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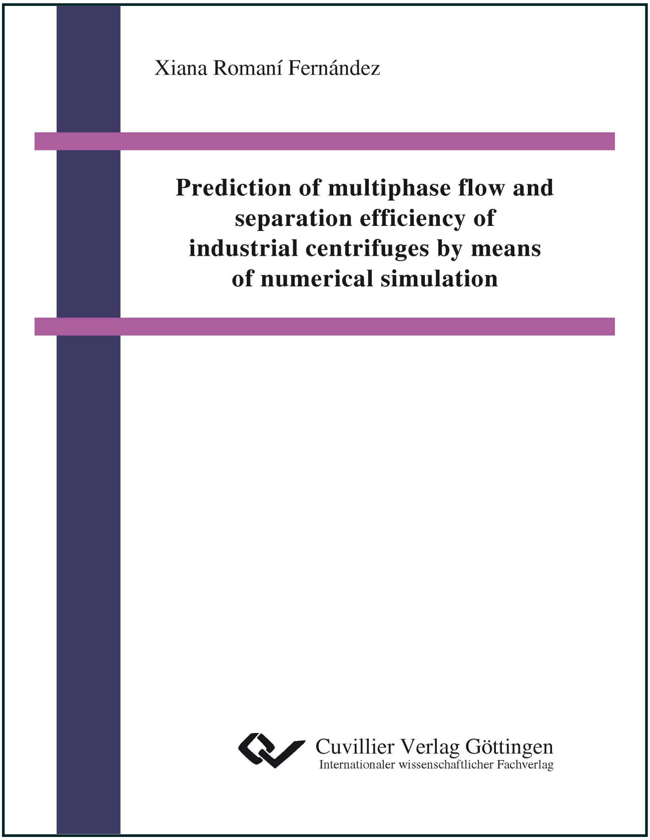
Prediction of multiphase flow and separation efficiency of industrial centrifuges by means of numerical simulation



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1 Introduction

Centrifugal separation of particles in a suspension is one of the most common problems appearing in industry such as chemical processing, mining and mineral processing, solid-fuel industry, food processing, waste water treatment, pharmaceutical processes and biotechnology. The separation of the cream in the whole milk to obtain skimmed milk, the isolation of blood components, the purification of beverages, the removal of catalysts in multiphase reactors or the cleaning of waste water and industrial fluids such as lubricants or coolants, are some relevant examples where centrifugal separation is needed.

Centrifuges are, together with sedimentation tanks and filters, one of the most commonly employed devices for mechanical solid-liquid separation. Sedimentation uses as its driving force a mass force based on the different densities of the phases. In sedimentation tanks this driving force is gravitational force while in centrifuges it is centrifugal force. Filters, instead, make use of differential pressure as a driving force. The choice of which technique should be applied for a specific separation task depends on the properties of the mixture, the purpose of the separation, whether the thickening of the solid phase or the clarification of the liquid, and the available time and space. The rapid development of the centrifugal technology nowadays allows high efficiencies separating very small particles, even in the nanometre range, from multiphase flows.

Centrifuges are classified into sedimenting and filtering centrifuges. Sedimenting centrifuges, the topic of this work, have a solid rotating bowl, while filtering centrifuges have a perforated bowl in which a filter medium is located. Sedimenting centrifuges consist of a rotating vessel filled with a suspension where the separation takes place. They can either be operated continuously or in batch mode. Screen bowl centrifuges, also called decanters, are continuous centrifuges used mainly for concentrated granular and pasty materials with particles sizes from 2 µm up to 30 mm, although they can operate with a wide range of concentrations. Disk centrifuges also operate continuously and are able to separate large quantities of low concentrated suspensions and very small particles, down to 0,4 µm (Luckert, 2004). Examples of semi-continuous centrifuges are tubular bowl and basket centrifuges. Tubular bowl centrifuges reach high centrifugal forces of up to 40.000 times the gravitational force (Spelter et al., 2010) and its stability derives from its relatively high length to diameter aspect ratio, in the range of 4 to 8 (Luckert, 2004). Solid bowl basket centrifuges, the focus of this work, can have a conical or a tubular bowl, but with smaller length to diameter aspect ratio, usually lower than 0,75 (Leung, 1998). Radial or vertical compartments or blades are often included to improve the separation efficiency.

