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Common strategy to prevent the Danube's pollution technological risks with oil and oil products - CLEANDANUBE

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STUDY 4

Technical project of a new purification solution of water infested with oil

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CONTENT

Study no. 4

TECHNICAL PROJECT OF A NEW PURIFICATION SOLUTION OF WATER INFESTED WITH OIL

In the frame of this stage it will define the main features, constructive and functional of centrifugal separation plant (system), so as to meet its purpose.

It also defined the relationships for sizing calculation of main parts and the auxiliary equipment that are comprised in this plant. In the theoretical work has emphasized both aspects of biphasic separation, liquid-solid (in plants named Dicanters) and three- phase separation of liquid-liquid-solid (in plants named Tricanters).

The theoretical relationships presented provide input data for the modeling phenomena in fluid dynamics simulation (CFD). At this stage, is presented and an example of calculation by simulating the behavior of fluids in the system.

After analyzing the results it is found that the centrifugal separator designed configuration is very effective.

In the present stage are also attached the main sub-assemblies drawings that make up the plant.

4.1 Introduction

Following previous studies on the development of technological and constructive parameters and effective technological schemes to prevent and remove the effects of pollution the Danube, were determined technical conditions for the design and construction of treatment plants based mainly on centrifugal separation technology.

In Fig. 4.1 is represented overall design of the separation plant. At this stage it will define the main features of the constructive and functional installation of centrifugal separation, followed the next stage to focus on ancillary elements that goes into the system .

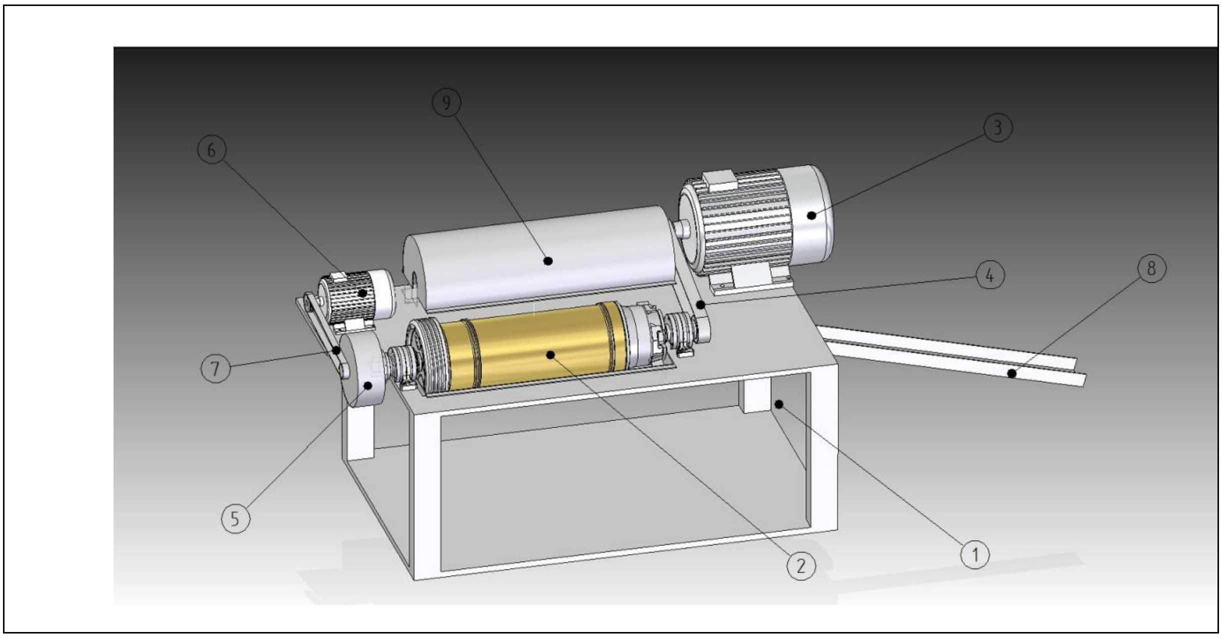

Fig. 4.1General configuration of the installation of centrifugal separation.

1.Frame
2. Centrifugal separator.
3.Main drive
4.Belt drive centifuge bowl
5.Gearbox
6. Gearbox drive motor
7 Belt drive gearbox
8.Guide for solid discharge
9.Casing

The main features of centrifugal separation facilities are:

Axis position –horizontally. This configuration has the advantage that both bearings separator and drive motors are not required to axial forces.

-**Effectiveness factor Z or G force** (maximum overload in the centrifugal separator)value will be around 3000. For this reason, it is necessary for a scroll diameter of 350 mm an estimated maximum speed of the centrifugal bowl about 4000 rpm.

- Construction type of the separator: in counter-current. This configuration has the advantage of increased retaining time of the mixture into a separator, in order to assure an effective separation.

-Drive system. Drive systems of bowl and the scroll will be fitted with variable speed motors controlled with frequency converters to ensure optimal system programmed. Will be dimensioned the maximum necessary power for the aggregates. These powers should be provided in all the situations where the centrifugal separator is installed on the ship, on a ground vehicle or ground fixed.

-Conditions for bearings

Centrifugal separator plant will be equipped with ball or roller bearings. The lubrication for these components will be grease.

For differential gearbox lubrication system is oil by centrifugation. This closed lubrication system is effectively when power transmitted to drive the scroll is less than 50 KW.

-The quantity of wastewater processed.

The feed rate must be about 15÷25 m^3/h. This is function of the mixture caracteristics.

-Command and control system.

This should display the functional parameters of the plant (processing rate flow, rotational speed of moving parts, temperatures in the bearings) and stop the operation when critical values are exceeded. These values of critical functional parameters will be determinated in the next stage of the project.

-**Protective barrier.**

Water quality monitoring stations will be equipped with systems to prevent the spread of pollution upstream. Following studies, will be designed multilayer barrier to the spread of the upstream petroleum.

-**The filtration and suction system.**

The filtration system must be dimensioned in such a way to allow the crossing of designed flow in the quantity and quality needed for the suction pump to function. The suction pump must ensure the designed flow for the centrifugal installation with an adjustable output pressure of 4÷7 bar. When centrifuge plant is installed on a ground vehicle or fixed to the ground, it is necessary that the collection system to be mounted on a floating platform, remotely controlled.

-**Sizing of storage elements** (storage tanks) of petroleum and solid fractions. For the operation on the ship and on the ground it is necessary to be sized the tanks for the recovery of oil and the solid fraction to ensure a continuous and as long as possible removement of the effects of pollution. Following the results presented in the Study no.3 solid concentration fractions in Danube waters generally do not exceed 4000 mg/l. After one hour of operation with a feed flow of 20 m^3, will result a solid fraction about 24 kg for a 40% water concentration in mixture.

4.2 Basic Components

Type centrifugal separator designed in the context of this program can be used both as solid-liquid phase separator, and is part of dicanters (used for clarification of liquids) or three-phase separator consisting of two immiscible liquids and solid and are known as the tricanters.

In the following will be addressed in detail the theory and construction of centrifugal separators included in tricanters class and how can also be used in other applications.

Many of the basic components of the centrifugal separator have been introduced in describing the construction of the centrifugal separator. These need to be described in more detail. The four major component assemblies are the rotating

as the frame and casing together, and the drive and back-drive assemblies.

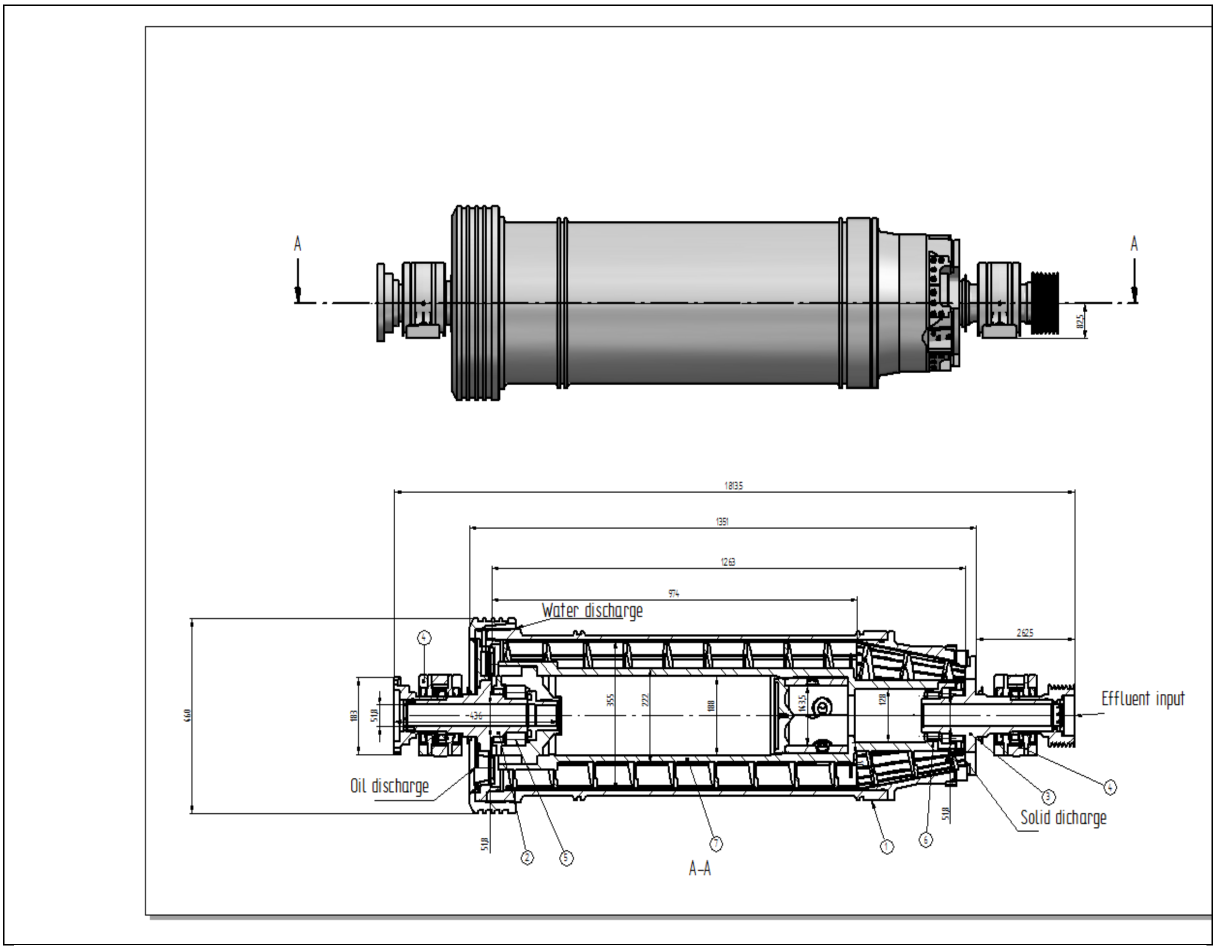

Fig 4.2 Main components of centrifugal separator

1.Bowl assembly

2.Front hub assembly

3.Rear hub assembly

4.Hub pillow block bearing

5.Front bowl bearing assembly

6.Rear bowl bearing assembly

7.Scroll assembly

The rotating assembly includes the bowl, conical drying zone, scroll and gearbox. It is the most important (and expensive) part of the centrifugal separator, where all the work is done, and which contains the most sophisticated technology, both process and mechanical. For such a heavy component, weighing up to several tons and producing a force field of several thousand **g**, a high level of precision engineering is required, followed by precise balancing.

