
- \(c_f\) [-] friction coefficient
- \(c_f\) rotor [-] average friction coefficient of the entire rotor body
- \(H\) [m] cylinder (rotor) height
- \(I\) [kg m\(^2\) s\(^{-1}\)] angular momentum
- \(M\) [N m] torque
- \(Q\) [kWh m\(^{-3}\)] flow rate
- \(p\) [Pa] static pressure
- \(P\) [W] power
- \(P_{ang}\) [W] angular momentum power loss
- \(P_{disc}\) [W] aerodynamic power loss to drive a horizontal disc surface
- \(P_{cyl}\) [W] aerodynamic power loss to drive the cylindrical part of the rotor body
- \(P_{el}\) [W] electrical power loss from the centrifuge motor and vfd
- \(P_{mech}\) [W] mechanical power loss from the centrifuge drive
- \(P_{pump}\) [W] power to drive the external feed pump
- \(P_{rotor}\) [W] aerodynamic power loss to drive the whole rotor body
- \(P_{win}\) [W] aerodynamic power to drive the whole rotor body
- \(r\) [m] radius
- \(r_{in}\) [m] inner circle radius
- \(r_{out}\) [m] outer periphery radius
- \(R_1, R_2\) [m] inner/outer radius
- \(v_\theta\) [m s\(^{-1}\)] azimuthal velocity, tangential velocity

Greek letters

- \(\eta\) [-] feed pump efficiency
- \(\rho\) [kg m\(^{-3}\)] density
- \(\sigma\) [kWh m\(^{-3}\)] specific power consumption
- \(\omega\) [rad s\(^{-1}\)] angular velocity
- \(\theta\) [rad] cone angle

Abbreviation

- vfd variable frequency drive

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