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Centrifuges have various solid discharge systems, from manually to fully automatically with peelers or scrapers. In tubular and basket centrifuges the discharge of clarified liquid occurs semi-continuously until the sediment hinders the settling and the quality of the effluent significantly decreases. At this point, the feed stops, the bowl slows down and the solids are discharged prior to a subsequent centrifugation cycle. In other types of centrifuges with rotating buckets or tubes, the performance is entirely discontinuous and there is no effluent during the centrifugation. These kinds of centrifuges are in common use in laboratories but not prevalent in industrial processes.

Fig. 1.1 represents the main centrifuge object of this research, a semi-continuous solid bowl centrifuge with an automatic sludge discharge system. The liquid containing impurities enters the centrifuge and reaches the feed accelerator. It then leaves the accelerator and strikes the rotating liquid pool where the sedimentation occurs. The particles settle on the bowl wall, while the clear liquid, the so-called centrate, leaves through the boreholes at the top of the bowl. After flowing through these boreholes, the clear liquid is collected in an annular chamber, where a skimming pipe drives it out of the machine. During the centrifugation, radial baffles mounted on the shaft rotate with the same angular velocity as the bowl in order to accelerate the liquid. Once the maximum solid load at the wall is reached, the bowl slows down. When the centrifuge stops, the baffles move separately to facilitate the sludge discharge thanks to the scrapers attached at the edge of the baffles.

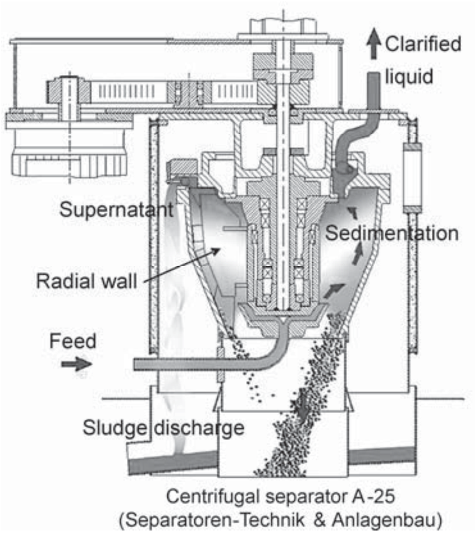

Fig. 1.1. Centrifuge A-25 from the company "Separatoren-Technik und Anlagenbau GmbH".

Despite centrifugation being a well-established operation, the knowledge of the complex phenomena occurring inside of centrifuges is limited. The prediction of the separation efficiency of a centrifuge is often restricted to the cut size, the smallest particle size that can be separated. Some analytical models allow the calculation of the smallest particle size that can settle on the bowl wall, assuming a certain internal flow

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in the centrifuge. In a rotating bowl, the main flow is in a tangential direction. However, due to the throughput, a secondary flow in the axial direction arises. The flow in the axial direction establishes the residence time of particles in the bowl and is thus the one that determines the separation condition for a certain rotational speed. The two most common models for the axial flow in centrifuges are the plug flow model and the boundary layer model, explained in chapter 2.1.2.

Nevertheless, the predicted cut size of a centrifuge frequently diverges from the real separation efficiency due to the fact that the assumptions of the analytical models are not strictly accurate in real industrial centrifugal conditions. In industrial basket centrifuges with complex shapes and internal assembly parts, it is difficult to determine the flow pattern. For example, some turbulence commonly appears near the inlet and it is not effectively possible to use the entire length of the centrifuge for settling. Moreover, most of these models apply Stoke's law to the settling of the particles, which is only correct for the laminar flow regime and single particle sedimentation. Furthermore, it is desirable to describe the separation performance of a centrifuge as a separation efficiency curve dependent on the particle diameter and not just as a cut size, which is not possible with the current analytical methods.