Bearings and seals used in the rotating assembly and gearbox are an important part of the centrifugal separator. Bearings in general have to be lubricated to work properly. To do this, seals separate the lubricated bearings from the process environment, both to protect the bearings and to avoid contamination of the product or environment, by the lubricant. Seals are also needed to contain process liquids and vapors within the centrifuge casing. Seals are especially important where the process requires a positive pressure or vacuum, and where vapors are flammable or toxic.

4.2.1 Orientation

The rotational axis of the centrifugal separator can be horizontal or vertical. The vertical designs are most frequently used for special applications and are described in Study no.3. Thus the horizontal design will be taken from here on as the basic design.

4.2.2 Flow

The flow of clarified liquid and solid in the centrifugal separator can be either co-current or countercurrent. In the co-current design, both solids and liquid travel in the same direction, axially, in the separating zone, with the clarified liquid diverting to the opposite end to the solids discharge through off-take channels. With the countercurrent design, solids and liquid travel in opposite directions, axially, in the separating zone, and discharge at opposite ends. Both designs have strong proponents and arguments. For the this application, countercurrent flow is assumed.

Conventionally the front end of the centrifugal separator is the liquid discharge end and the solids end is referred to as the rear. The solids discharge is more usually referred to as the solid. While defining flow and positional conventions, it is worth mentioning that later in the book when discussing the interior of the bowl, terms such as "up", "over", and "bottom" for instance will be used. These terms relate to the centrifugal field, and thus "bottom" refers to the bowl wall, "up" and "over" mean towards the bowl axis.

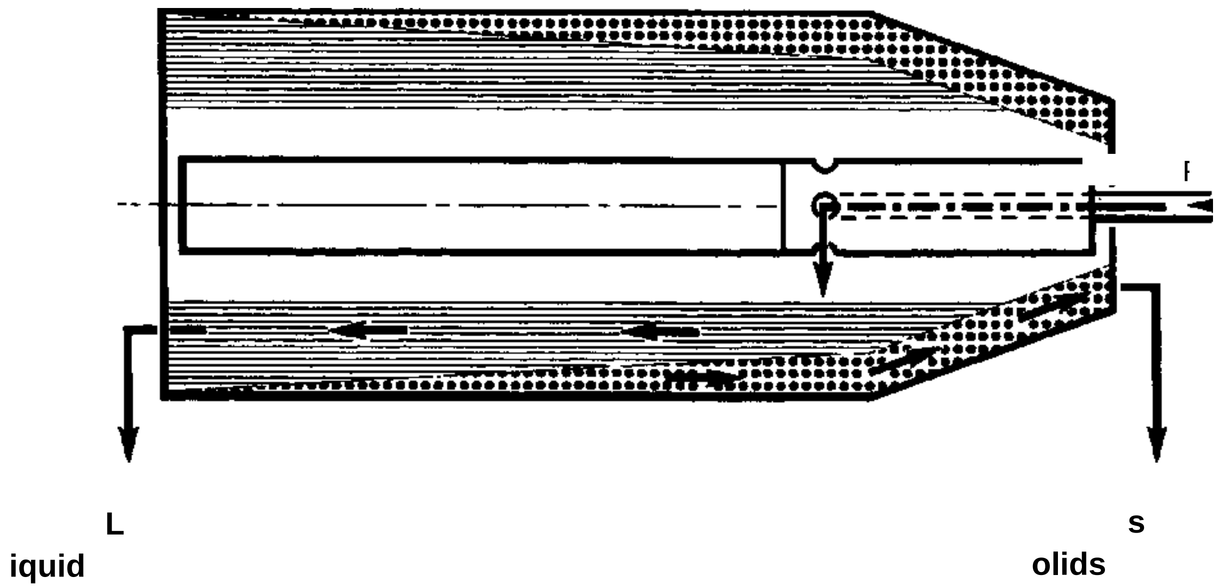

Figure 4.3.Countercurrent flow.

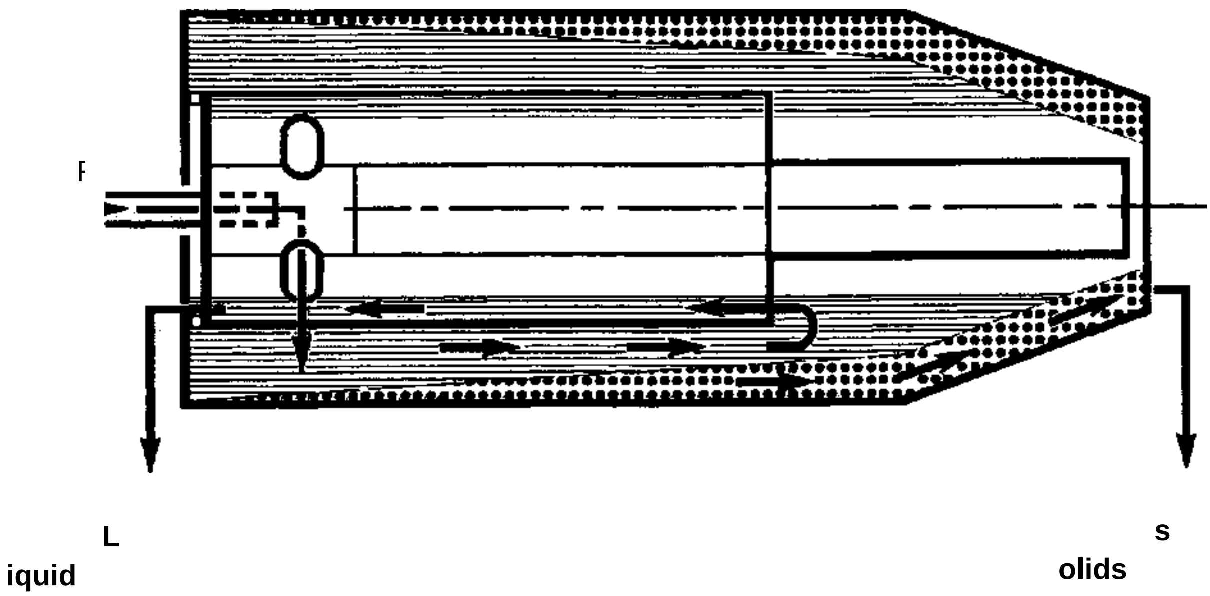

Figure 4.4.Co-current flow.

4.2.3 Materials of construction

Materials of construction of the centrifugal separator are many and varied. It is more usual to make the contact parts, particularly in the rotating assembly, of some form of stainless steel. This is to avoid assembly problems and misalignment due to corrosion on mating surfaces. This has to be avoided with high speed rotating equipment. Nevertheless, it must be said that there are many centrifugal separators in operation with bowls of carbon steel, where their manufacturer claims to be able to overcome, or avoid, corrosion. For stationary contact components there is no need for a high grade of stainless steel. When the process used is non-arduous, simple neoprene seals and gaskets will suffice. Supporting framework will be in ordinary or even cast steel. For parts that goes into housing and screw austenitic steel 316L is chosen with the following characteristics:

STAINLESS STEEL - 316 / 316L

RELATED SPECIFICATIONS:

Germany 316 - W.Nr 1.4401 X5CrNiMo17 12 2
316L - W.Nr. 1.4404 X2CrNiMo17 12 2

Great Britain 316 - BS970 – Part 3 – 1991 316S31
316L - BS970 – Part 3 – 1991 316S11

Japan JIS G 4303 SUS 316
JIS G 4303 SUS 316L

USA ASTM A276-98b 316/316L
AISI 316L
316L - UNS S31603

DESCRIPTION:

316/316L is an 18/8 austenitic stainless steel enhanced with an addition of 2.5% Molybdenum, to provide superior corrosion resistance to type 304 stainless steel. 316/316L has improved pitting corrosion resistance and has excellent resistance to sulphates, phosphates and other salts. 316/316L has better resistance than standard 18/8 types to sea water, reducing acids and solution of chlorides, bromides and iodies.

APPLICATIONS:

Include pumps, valves, marine fittings, fasteners, paper and pulp machinery, and petro chemical equipment.

Chemical analysis:

Steel 316L

	C %	Si %	Mn %	P %	S %	Cr %	Ni %	Mo %	N %
Min.	-	-	-	-	-	16.00	10.00	2.00	-
Max.	0.03	1.00	2.00	0.045	0.03	18.00	14.00	3.00	0.10

Mechanical properties:

316L(Annealed to ASTM A276)

Finish	Dia or Thickness mm	Tensile Strength Mpa min.	Tensile Strength Mpa min.	Tensile Strength 50mm% min.
Hot finish	All	515	205	40
Cold finish	12.7	620	310	30
Cold finish	12.7	515	205	30

SIZES AVAILABLE:

Round bars supplied cold drawn to h9 condition up to 25.4mm, smooth turned and polished up to127mm and peeled over 12.7mm.Square and Hex bars are cold drawn to h11 tolerance.

Size Range: 6.35mm – 310mm

4.2.4 Bowl assembly

The bowl in a modern centrifugal separator is a cylindrical tube with a flange at either end, on which are bolted at one end the liquid discharge bowl hub, and, on the other end, the solid discharge hub, or the conical drying zone followed by the solid discharge hub (fig4.5) . The first cylindrical bowls used a filler piece in the end of the bowl to form the conical drying zone. On modern bowls, particularly the larger ones, the conical drying zone is bolted to a flange at one end of the cylindrical section, although with some overlap to provide mechanical location.

The thickness of the bowl wall is dictated by the material of construction used, the maximum speed at which the bowl will be rotated, and the maximum weight of process material, feed, liquids or solid, likely to be held in the bowl. Thus the density of the process materials in use can have a major effect on the safe working speed of the bowl.

The inside surface of the bowl can be plain machined. However, some effort is often made to encourage solid to stick to the bowl, to aid scrolling instead of slipping round with the scroll. The means of doing this could be by knurling the inside of the bowl for instance. This can wear smooth relatively quickly. More often longitudinal ribs are welded, or a liner with similar ribs is fitted.

At each end of the bowl the outside bowl diameter can be increased to provide, if necessary, excess metal for removal during balancing. In' particular, it can provide a position for machining grooves, which will mate with corresponding baffles in the casing. Together, the grooves and baffles form labyrinths to counteract cross-contamination of the products discharging at either end of the casing.