Due to the increasing computing resources and the variety of simulation models, Computational Fluid Dynamics (CFD) has become an important tool to study the flow within industrial machinery, and thus within centrifuges. The aim of this work is to develop a tool able to predict accurately the separation efficiency in an industrial centrifuge by means of numerical simulation of the multiphase flow.

The followed strategy implies, firstly, the simulation of the continuous phase flow in the centrifuge. Since most centrifuges have an air core open to the atmosphere a liquid-gas multiphase flow model is necessary. For this liquid-air flow simulation, a commercial CFD software ANSYS FLUENT® was used. After the continuous phase has converged, the particulate phase is simulated. For the simulation of particulate flows in CFD different approaches can be used (van Wachem and Almstedt, 2003). Neglecting particle-particle interactions and taking into account only the hydrodynamic forces, the trajectory of the particles can easily be tracked. It is thus possible to describe the separation efficiency of the simulated centrifuge depending on the particle properties and the operating conditions.

In order to simulate the sediment build-up and its behaviour, it is necessary to consider particle-particle interactions. For this purpose it is appropriate to apply the Discrete Element Method (DEM) (Cundall and Strack, 1979), originally developed for the simulation of bulk granular materials and then extended to the whole particle processing technology. This method accounts for particle-particle and particle-wall interactions by means of various contact models. Other forces such as electrostatic or Brownian forces can be also implemented when necessary. In centrifugal field, the hydrodynamic forces (drag, lift, torque) play a crucial role in the particles' movement

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and must be included. Hence, a coupling of the DEM simulation with the CFD simulation is necessary. For each position in the CFD computing field, the velocity, pressure, density and viscosity of the flow must be transferred to DEM software EDEM®, produced by DEM Solutions Ltd. The particles settle on the bowl wall forming a sediment which influences the flow. Thus, the information regarding the particle positions must be passed on to the CFD software in order to recalculate the flow considering the pressure drop caused by the sediment. This complex and coupled system is calculated numerically.

The advantage of this method is that the internal flow, the particle trajectories and the deposition of the particles can be represented accurately. It allows a detailed description of the flow and particle behaviour under various operating conditions, which is of high importance for the design, optimisation and laying-up of centrifuges. However, the simulation results should be examined carefully from a scientific point of view and must be validated with experimental work. The aim of this study is to provide answers to open questions such as the following. To what extent does the simulated flow represents the real flow within centrifuges? Can this methodology be used to accurately forecast the grade efficiency of industrial centrifuges? If so, are there limits for the calculated particle sizes and what is the basis of these limitations? To what extent is it possible to predict the build-up of a sediment in a centrifugal field using the coupling between CFD and DEM? Does the simulated sediment structure and behaviour correspond to the actual behaviour? The methodology used was tested with various centrifuge geometries and operating conditions. However, due to the use of specific models for the turbulence and the interaction between phases, this methodology might be only partially valid for completely different operating conditions and geometries. Nevertheless, this study provides a good starting point for further simulations of centrifuges.

Outline

This thesis begins with an overview of the theoretical background and the state of the art described in the current literature of both, flow and sedimentation in centrifugal field and multiphase flow simulation methods. The methodology used for the numerical calculation of the flow will then be presented together with the results of the flow patterns and its discussion. The various approaches pursued to validate the simulated flow, the Laser Doppler Anemometer measurements, residence time distribution measurements and colour marking tests are shown in chapter 5. The simulation of single particles leading to the particle trajectories will be explained in chapter 6. Furthermore, in chapter 7, there is a description of the DEM employed and its coupling with CFD in order to simulate the sediment build-up and its behaviour. Chapter 8 deals with the validation of the particles simulation by comparing simulated with experimental results performed in industrial centrifuges. The last chapter gives a summary of the obtained results and, moreover, some suggestions for future research.

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2 Centrifugation. Fundamentals and state of the art.

This chapter is dedicated to the fundamentals of centrifugal solid-liquid separation, main issue of this work. It includes an overview of the investigations about the different theories for the flow in centrifugal field and the analysis of particle sedimentation.

2.1 Flow in centrifugal field

The prediction of the separation efficiency of a centrifuge is often limited to the cut size, the smallest particle size that can settle on the bowl wall for certain operating conditions. Some analytical models allow the calculation of the cut size assuming a certain internal flow in the centrifuge. In a rotating bowl the main flow is tangentially. However, due to the mass conservation of inflow and outflow, also a secondary flow in the axial direction must occur. The axial flow is the one that determines the residence time of the particles inside the centrifuge and therefore the settling condition. The two most common flow models assumed in centrifuges are the plug flow model, where the whole liquid in the rotating pool has the same axial velocity directed to the outlet, and the boundary layer model, where just a thin layer of liquid at the interface between air core and rotating pool has axial velocity directed to the outlet. These two models will be explained in detail in chapter 2.1.2.

2.1.1 Tangential flow

A cylindrical coordinate system is useful to describe the velocity of the fluid at any given point in a centrifuge. This way, the velocity and the position can be divided into three components: radial, tangential and axial.

The main flow in the centrifuge occurs in a tangential direction. In a rotating bowl a stratified flow with increasing tangential velocity along the radius is formed. Assuming that the rotating liquid achieves the tangential velocity of a solid body rotation, the angular velocity ω remains constant along the radial position r and corresponds to the angular velocity of the bowl ω_{bowl}:

$$\omega = \omega_{bowl} \neq f(r)\,. \tag{2.1}$$

The tangential velocity can be then calculated as:

$$u_{tangential} = r\,\omega\,. \tag{2.2}$$

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However, the real tangential velocity found in centrifuges can differ from the solid body rotation, mostly because of inadequate inlet acceleration but also due to the friction between the layers in real fluids. Thus, Reuter (Reuter, 1967a) and Gösele (Gösele, 1968) proposed similar power law equations for the tangential velocity which deviates from the solid body rotation. This deviation is considered in the exponent n.

$$u_{tan\,gential} = u_{bowl}\left(\frac{r}{R}\right)^{n}, \tag{2.3}$$

where $u_{tangential}$ represents the tangential velocity at a radial position r, u_{bowl} the tangential velocity of the bowl and R the radius of the bowl. The exponent n can take different values depending on the tangential velocity profile achieved in a centrifuge under certain operating conditions, which mainly depends on the way the feed is accelerated. Fig. 2.1 shows some of the possible tangential flow patterns that can appear in a rotating bowl. If the fluid is introduced radially to the rotating pool without having been sufficiently accelerated, there is slip and the fluid rotates with a lower velocity as the bowl ($n > 1$). For an efficient feed accelerator that brings the impacting fluid to the tangential velocity of the pool, rigid body rotation is attained and n takes the value of 1. If the liquid is accelerated at the inlet with a higher tangential velocity, $n < -1$ could occur but this is difficult to realise and rarely happens in centrifuges. Usually $n \geq 1$ and a relative tangential velocity appears in the fluid near the inlet zone, which can be accelerated along the length of the centrifuge and can even achieve the solid body rotation for a sufficient length.

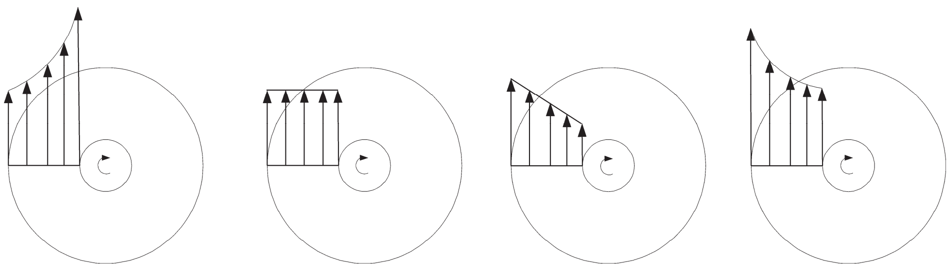

Fig. 2.1. Possible tangential velocity profiles in a rotating bowl.