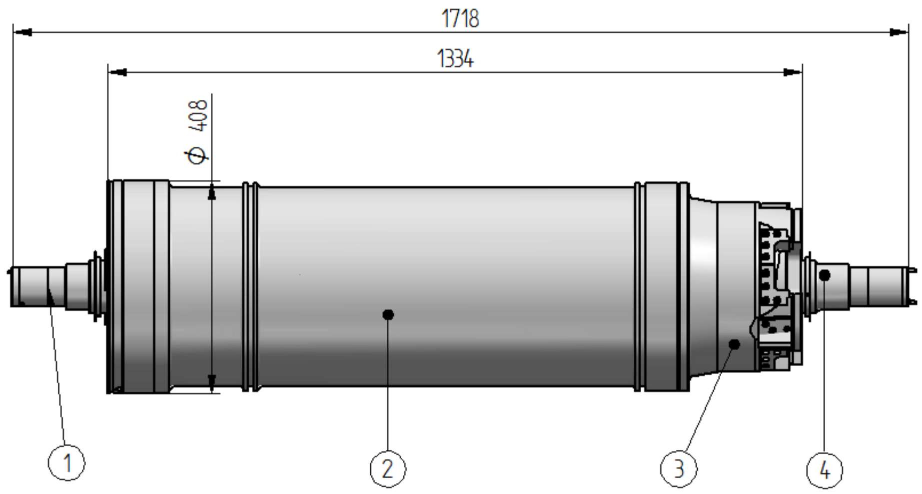

Fig.4.5 Basic bowl assembly

1.Front hub
2. Bowl shell
3.Conical drying zone
4.Rear hub

4.2.5 Front hub assembly

The front hub (the liquid discharge hub) bolts to one end of the bowl (see fig 4.6). It has an inner spindle to locate the scroll, its bearing and seals, and an outer spindle for the fitting of the front main bearing and pillow block. Seals will also be fitted to the outer spindle as required. The discharging liquid is commonly.

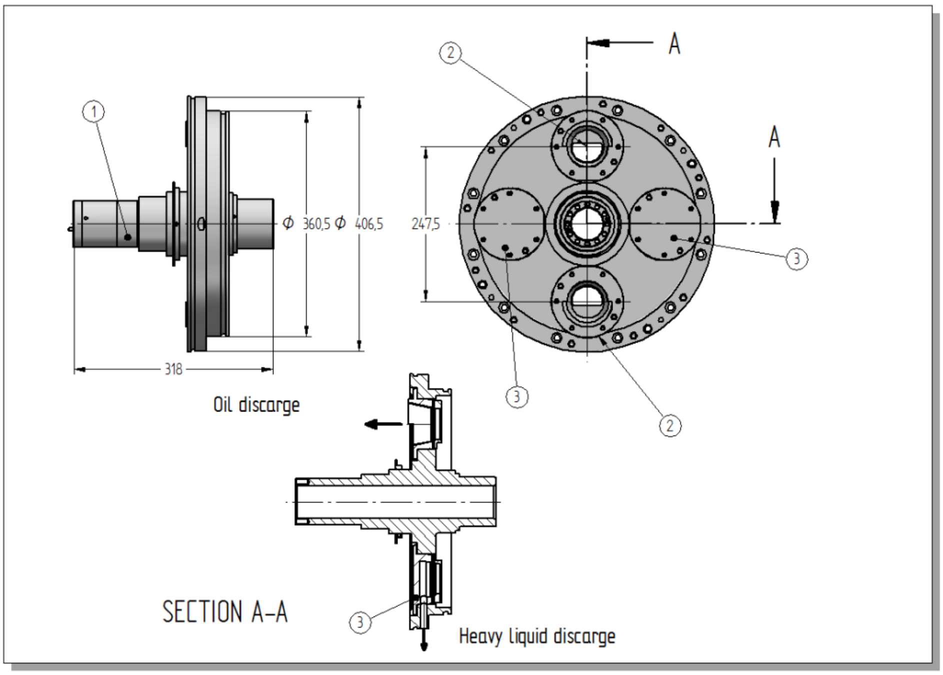

Figure 4.6.The centrifugal separator front hub assembly.

1.Front hub body
2. Oil weirs
3.Heavy liquid discharge weirs.

4.2.6 Liquids weirs

In a basic three-phase centrifugal separator the liquids discharges from the front hub over weirs. The oil weirs(fig.4.6,poz.2) cause a pond to form in the bowl. The level of oil in the bowl, the distance from the bowl wall to the inner edge of the weir, is known as the pond height. The pond level is adjusted using diaphragms.

Accurate location of the weir plates is necessary to enable adjustment of the pond level to within, say, 1 mm or better.

For best process control, the weir width needs to be maximised to reduce the level of cresting over the weir. The crest is the extra level of liquid above the weir inner edge, necessary to effect flow, as seen over weirs in rivers. This cresting varies with feed rate, but will be an inverse function of the weir length. Thus the larger this is, the smaller is the variation due to feed rate, or more properly, liquids flow changes.

The weirs for heavy liquid phase (fig.4.6,poz.3), are provided with holes arranged on the circumference of the bowl.

Section of these holes is adjustable depending on the feed rate and composition of the mixture processed, so that the separation surface between the two phases to be stationary.

If discharge holes of heavy liquid phase is clogging, the centrifugal separator can be used as dicanter.

4.2.7 Bowl shell

The bowl shell is a main part of the centrifugal separator .On the inside surface of the bowl shell are welded longitudinal ribs(Fig 4.7). The ribs are provided to combat erosion, but more particularly to form a key for the settled solid to improve scrolling efficiency. The ribs will be held in position in the bowl by tack, or spot, welds. On the smaller sizes of centrifuge the liner will be full length. On larger machines it can be full length, but sometimes it will cover only a partial length of the bowl from the conical drying zone junction forward to a little way past the feed zone.

The diameter of the scroll and the profile of the larger end of the conical drying zone need to be adjusted to accommodate the bowl shell.

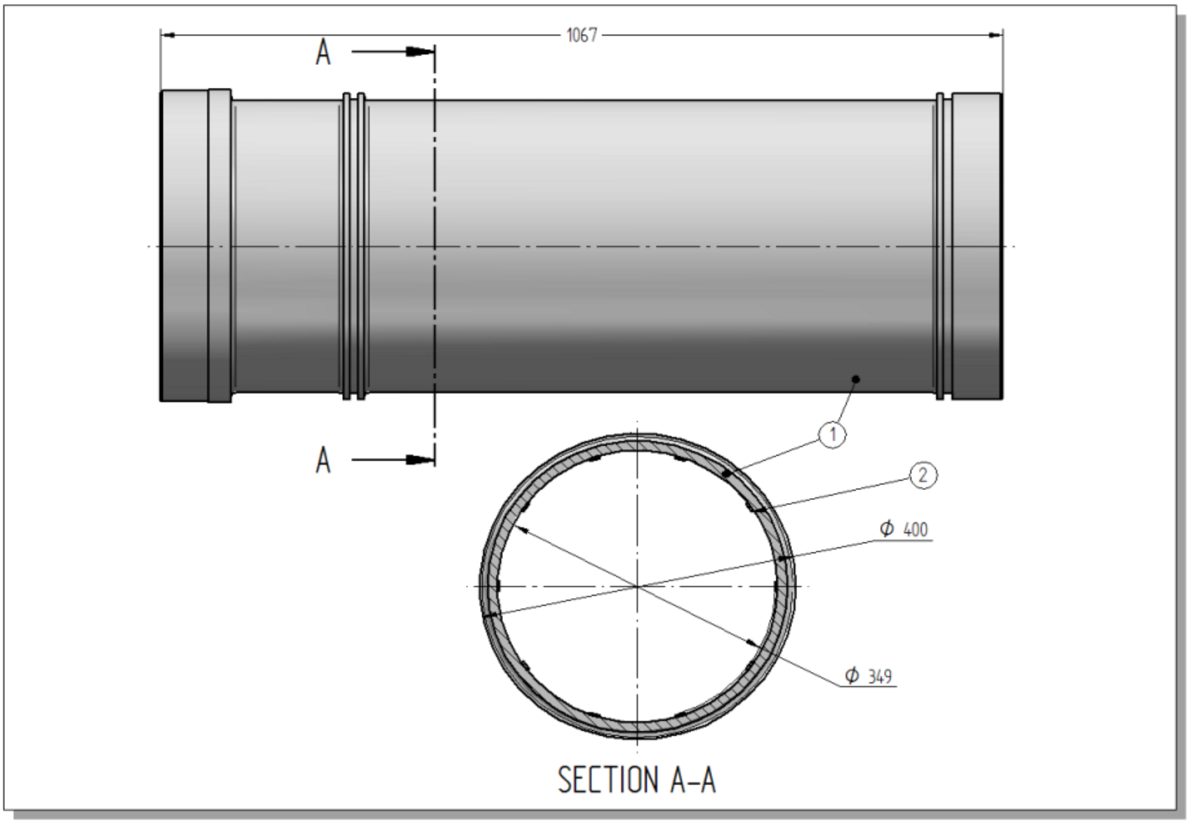

Figure 4.7. The bowl shell with ribs

1. Bowl shell
2. Ribs

4.2.8 Hub pillow block bearings

One of the two pillow block bearings (fig 4.2,poz.4) is fitted to the front hub. It is supported in a housing and sealed with a non-contacting flinger, wind back and labyrinth parts on each side. The housing is accurately mounted to the main frame and aligned with the bearing housing at the opposite end of the rotor.

The bearing is chosen to result in a long L10h life (above 100000 hours) at its speed and load conditions. For heavy loads lubrication is usually by oil, static, circulating or mist. Circulating oil, while usually the most expensive, is the best and most reliable. Most actual bearing failures are due to lubrication failure or foreign contamination, not load. A circulating system flushes out contaminants and introduces only cooled, filtered oil to the bearing. The oil drains from each housing must be large enough to discharge the oil quickly, after it passes through the bearing.

For this application was adopted, due to relatively low loads, bearings lubricated with grease.

Bearing housing seals must have sufficient axial clearance to permit thermal expansion of the rotor, and at least one bearing must float axially.

All well-designed centrifugal separators permit the re-greasing of the scroll bearings without the requirement to disassemble the casing.

4.2.9 Conical drying zone

The conical drying zone (fig. 4.8) is the conical section at the end of the bowl, and is considered a part of the bowl assembly. The front hub and the conical drying zone together enable a pool of liquid to be held in the bowl. Being a component in contact with the process liquid, the conical drying zone will be fabricated in the same material as the bowl.

The conical drying zone will be flanged and bolted to the end of the bowl or inserted into the end of the bowl as a filler piece.

To the rearmost end of the conical drying zone is fitted the bowl's rear hub. There are a number of possible configurations involving these two components, to

facilitate the solid discharge. The discharge holes can be round, slotted or specially shaped. These holes, or ports, are generally in the conical drying zone, but can be in the rear hub, or both.

The half included angle of the cone shape of the conical drying zone is commonly known as the conical drying zone angle. A different conical drying zone angle, or a combination of angles in a compound conical drying zone, could be selected to facilitate better dryness, better washing, or perhaps easier scrolling, depending upon the process application. A conical drying zone angle of 8 to 10 degrees is a common value chosen for many processes. The conical drying zone is usually ribbed or grooved to assist in conveying the solids..