Except from the inlet and outlet zones there is no radial mixing in the bowl. The stability of this stratified flow is explained by Reuter (Reuter, 1967a) and Gösele (Gösele, 1968) thanks to the centrifugal force. A fluid element at a certain radius r_1 has a lower tangential velocity compared to a fluid element situated at a larger radius r_2, and thus a lower kinetic energy. In order to transport a fluid element in the radial direction from r_1 to r_2, a certain work against the centrifugal force must be done to overcome the pressure gradient. Radial circulations just happen at the spin up and spin down of a filled rotating bowl, which will be explained in chapter 2.1.3, or due to distorting elements in the flow. Thus, a velocity slip between the rotating fluid and the bowl can lead to secondary radial flow.

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In order to avoid the slip between the rotating liquid pool and the bowl radial baffles or vanes, which rotate with the same angular velocity as the bowl, can be assembled. Sokolow (Sokolow, 1971) accounted the effect of radial built-in vanes by increasing the perimeter of the contact between moving parts of the bowl and the fluid to be accelerated. The effect of these internals on the sedimentation was studied by Schaflinger (Schaflinger, 1987; Schaflinger et al., 1986) who argued that the assembly of radial walls, which divide the centrifuge in chambers, hinders the negative effect of the Coriolis acceleration that caused the slip of the fluid. This way a convection flow of the continuous phase is originated within each chamber and the tangential velocity increases. Nevertheless, the effect of internals such as radial baffles, blades or deflectors that divide the centrifuge into radial chambers, on the secondary flow has not been thoroughly investigated yet.

The centrifugal force acts as a mass force directed to the bowl wall over the rotating pool originating a hydrostatic pressure gradient on the fluid. Applying the Navier-Stokes equations in cylindrical coordinates under the assumption that the fluid rotates as a rigid body and neglecting any radial and axial velocities, a differential equation for the pressure p can be obtained:

$$\rho \frac{u_{tan\,gential}^2}{r} = \frac{dp}{dr}\,. \tag{2.4}$$

The integration of Eq. (2.4) between the interface radius r_0 and the bowl radius R leads to the following equation where the hydrostatic pressure grows up with the square of the radial position:

$$p = \frac{\rho\omega^2}{2}\left(r^2 - r_0^{\,2}\right). \tag{2.5}$$

2.1.2 Axial flow. The Plug Flow theory and the Boundary Layer theory

The first publications dealing with the flow and sedimentation process in centrifugal field assume the simplest axial flow possible in a device. This flow is achieved if the whole volume of the centrifuge is flowed through, ie. the whole volume of the device is being continuously changed by new suspension (Fig. 2.2). Many authors (Horanyi and Nemeth, 1971; Sokolow, 1971; Stahl, 2004; Svarovski, 1977; Trawinski, 1958) supposed this kind of flow for the calculation of the equivalent clarification area in order to determine the cut size.

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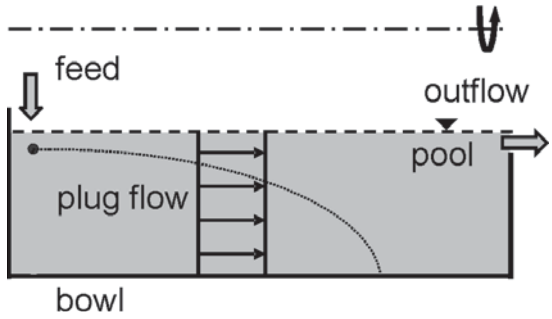

Fig. 2.2. Schema of the plug flow in a solid bowl centrifuge with the corresponding particle settling trajectory.