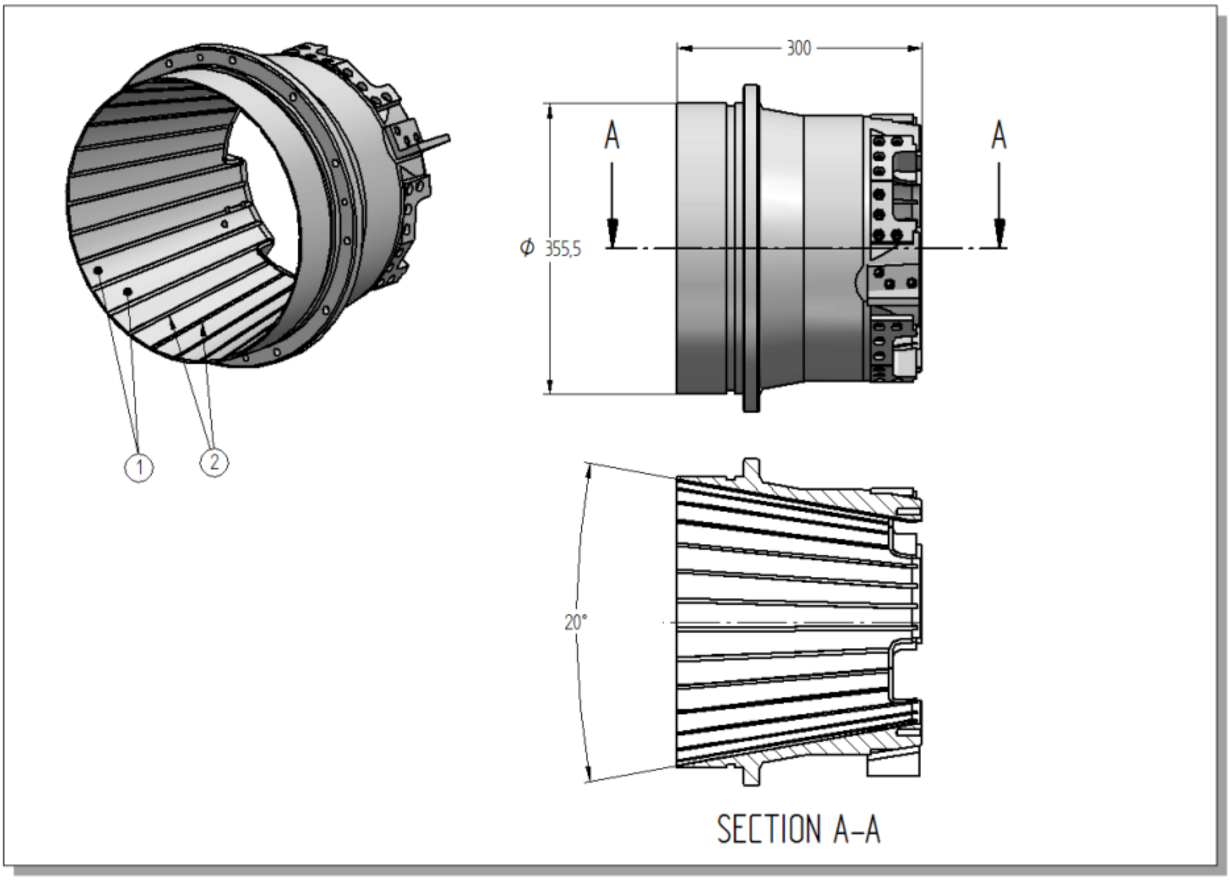

Figure 4.8 The centrifugal separator conical drying zone with grooves.

4.2.10 Solid discharge

The solid discharges at the rear of the conical drying zone, between the conical drying zone proper and the rear hub. In its simplest form, the solid discharge will be a series of radial holes around the conical drying zone end. These holes will usually be lined with some form of erosion protection, quite often in the form of a sintered tungsten carbide cylinder in a steel holder.

For process reasons it is important to have a defined solid discharge diameter. This is the diameter of the inner edge of the conical drying zone (radially, outer axially), over which the solids decant into the casing. Thus, prior to the discharge ports will be a ring or ledge providing a definite discharge level.

4.2.11 Scroll

The scroll (or conveyor) is in the form of an Archimedean screw, fitting inside the conical drying zone and bowl between the two end hubs, with a small clearance of less than 2 mm radially. It has a number of functions. Not only does it convey the solids, after they form a dry solids, along the cylindrical bowl section and up the conical drying zone, it also accepts the feed and accelerates it up to bowl speed.

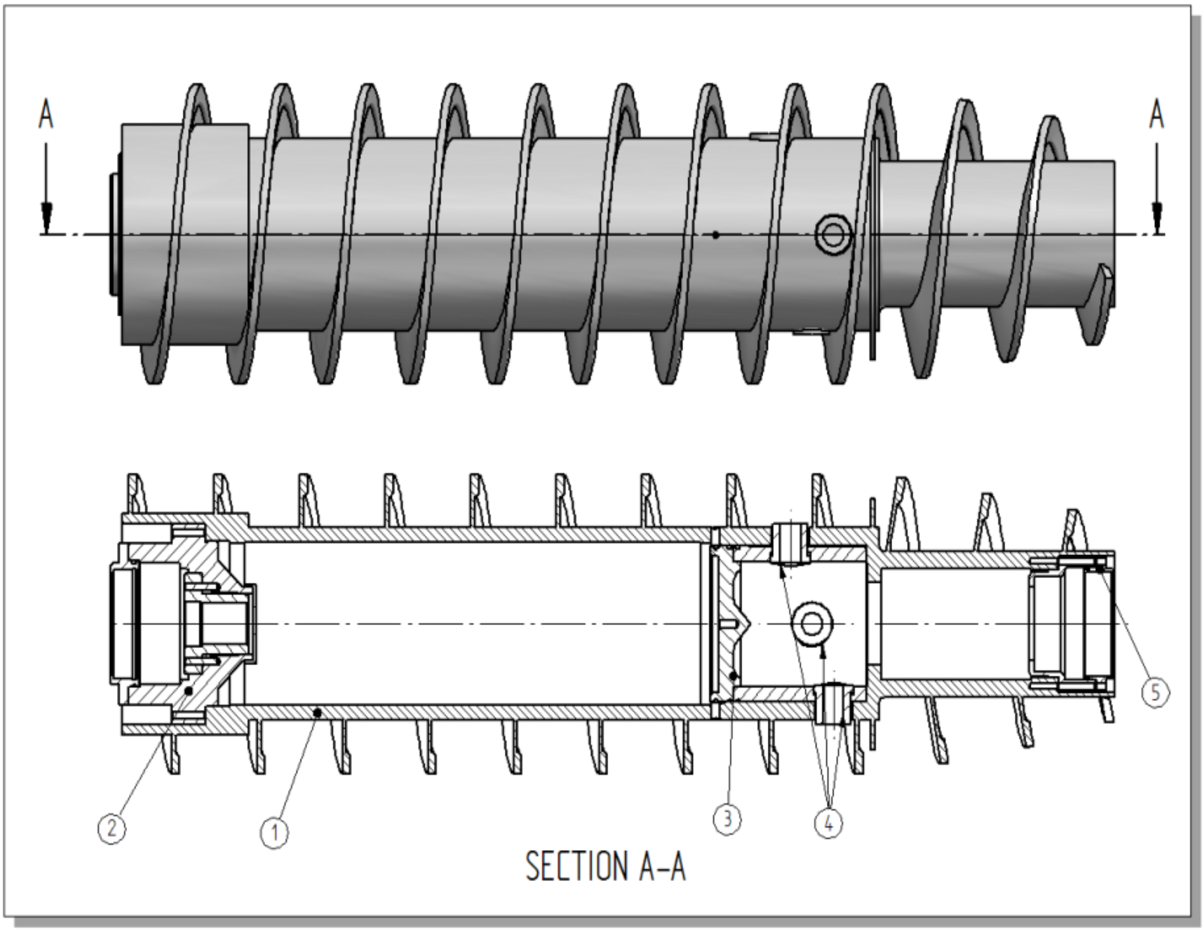

Figure 4.9. The scroll assembly

1. Scroll hub
2. Front scroll bearing housing
3. Feed zone
4. Feed ports
5. Rear scroll bearing housing

In its simplest form, the scroll(fig. 4.9) has a cylindrical central hub with a set of flights welded onto it, to form one continuous helix. The scroll bearings and associated seals are housed in both ends of its central hub. Somewhere in between the bearings will be a chamber called the feed zone, sealed and isolated from both bearings.

In some applications, where the solid particles are too fine to separate on their own, it is necessary to use a flocculating aid. The flocculant can be added upstream of the centrifugal separator, but there are many circumstances where, for best efficiency, it is admitted in the bowl. On these occasions there will be an extra chamber, the "floe zone", built into the hub of the scroll. Where necessary this floe chamber can be used as a rinse chamber instead, to admit rinse liquid onto the scrolling solid.

4.2.12 Scroll hub

This part of the scroll is a substantial tubular steel construction. It may be tapered at the conical drying zone end. It could, if necessary, be enlarged in diameter at each end to take the scroll bearings.

In each end of the scroll hub will be the scroll bearings with their associated seals. Adjacent to one of the bearings (in the basic design it will be the front bearing) will be some form of bushing. This bushing could be splined, keyed or specially shaped, e.g. lobed, to mate with the gearbox shaft, and so provide the scroll drive.

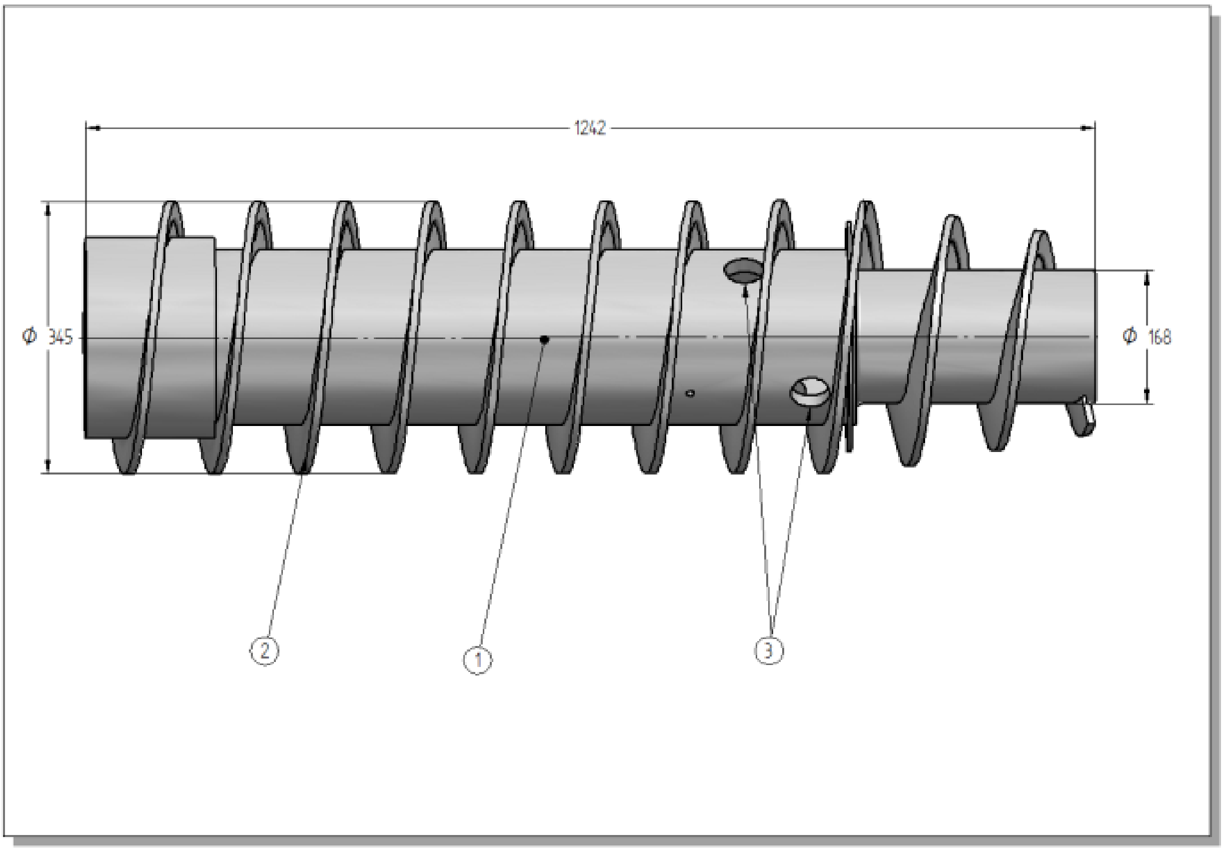

Figure 4.10. The scroll hub

1. **Scroll hub body**
2. **Flights**
3. **Feed ports**

The feed zone will be built into the hub to discharge at the start of the cylindrical section of the bowl adjacent to the conical drying zone. Next to the feed zone, a second chamber for flocculant or rinse may be fabricated within the hub. A buffer chamber between the feed and additive chamber will sometimes be built, with simple exit ports into the pond. By putting distance between the feed and the additive chamber by use of the buffer chamber, there is less chance of the additive chamber being contaminated by feed material.