Following this theory, the residence time in the centrifuge can be calculated with the flow rate Q and the volume of the liquid pool V, which is a function of bowl radius R, pool radius r_0 and length of the centrifuge L:

$$t_{plugflow} = \frac{Q}{V} = \frac{Q}{\pi\left(R^2 - r_0^2\right)L}. \tag{2.6}$$

In this case, the whole fluid in the pool has a constant axial velocity inside the bowl and there are no axial velocity gradients. Thus, a particle has to travel the distance from the point where it is fed up to the bowl wall before it goes out of the centrifuge in order to settle.

$$v_{axial} = \frac{Q}{\pi\left(R^2 - r_0^2\right)}. \tag{2.7}$$

The determination of the cut size using Eq. (2.7) for the calculation of the axial flow rarely agrees with the results obtained in the praxis. For this reason Bass (Bass, 1959a, 1962) developed in the late 50's another theory for the axial flow in solid bowl centrifuges as the one assumed for the basic calculation of the cut size. He supposed that the liquid flows just in a circular ring cross section at the interface between air core and liquid pool. This ring was supposed to go from the surface of the pool r_0 to the radius of the weir r_{weir}. He also assumed that the angular velocity of each volume element in the centrifuge was constant and achieved the velocity of the rotating bowl. A theoretical analysis by means of an energy balance for ideal fluids leaded to the following equation for the averaged velocity in this layer:

$$\bar{v}_{axial} = \frac{Q}{\pi\left(r_{weir}^2 - r_0^2\right)}. \tag{2.8}$$

Bass defined the parameter a as the hydrodynamic characteristic of the centrifuge, which was found to be dependent on the operating parameters flow rate Q and rotational speed ω:

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$$a = \sqrt{r_{weir}^2 - r_0^2} = \sqrt[3]{\frac{3}{2\pi}\frac{Q}{\omega}} \,. \tag{2.9}$$

He proved the existence of this layer with a colour tracing method. Its thickness was measured with a selecting pin to determine the position of the surface r_0. The experimental values agreed with the theoretical analysis in which the average layer velocity was a function of flow rate and rotational speed of the bowl.

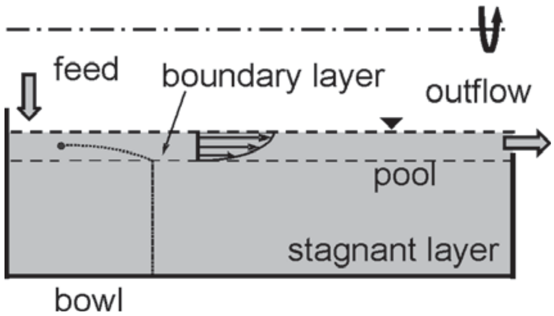

Fig. 2.3. Schema of the axial boundary layer model in a solid bowl centrifuge with the corresponding particle settling trajectory.

Some years later, Reuter (Reuter, 1967a, b) made some investigations of the flow in a solid bowl centrifuge with an overflow weir. He also predicted a layer with axial flow at the interface in the inner side of the liquid pool. He was the first one who used the colour tracing method not just to identify the layer but also to measure its axial velocity using video recording. Instead of a constant velocity in the axial layer he observed a layer with a gradient in the radial direction, as shown schematically in Fig. 2.3. The highest axial values were measured direct at the interface and the values diminish with the radio until zero. Positive values of axial velocity were found for radius bigger than the weir radius, which means that the layer thickness is bigger than the weir radius. Some negative values for the axial velocity appeared at the bowl wall, indicating that there is a recirculation. This recirculation, also observed by Bass, was explained as a consequence of the shear stress between the axial boundary layer and the stagnant rotating pool. His analytical treatment leaded to the following equations for the maximal axial velocity in the boundary layer, reached at the interface, and for the axial layer thickness δ:

$$v_{axial\,max} = \frac{\omega^2\ \rho}{24\ \mu\ L}\delta^4 \,, \tag{2.10}$$

$$\delta = 1{,}478\sqrt[5]{\frac{L}{r_{weir}}\frac{\mu\, Q}{\rho\, \omega^2}} \,. \tag{2.11}$$

Thus, the layer thickness depends on the geometric parameters of the centrifuge (first term under the square root), on the physical properties of the fluid, density ρ and viscosity μ, and on the centrifuge operating parameters. The comparison between the

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theoretically calculated layer thickness and the ones observed in the experiments was good with an averaged deviation of 7%.

Gösele (Gösele, 1968) described a theoretical treatment of the flow in tubular centrifuges also considering an axial boundary layer. Starting from a Bernouilli balance, he developed a differential equation to calculate the axial velocity of each radial position of the outgoing flow depending on the radial position of the path line. This way, it was possible to predict the maximum radius of the layer r_{layer}, which is the radius corresponding to an axial velocity of zero. The thickness of the layer (r_0-r_{layer}) depends on the geometry and on the value of n, which indeed is also dependent on the values of the flow rate and rotational speed for a centrifuge with a certain feed accelerator and geometry:

$$n\left(\frac{r_{layer}^{\ 2}}{r_{weir}^{2}}-1\right)=1-\left(\frac{r_0}{r_{layer}}\right)^{2n} . \tag{2.12}$$

Eq. (2.12) predicts higher values for the thickness as the one obtained experimentally by Bass and Reuters. The reason is that the analytical treatment was made considering an inviscid fluid, but for real fluids there is a certain friction between the layers which leads to energy loses and hence to thinner layers.

Faust, together with Gösele, (Faust and Gosele, 1985) even proposed a model for the flow in decanters. They performed colour tracing measurements in a transparent decanter centrifuge with an advancing screw conveyor. The flow profile found for the clarification zone is represented schematically in Fig. 2.4. The inlet is under-accelerated when entering the pool and has a relative movement in the negative direction with respect to the rotation until it reaches the conical end of the screw conveyor. Then an axial boundary layer, which comprises about half of the liquid pond, is formed. The liquid pond underneath this layer moves very slowly forming Taylor vortices parallel to the rotation axis.

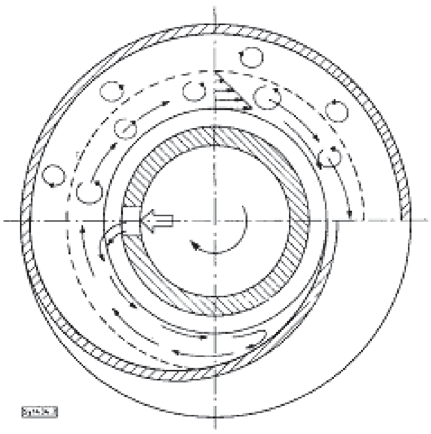

Fig. 2.4. Schema of the flow in the clarification zone of a decanter with an advancing screw conveyor (Faust and Gosele, 1985).

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Sokolow (Sokolow, 1971) argued that there were experimental investigations showing that the axial boundary layer in a cylindrical bowl centrifuge was possible but also enough experimental observations where the flow occurred along the whole cross section of the pool in favour of the plug flow theory. He found both of the models for the axial flow in different centrifuges and for different operating conditions. He also established a dependence of the layer thickness on the value of the exponent n. In cases where there was a big slip between the pool and the bowl, the layer was found to be thin. In a case of a centrifuge with radial walls, which avoid the slip, the determined layer was thicker.