The natural vibrational frequency of the scroll can be a limiting feature controlling the maximum speed of the centrifuge. This becomes especially critical when the *L/D* (length to diameter) ratio reaches 4.0 and more, and modern centrifugal separators are getting longer in order to give higher separating capacity. If the hub diameter gets smaller, the scroll flexibility increases, thus lowering the natural frequency. Increasing the hub diameter will solve this problem, but with modern centrifugal separators using deeper ponds in many applications, the hub becomes immersed in the pond. Immersed hubs can result in more hydraulic turbulence, and thus lower separation due to friction on the liquid surface, and possible build-up on the hub due to a sticky floating phase. Surface non-concentricity results in mechanical vibration due to non-symmetrical buoyancy effects, so high precision is needed in geometry. Air flow and degassing of the feed stream become more complicated with submerged hubs. Some new designs avoid these problems by permitting small hubs designed with high stiffness and high natural frequency .However, within the last decade, immersed hubs have been designed to float on the pond, considerably reducing potential vibration and enabling higher speeds.

4.2.13 Flights

The scroll flights are fabricated from segments of annular discs suitably stretched and welded end to end, to form a regular helix. Naturally the helix profile has to be tapered to suit the conical drying zone section. Each section is welded in turn to the scroll hub and then welded to the adjacent section., Double welding (both sides) with grinding afterwards is essential where hygiene is of importance. However double welding is common practice, even when hygiene is not required.

The flights will be normally perpendicular to the centrifugal separator axis or bowl wall. On the conical drying zone the flights will either remain at 90° to the axis or will be at

90° to the conical drying zone surface, depending upon the centrifugal separator manufacturer's choice.

It is not always appreciated what a complex shape the surface of a flight is. The usual pitch angle for a centrifugal separator is a little over 5° . The pitch angle is the angle the tip of the flight subtends to a right circle of the bowl. To maintain the flight at a constant angle to the axis, the pitch angle of 5° nearly doubles at the root of the flight.

If the flights are not to be protected from wear, then their tips will be ground smooth and perhaps chamfered, to provide a minimum of area in contact with the heel, to minimize torque.

4.2.14 Feed zone

There are a large number of designs, both complex and simple, for the feed zone. The feed enters the feed zone (fig.4.9 poz.3) chamber from the feed tube. Once in the feed zone, it has to be accelerated up to the bowl speed before spilling into the pond via the exit ports. To assist the feed up to speed, accelerator vanes will sometimes be found on the "target", the plate opposite the feed tube end. These vanes could be radial, at an angle to the radii, or curved.

In extreme cases of wear, parts inside the feed zone are hard surfaced or specific erosion resistant components are used. Hard surfacing is often used on the accelerator plate, particularly on leading edges and the tips of the vanes, where most wear takes place. Some shaped accelerators have been made completely of urethane rubber.

When the feed leaves the feed tube, in most cases it is at a high axial velocity. When it hits the rotating target, some splashing inevitably occurs. In fact a dense aerosol mist is often produced. At the back of the feed zone a tube is sometimes built in, to surround the end of the feed tube. On the outside of this tube, small accelerator vanes are welded to accelerate and condense the mist and also to accelerate liquid up to speed, should the feed zone become flooded. Ideally, air is allowed to enter the feed zone from around the feed tube. It will be drawn in by the fan effect of the feed zone and thrown out of the feed zone exit ports. The air would then pass along the bowl to exit over the liquids. This draught helps to prevent splash back of feed from the feed zone.

The exit ports from the feed zone are themselves subject to a considerable range of designs and innovations. It is not usual to have just one exit port. For symmetry and balance an even number of ports is usual, two, four, six or eight. The basic design has these ports fitted with tubular nozzles, often lined with a ceramic wear protection.

New feed zones have been introduced recently to reduce feed particle attrition, by slowing and extending the acceleration time to bring the feed up to speed, and to reduce the inlet turbulence in the separation zone.

4.2.15 Wear protection

Various levels and grades of wear protection may be applied to the scroll depending upon the application. The main areas on the scroll requiring protection are the feed zone, flight scrolling surfaces and the flight tips. Mechanically interchangeable wear inserts are more economical to replace than welded or bonded wear protection.

4.2.16 Gearbox

The gearbox is a major component of the rotating assembly, which creates the differential speed between the bowl and the scroll.

There are two main types of gearboxes used on centrifugal separators. These are the epicyclic gearbox and the Cyclo gearbox, made by Sumitomo of Japan. However there are a number of centrifugal separators which have eliminated the gearbox by using a hydraulic system called a Rotodiff manufactured by the Swiss company Viscotherm.

The epicyclic system consists of a pinion shaft and gear, which engages three planetary gears (mounted on carrier plates) which in turn engage a ring gear fixed to the gearbox casing. For the centrifugal separator the epicyclic gearbox involves two stages, although recently three stages have been in use. The carrier plate of the first stage holds a second pinion shaft carrying the sun gear for the second stage. The ratio of the gearbox is the product of the ratios for each stage. The maximum practical ratio for any stage is just over 13, giving a maximum ratio for a two-stage epicyclic gearbox of 170 to 180.

If the central pinion shaft is held stationary, the differential speed between scroll and bowl will be the bowl speed divided by the gearbox ratio. If the pinion shaft is allowed to rotate at some speed below the bowl speed, then the differential between bowl and scroll will be the difference between bowl speed and pinion speed, divided by gearbox ratio.f If the pinion speed is controlled by using a brake, or a variable speed motor, differential speed may be varied from close to maximum, when the brake is at its slowest speed, to nearly zero, when the brake is almost at bowl speed. Reducing the pinion speed below zero, i.e. by reversing, enables higher differential speeds to be obtained. Using an epicyclic gearbox causes the scroll to rotate slower than the bowl, whereas it is normally faster when using a Cyclo gearbox. Generally the scroll flight helix is "left handed" with an epicyclic gearbox and right handed with a Cyclo gearbox.

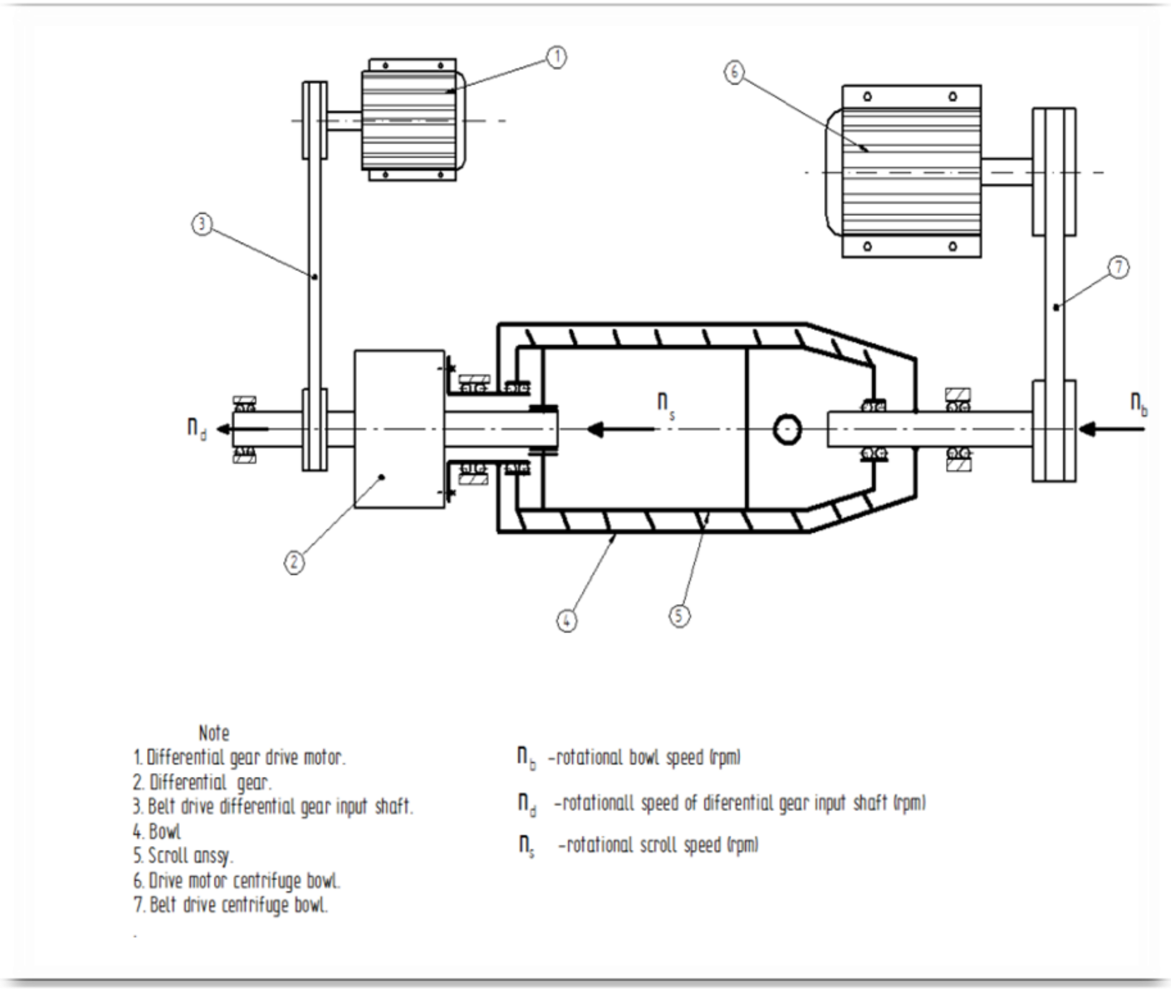

Fig. 4.11 Kinematic diagram of centrifugal separator

Kinematics scheme of a centrifugal separator with a scroll flight helix on the left handed and equipped with a epicyclic gearbox is presented in Fig.4.11.