Lohe and his co-workers (Lohe et al., 1971) solved the Stokes equations (neglecting the inertia) for a simplified problem near the weir and using certain boundary conditions. They came to the conclusion that alone the internal friction of the fluid could explain the boundary layer behaviour of the axial flow in solid bowl centrifuges. Using a dimensional analysis they obtained the following relationship between variables:

$$\delta \approx \left(\frac{\mu\, Q}{\rho\, r_0\, \omega^2} \right)^{\frac{1}{3}} . \tag{2.13}$$

After having studied the position of the air-liquid interface in overflow centrifuges, Meyer (Meyer, 1978) also investigated the thickness of the boundary layer formed in these devices. He developed an equation for the layer thickness δ from the Navier-Stokes equation and the equation of the velocity distribution in the layer analogous to the velocity distribution in a trickle-film. Eq. (2.14) shows the same dependence on the operating parameters proposed by Lohe, Eq. (2.13).

$$\delta = \left(\frac{3\, Q\, \mu \left(2 - \frac{r_0}{r_{weir}} \right)}{2\pi\, r_{weir}\, \rho\, \omega^2\, r_0\, \frac{\Delta\varepsilon}{L}} \right)^{\frac{1}{3}} , \tag{2.14}$$

where $\Delta\varepsilon$ represents the increment of the water level from the bottom to the top of the centrifuge, ie. along the length L. His measurements with colour tracer showed that the thickness of this layer was not constant in the length of the centrifuge, although it was assumed to be in Eq. (2.14). Furthermore, the thickness was dependent on the flow rate, increasing its value for higher volume rates, and on the slip of the layer regarding to the solid body rotation. In contrast to Sokolow, he found that in the cases where there is just a slightly slip regarding to the solid body rotation, ie. for liquids with a higher viscosity and centrifuges with an efficient feed accelerator, the boundary layer thickness decreases.

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The colour tracing method used until the 80's to identify and measure the axial boundary layer at the interface had some deficiencies regarding an adequate injection of the coloured ink and its diffusion. For this reason, Glinka (Glinka, 1983) investigated the axial flow in overflow centrifuges by means of an electrolytic tracing method. Using two electrodes with a variable radial position one at the bottom and one at the top of the centrifuge, he was able to measure the axial velocity profile along the whole liquid pool. He found a strong turbulent zone near the inlet which could occupy up to one third of the centrifuge length. One of the reasons for these turbulences is the liquid coming from the inlet accelerator, which is under-accelerated regarding the rotating pool. Another reason is the deviation in the tangential and axial direction of the water jet from the feed accelerator which impinges the pool with a certain radial velocity. This entry length was already noticed by Bass (Bass, 1962) who proposed a minimum length for centrifuges to acquire a stable flow:

$$L_{min} = 14\,R\,. \tag{2.15}$$

Glinka agrees with this equation that, in his opinion, establishes a maximal requirement because of the stabilising effect of the friction in real fluids and of the high rotational speeds. His results showed a layering in the upper zone of the centrifuge after passing the inlet zone. The axial velocities were much higher at the interface, but there were also positive values of the axial velocity in the remaining liquid pool. He also found a recirculation layer, with negative values at the bowl wall. He argued it with the formation of a boundary layer at the top and at the bottom of the centrifuge during the spin-up process. Also the slip of the layer at the interface, which must be accelerated along the length of the centrifuge, could be the cause of this recirculation. He concluded that there are two limit states for the axial flow in the centrifuge, the plug flow and the layer flow model, which can appear superposed.

Nowadays authors, such as Leung (Leung, 1998), describe both of the flow models that can be found in centrifuges and can be used to calculate its cut size. The tendency to one or to the other depends on the fluid characteristics, the operating conditions and the centrifuge geometry, especially on the geometry and efficiency of the feed accelerator. The presence of the axial boundary layer and its thickness is of mayor importance to determine the settling criterion of the particles in centrifugal field. The different settling paths made by the particles in the two flow models can be seen in Fig. 2.2 and Fig. 2.3. The thinner the axial boundary layer, the shorter is the distance that the particles must overcome to settle, but the lower is the residence time because of the high velocity of the layer. A thick axial layer has the advantage of lower velocities and thus higher residence time, but also the disadvantage that the particles must overcome a longer distance to reach the stagnant pool to settle. Furthermore, the room available for the sediment decreases in the last case and the possibility of the settled particles to be resuspended increases.

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