With the sign convention shown in Fig.11 overall transmission ratio is:

in=(nb-nd)/(nb-ns) [4.1]

For a Cyclo gearbox overall transmission ratio is:

in=-(nb-nd)/(nb-ns) [4.2]

For our application overall transmission ratio is within the range of 50 to 75.

If is adopted a cyclo gearbox can maintain the same configuration as in fig.4.11, with the only difference that reverses the rotation of the gerarbox drive motor.

4.2.17 Frame

On smaller centrifugal separators, the frame has often been made from cast iron. More usually it is fabricated from steel channel or box sections. The frame needs to be a rigid support for the rotating assembly. The surfaces for the main bearing pillow blocks are accurately machined in the same plane, and inline, to ensure no end-to-end misalignment of the rotating assembly, which would cause premature bearing failure. Some manufacturers fill part of the main frames of their larger machines with concrete effectively to form an inertia block, and for noise reduction. Some frames have been used as a reservoir for the lubricating oil for the main bearings.

4.2.18 Feed tube

In its simplest form the feed tube is a plain cylindrical tube. A clamp or flange holds it on a support extension from the main frame. It extends to the feed zone and within a few centimeters of the accelerator in the feed zone

The geometry of many centrifugal separators is such that there is a risk of resonant vibration of the feed tube at frequencies around the bowl speed. To counter this, feed tubes have been made slightly tapered, and made of lighter materials such as glass fibre and even carbon fibre, and sometimes composite material.

Entering the scroll with the feed stream is a flow of leakage air, which passes through the clearance between the scroll and the feed tube. This air flow must eventually be vented, and in those applications where odour or toxicity is an issue, minimising this air in-flow is important. A simple, lightly contacting lip seal is often used. On critical sealing processes, mechanical seals are required.

4.2.19 Casing

The casing acts as the collector for the products discharged from the rotating assembly, and channels them to receivers for onward handling. The casing must, obviously, keep the separated products apart.

Simply stated, this casing is a stationary collector for solid at one end, and liquids at the other. There are many design variants and each manufacturer has its own recognisable design whether it be just its finish or its shape and functionality.

In its more usual format, a lid is hinged onto a bottom half and bolted with a flat rubber gasket between the two halves.

4.2.20 Main drive

The main centrifuge drive is usually an electrical motor mounted on slide rails on the frame or sub-frame and connected to the centrifuge drive pulley with V-belts. A purpose built belt guard will cover the two pulleys and belts. Motors on the larger centrifugal separators can consume a few hundred kilowatts of power. Such large motors are more often directly mounted on the floor, in which case a special belt-tensioning device is incorporated to allow for the differential movement of the rotating assembly.

The main motor has to accelerate a high inertial load on start-up. When the bowl is at speed; the main motor has to provide the power to accelerate the process material up to speed, the power for scrolling the solid, and most of the braking power for the back-drive system.

There are various methods of starting a centrifugal separator with an inherently high inertial load. A variable speed drive might be considered, if changes in machine speed are deemed to be important due to varying process conditions. However, a large majority of centrifugal separator applications use a fixed speed main drive motor. Once at speed, a centrifugal separator's motor has one of the simplest duties. It is never subjected to cyclic overloads, never subjected to continuous vibration and never subjected to severe braking, electrically or mechanically. It is seldom stopped and restarted.

There is, however, one duty that the main motor has to perform that differs from most other drive motor applications. It has to have the thermal capacity to accelerate a high inertial load to speed, smoothly over an extended period of time, without undue damage to motor windings by excessive temperature rises. This cannot be achieved without certain limitations, such as the number of starts from standstill to full speed being limited to a certain number within a fixed period of time. The user's requirements, such as installing the motor starter remotely, perhaps interlocking with other equipment starters and microprocessors, may add to the restrictions imposed on the main motor specification.

The time taken for the motor and centrifugal separator to full speed is an important factor in the selection of the main motor. It is dependent upon the inertia of the motor and centrifugal separator the torque/speed characteristics of the motor, the reactive torque from the centrifugal separator, windage and friction losses, and the speed at which the centrifugal separator is to be run.

Motors can be two-, four- or six-pole giving synchronous speeds of 3000, 1500 or 1000 rpm at 50 Hz, respectively. The most commonly used motor is the four-pole, which is a more usual standard in motors and is capable of being better balanced than the two-pole. Because of the low speed the six-pole motor would be an unusual choice.

The torque available from the main motor varies according' to the method of start up, whether it is star-delta or direct-on-line. A motor connected in star produces a starting torque one third of that provided when starting direct-on-line. Direct-on-line starting torque can be two and a half times the motor's full load torque, with a starting current of six times full load current.

A standard motor has a very steep direct-on-line torque/speed curve characteristic, rising from a minimum of 180-250% full load torque, to a maximum of 300-500% full load torque. This maximum, called the pull out torque, occurs at about 85% of full speed.

Main motors need starter overload and short circuit protection. High rupture fuses (HRC) will protect the motor against short circuit conditions, and will interrupt the electrical supply in milliseconds of the fault occurring. It is essential that fuses of this type are always fitted. Conventional overload protection, thermal or magnetic, can offer no protection to a motor with an extended acceleration time. Thermistor overload protection is the only true protection for a motor under these conditions. A thermistor is embedded in each of the motor's three windings and connected in series. The resistance of these thermistors is designed to increase rapidly at a set temperature, depending upon the insulation class of the motor. The thermistors are connected to an electronic amplifier control unit in the starter enclosure, and will trip the starter contacts when required. The device will not reset until the motor has sufficiently cooled.

In Europe the main motor is usually an AC motor, using a star-delta starter. An inverter for the main motor is becoming more common, particularly with the smaller centrifugal separators. The inverter enables a soft start, and allows speed adjustment for different process requirements. Inverter motors can cause unwanted electrical interference, and harmonic wave forms, on the main electrical supply lines. These problems can be minimised by using electrical filters and the latest advanced electrical technology.

4.2.21 Gear drive motor

The gear drive motor system is a means of controlling the speed of the gearbox pinion shaft (and thereby the scroll differential speed) using, for instance, a motor or a brake. This could be offset from the gearbox shaft, in the same manner as the main drive, and connected by a belt. This belt would be a timing belt because of the accurate control required. Normally the back-drive is connected directly and in line with the gearbox pinion

The main component of the back-drive assembly can be an eddy current brake, inverter motor or a DC motor. The Viscotherm Rotodiff hydraulic scroll drive is a variable speed device, powered by a fixed speed hydraulic motor.

Technical conditions which include drawings of the main parts that goes into centrifugal separator are presented in Annex 4.1

4.3 BASIC THEORIES

As with every specialist subject, it is easy for the centrifuge engineer to assume that his basic theories are universally understood. To ensure that this work is comprehensible to the widest possible readership, a few basic concepts will be covered. These will include G-force, differential speed, and mass balances across the centrifugal separator.

4.3.1 Acceleration force

The centrifugal acceleration force, commonly known as G-force, is the basic motive force for separating the solids from the liquid in any sedimenting centrifuge. Thus, in a handbook about the centrifugal separator no apologies are needed when covering G-force as the first basic concept.

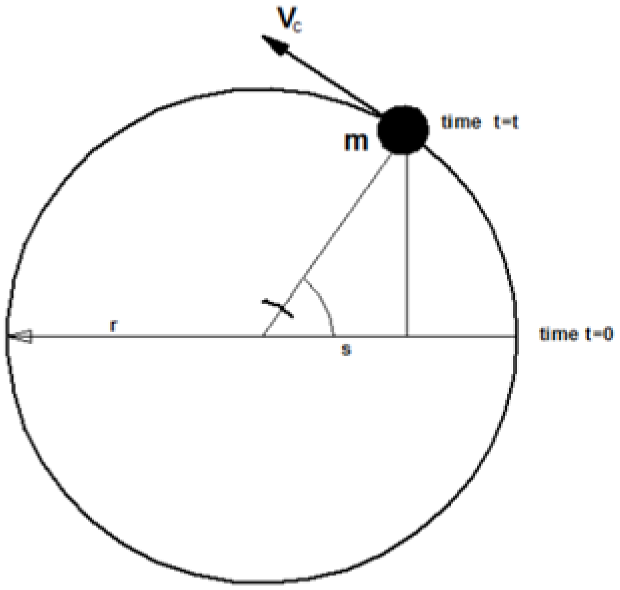

Fig 4.12 Dinamics of a particle moving in a circle

Consider Figure 4.12. A particle, of mass m, rotates at a tangential velocity, v_c and angular velocity, ω;, in a circle of radius, *r.* After a time, t, the particle has moved to a point on the circle radius, r, which subtends an angle, ϒ, where ϒ = ωt, from its position at time t=0, the extreme right of the horizontal diameter of the circle.

At time t, the horizontal distance of the particle from the centre of the circle is s, and at time *t=0* it was *r:*

$$s = r\cos(\gamma) \qquad (4.3)$$

$$s = r\cos(\omega t) \qquad (4.4)$$

The horizontal acceleration of the particle towards the centre is the second differential of s:

$$\frac{d^2 s}{dt^2} = \frac{d\{-\omega r \cdot \sin(\omega t)\}}{dt} \qquad (4.5)$$

$$= -\omega^2 \cdot r \cdot \cos(\omega t) \qquad (4.6)$$

At time t=0:

$$\frac{d^2 S}{dt^2} = -\omega^2 \cdot r \cdot \cos(0) \qquad (4.7)$$

$$= -\omega^2 \cdot r \qquad (4.8)$$

Thus, anything moving in a circle of radius r, at an angular velocity ω, will experience an acceleration towards the centre of the circle of ω^2 r.

In the centrifuge, it is the liquid that moves round in a circle, and the particles in suspension are free to move relative to the liquid. Thus, relative to the liquid, the suspended particles experience an acceleration, $\omega^2 r$, radially outwards.

Thus, the gravitational force, F, on a particle of mass m, is the product of its mass and acceleration, where:

$$F = m \cdot r \cdot \omega^2 \qquad (4.9)$$

In centrifuge parlance the term "G" (or G-level) is often used. This is the number of times the acceleration in the centrifuge is greater than that due to gravity alone. Thus:

$$G = \frac{\omega^2 \cdot r}{g} \qquad (4.10)$$

Note that the G-level within a centrifuge will thus vary, proportional to the radius, throughout the depth of the liquid, and is proportional to rotational bowl speed squared.

4.3.2 Differential

The difference in rotational speed between the bowl and the scroll is commonly referred to as the scroll differential speed, $N = n_b - n_s$. Scroll differential speed N is calculated from a knowledge of the rotational bowl speed, n_b, the rotational scroll speed n_s, and the gearbox ratio, i_n ,from equation (4.1) and (4.2).

To process a diverse range of polluted water compositions we propose that the value of N can be adjusted in the range 0-50 rpm.

When an epicyclic gearbox is used, the scroll rotates at a speed less than the bowl speed, while with a Cyclo gearbox the scroll rotates at a speed above the bowl speed. This fact can have an effect on process performance with short bowls, if the feed is not fully accelerated to bowl speed on entry. With the scroll rotating faster than the bowl, the liquid has to get up to speed to find its way to the liquid discharge hub. With scrolls rotating slower than the bowl, the liquid could wind its way around the helix of the scroll to the liquids discharge ports, without ever getting up to bowl speed. Note, to effect scrolling in the required direction, the flight helix on a scroll using an epicyclic gearbox would be left-hand, and with a Cyclo gearbox it would be right-hand (assuming conventional equipment and operation)(see fig. 4.11).

With modern technology, the speeds of the bowl and gearbox pinion can be continuously measured with tachometers or proximity probes, and their signals fed to a simple PLC to work out, and even control, the differential. The PLC would need the gearbox ratio programmed in to execute this duty.

4.3.3 Scroll torque

Scroll torque, T, is a measure of the force exerted by the scroll in moving the separated solids through the bowl, up the conical drying zone and out of the centrifugal separator. It equals the pinion torque, T_p, times the gearbox ratio:

$$T = i_n \times T_p \qquad (4.11)$$

It is not easy to measure scroll torque directly, whereas pinion torque can usually be obtained from instrumentation on the pinion braking system. This signal is often given to a PLC for control purposes. Scroll torque is a vital measure in the control of modern centrifugal separator systems.

4.3.4 Process performance calculations

Consider the two-phase centrifugal separator separation system in Figure 4.13. Input is the feed at a rate of Q_f, of density ρ_f, with a solids fraction x_f, and an additive, often a flocculant, of density ρ_p, at a rate of Q_p, with a solids fraction x_p. There are two products, solid at a volumetric flow rate Q_s, at density p_s, and solids fraction x_s, and a liquids at flow rate Q_b at density p_b with a solids fraction of x_l. From measurements of some of the eight process parameters mentioned, it is required to assess the performance of the centrifugal separator. It is normal to monitor the feed rate, Q_f, and the additive rate, Q_p, with flow meters. Periodically, gravimetric analyses are conducted on samples of feed, solid, liquids and, if necessary, the additive. Performance is judged by how high is the solids recovery, R, and how low is the flocculant dose, P_D, when this is used. Recovery is the percentage of solids in the feed that reports to the solid discharge. Flocculant dose, sometime referred to as polymer dose, is the amount of dry polymer used per unit dry solids in the feed, usually expressed as kg/t db (kilograms per tone dry basis).

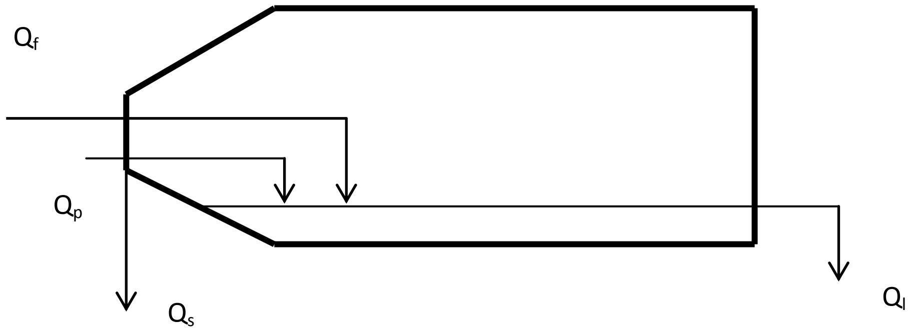

Figure 4.13.Centrifugal separator Mass Balance

As an intermediate parameter in the calculations, it is necessary to calculate the liquids rate, Q_l, by conducting a total and a solids mass balance across the centrifugal separator. Total mass balance:

$$Q_f \cdot \rho_f + Q_p \cdot \rho_p = Q_s \cdot \rho_s + Q_l \cdot \rho_l \qquad (4.12)$$

Solids mass balance:

$$Q_f \cdot \rho_f \cdot x_f + Q_p \cdot \rho_p \cdot x_p = Q_s \cdot \rho_s \cdot x_s + Q_l \cdot \rho_l \cdot x_l \qquad (4.13)$$

Eliminating $Q_s p_s$ from equations (4.12) and (4.13):

$$Q_l \cdot \rho_l = Q_f \cdot \rho_f \cdot \frac{(x_s - x_f)}{(x_s - x_l)} + Q_p \cdot \rho_p \cdot \frac{(x_s - x_p)}{(x_s - x_l)} \qquad (4.14)$$

Recovery of solids is calculated by subtracting the percentage loss of solids in the liquids from 100. Thus:

$$R = 100 \cdot \left(1 - \frac{Q_l \cdot \rho_l \cdot x_l}{Q_f \cdot x_f \cdot \rho_f}\right) \qquad (4.15)$$

Polymer dosage is given by:

$$P_D = \frac{Q_p \cdot x_p \cdot \rho_p}{Q_f \cdot x_f \cdot \rho_f} \qquad (4.16)$$

Polymer dose levels are frequently expressed in kg/t db, with the dry basis measure applying to both solids rate and polymer rate.

During these calculations, one must take care with the units used. Volumetric flow rates are invariably measured and the density terms are often ignored as they are usually close to unity. However, the density terms must be used when density values are significantly above unity. The gravimetric analyses of the samples should all be total solids (i.e. samples are evaporated to d the liquids generally are. However, liquids are generally analyzed in terms of suspended solids. Any dissolved solids in the liquids cannot be considered a measure of a centrifugal separator's inefficiency, as it is suspended solids which it separates, and any dissolved solids attached to the solid by virtue of its moisture content represents a bonus. Dissolved solids in the liquids are usually low and can be ignored, but when they are not, then they should be included in the equations when calculating liquids rate.

4.3.5 Particle Size Distribution

Very few process slurries contain particles of uniform size. A large proportion of slurries, processed by centrifugal separators, contain solids which have a particle size distribution which conforms closely to a logarithmic probability distribution. The logarithmic probability equation was derived by Hatch and Choate in 1929:

$$\frac{dz}{d\{\ln(d)\}} = \frac{Z}{\sqrt{2\pi}\ln(\sigma_g)} \cdot \exp\left[-\frac{\{\ln(d) - \ln(d_g)\}^2}{2\{\ln(\sigma_g)\}^2}\right] \qquad (4.17)$$

where Z is the total number of particles; *d* is the particle diameter; σ_g is the geometric standard deviation; d_g is the geometric mean diameter; and *z* is the number of particles less than diameter *d.*

Integrating this equation gives the formula for a cumulative number distribution:

$$C_n = \frac{1}{2} + \frac{1}{2} erf\left[\frac{\ln\left(d/d_g\right)}{\sqrt{2}\ln(\sigma_g)}\right] \qquad (4.18)$$

where C_n is the cumulative fraction of the number of particles below size *d*; and **erf** is a tabulated integral from -1 to +1.

It can be shown, by using equation (4.17), that the equation for the cumulative weight distribution is similar:

$$C_w = \frac{1}{2} + \frac{1}{2} erf\left[\frac{\ln\left(\frac{d}{d_g} \right)}{\sqrt{2}\ln(\sigma_g)} - \frac{3 \cdot \ln(\sigma_g)}{\sqrt{2}} \right] \quad (4.19)$$

where C_w is the cumulative weight or volume of particles below size *d.* In verting and simplifying equation (4.19):

$$\ln(d) = a_1 + a_2 erf^{-1}(2C_w - 1) \quad (4.20)$$

where$a_1$and $a_2$are constants, functions of $d_g$and σ_g.

Since Hatch and Choate first published their equation, special graph paper has been developed and printed whereby plotting the cumulative percent of particles by number or weight, oversize or undersize, against particle size, results in a straight line. The mathematics of the distribution are such that one can readily transfer between weight and number distributions, and even area and diameter distributions.

The diameter at the 50% line on the graph gives the geometric mean diameter for the type of distribution plotted, be it number, weight or whatever. The geometric standard deviation, which is the same for all types of distribution, is given by the size for which 84.23% of the number, weight, etc., is smaller, divided by the geometric mean size:

$$\sigma_g = \frac{d_{84.13}}{d_{50}} = \frac{d_{50}}{d_{15.87}} \quad (4.21)$$

The relationship between the various means is given by:

$$d_{gw} = d_g e^{3(\ln \sigma_g)^2} \quad (4.22)$$

$$d_{gs} = d_g e^{2(\ln \sigma_g)^2} \quad (4.23)$$

$$d_{gl} = d_g e^{(\ln \sigma_g)^2} \quad (4.24)$$

where d_{gw} is the geometric mean for a weight distribution; d_{gs} is the geometric mean for an area distribution; d_{gl} is the geometric mean for a diameter distribution; and $d_g$is the geometric mean for a number distribution.

Figure 4.14 shows a typical distribution plotted as a number, area, and weight distribution on the specially scaled graph paper. From a graph such as this, the two basic parameters, cr_g and d_g, can readily be obtained and from these more pertinent information can also be obtained.

For example, the total surface area, A_T, of the solids in the slurry can be calculated:

$$A_T = \frac{6}{d_{gw}} \cdot e^{\frac{1}{2}\{\ln(\sigma_g)\}^2} \quad (4.25)$$

This parameter is useful for paints and pigments, giving the covering power of the solids. It is also useful in assessing relative flocculant demand, as this is proportional to the surface area of the particles.

In general, the centrifugal separator removes the coarsest or densest particles from the slurry, leaving only the finest or least dense in the liquids. Knowing the required percentage solids removal from the slurry, the recovery, one can read the desired cut point from the distribution graph. This then gives an appreciation of the feasibility of the desired separation. Experience will tell whether the centrifugal separator would be capable of achieving the required cut point. For instance, a cut point of 2-5 µm would be feasible on a centrifugal separator with most slurries, but, say, less than 0.1 µm would probably be impossible, however high the density of the particles.

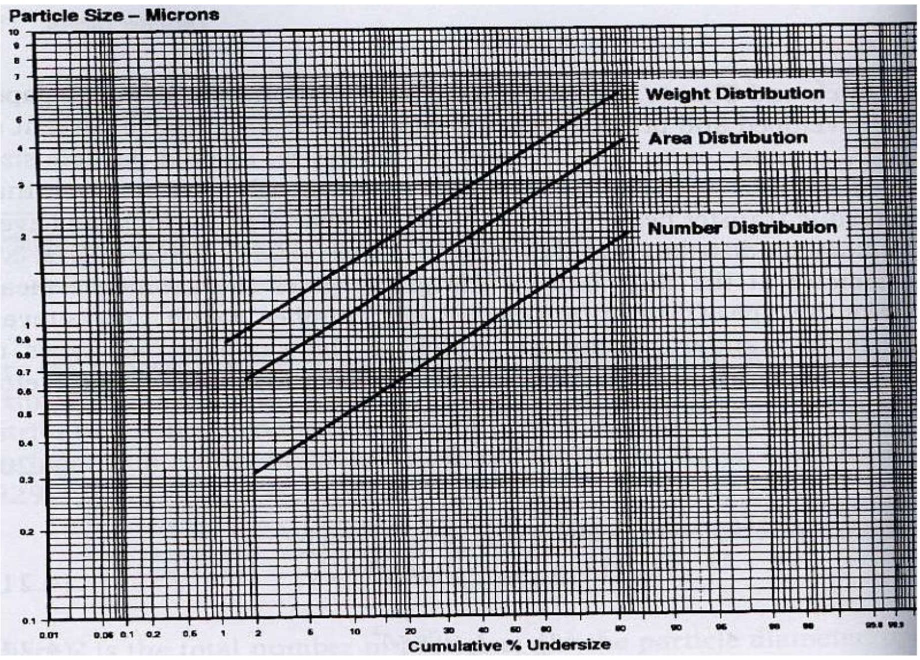

Figure 4.14. Cumulative weight, area and number log probability distributions

The cut point size is the smallest particle size that has to be settled in the centrifugal separator. Technically 50% of particles of that size settle and 50% are lost in the liquids; above that size the separational efficiency increases and below it vice versa. In consequence, the size distributions in both the solid and the liquids will also exhibit logarithmic probability distributions.

Figure 4.15 depicts examples of weight frequency distributions for feed, solid and liquids. Note that the liquids and solid lines intersect at the cut point size, and at a frequency level half that of the feed at that size. Hence, there is a 50% split between solid and liquids at the cut point.

These frequency distributions are plotted as cumulative weight distributions in Figure 4. 16.

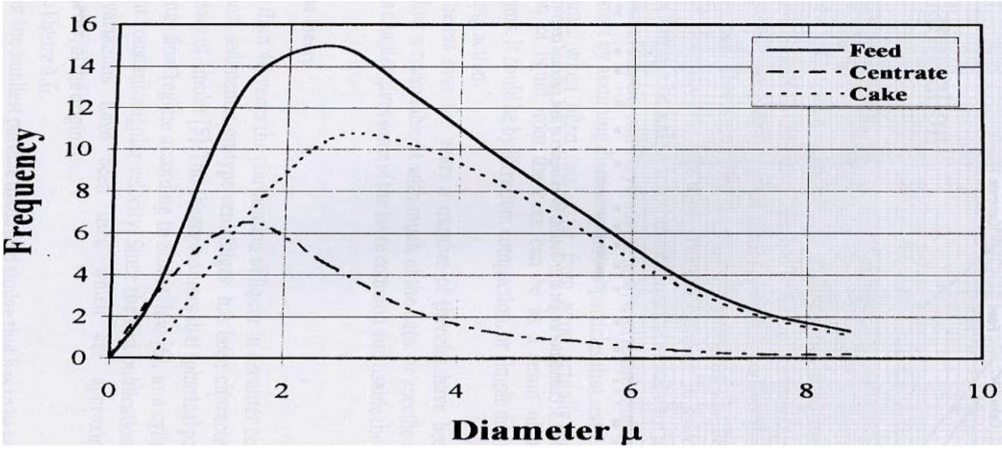

Fig. 4.15 Frequency distributions by weight for examples of feed, solid and liquids

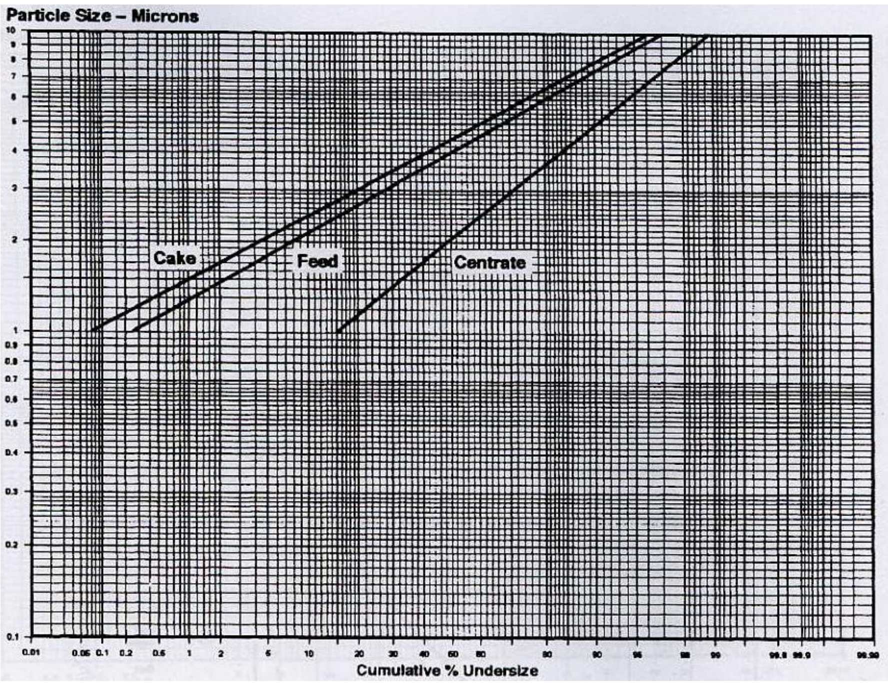

Fig 4.16 Cumulative weight distributions for examples of feed, solid and liquids

4.4 CLARIFICATION

The separation of solid particles and agglomerates from the suspending liquids, within a centrifugal separator centrifuge, has invited numerous theories. Few of these theories have provided exact results, which has given the opportunity for many more. This is no reflection on those providing the theories, as the system, let alone the processes used, are quite complex. In a centrifugal separator, liquid and solids flow in a helical path within a cylindrical vessel with a conical end. The rotational velocity can vary from the bowl wall to the pond surface. Many theories start by assuming discrete spherical particles that settle in a laminar flow regime, when often this is far from what actually happens. The expression of liquid from the solid can be as a result of one or more mechanisms. It could be by filtration, compaction, or simple drainage against the scrolling action.

Nevertheless over the years a number of theories have been proposed, which allow a reasonable fit with much of the data, or specific categories of data. This chapter gives some of the more common and usable theories.

4.4.1 Sigma theory

For more than 40 years the clarification of liquid in centrifugal separator centrifuges, in fact in most sedimentation-type centrifuges, has been characterized by the Sigma theory of Ambler. This theory assumes that spherical particles settle in a laminar flow regime according to Stokes' law , in a cylindrical bowl rotating at constant angular velocity. Since the first publication by Ambler, several variations have been used which are approximations, or developments, of the original.

Refer to Figure 4.17

Consider the smallest particle in the feed sludge that has to be separated, the cut point size, d_c. This particle has a density, ρ_s, and settles in a liquid of density, ρ_L, and viscosity, η_L. The feed slurry enters the centrifugal separator at a rate of Q_f, at a pond radius, r_1 at point X at time $t=0$. By the time the particle traverses the clarifying length of the centrifuge, L, in time $t=t_e$, the particle must settle to a radius r_2, at point *Y,* the bowl internal radius, if it is to be collected by the scroll. The centrifuge rotates at a constant angular velocity, ω.It is assumed that the fluid in the bowl also rotates uniformly at angular velocity *ω,* and travels along the bowl in plug flow. It is further assumed that the particle being considered is homogeneous and spherical, settling in a laminar flow regime.

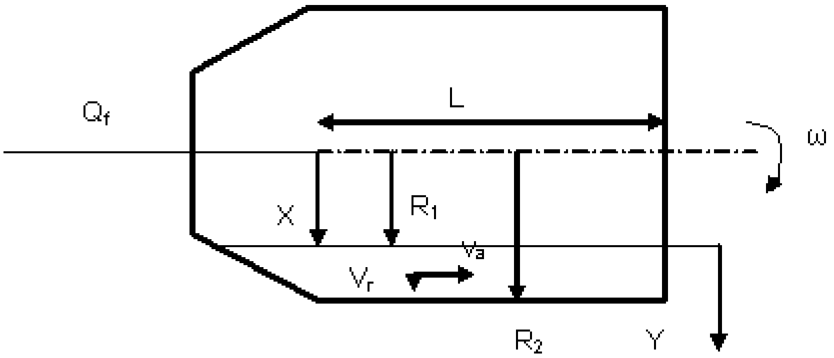

Figure 4.17. A particle settling in a centrifugal separator

At time *t=t,* the particle has a radial velocity v_r and a constant axial velocity of v_a. It is also assumed that the particle travels a negligible distance from the pond surface before it reaches its terminal velocity.

The axial velocity is given by:

$$v_a = \frac{Q_f}{\pi\left(r_2^2 - r_1^2\right)} = \frac{L}{t_e} \tag{4.26}$$

Thus:

$$t_e = \frac{\pi L}{Q_f}(r_2^2 - r_1^2) \tag{4.27}$$

The radial velocity at any time *t* is given by:

$$v_r = v_s \frac{\omega^2 r}{g} \tag{4.28}$$

where v_s is the Stokes settling velocity, given by:

$$v_s = \frac{d_c^2\left(\rho_s - \rho_L\right)g}{18\eta} \tag{4.29}$$

Where d_c is the particle size at the cut point; ρ_s is the density of the particle; ρ_L is the density of the liquid; and η is the viscosity of the liquid. Now:

$$v_s = \frac{dr}{dt} \tag{4.30}$$

Substituting equation (4.30) into (4.27) and integrating between the limits of $r=r_1$ to r_2 and *t=0* to t_e:

$$\ln\frac{r_2}{r_1} = \frac{\omega^2 v_s}{g} \cdot t_e \tag{4.31}$$

Eliminating t_e from equations (4.27) and (4.31):

$$Q_f = \frac{\pi L \omega^2}{g} \cdot \frac{\left(r_2^2 - r_1^2\right)}{\ln\left(r_2 / r_1\right)} \cdot v_s \tag{4.32}$$

The terms on the right-hand side of equation (4.32) consist of v_s, which is solely a function of the process material, and the remainder of the terms, that are solely functions of the centrifuge. These latter terms are collectively known as Sigma, ∑, the clarification capacity of the centrifuge:

$$\Sigma = \frac{\pi L \omega^2}{g} \cdot \frac{\left(r_2^2 - r_1^2\right)}{\ln\left(r_2/r_1\right)} \tag{4.33}$$

Sigma has units of area, which is consistent with non-centrifugal clarifying equipment.

From equations (4.32) and (4.33):

$$Q_f = v_s \cdot \Sigma \tag{4.34}$$

Equation (4.33) is the equation for Sigma preferred today, and is particularly recommended when deep pond centrifugal separators ($ri/r_2 \leq 0.65$) are used. When Ambler first derived his formula, only shallow ponds ($ri/r_2 \geq 0.75$) were used, and he used different starting assumptions for his derivation.

With a shallow pond it is assumed that the incoming feed distributes itself evenly throughout the depth of the pond, in the annular plane of the point of entry. The theory then develops the same equations in the same way, assuming that half of the particles of the smallest size that have to be separated will be removed. This is consistent with the definition of cut point.

The last particle of the half of the smallest particles to be settled will start at a radius $r_x$at the feed point, at which half of all particles will start inward and half outward of this point in this plane.

So:

$$r_2^2 - r_x^2 = r_x^2 - r_1^2 \tag{4.35}$$

whence:

$$r_x^2 = \frac{r_2^2 - r_1^2}{2} \tag{4.36}$$

Now substituting equation (4.30) into equation (4.28) again, but this time integrating between the limits of $r=r_1$and $r=r_x$and t=0 to $t=t_s$:

$$\int_{r_1}^{r_x} \frac{dr}{r} = v_s \frac{\omega^2}{g} \int_0^{t_e} dt \tag{4.37}$$

Solving equation (4.37), eliminating r_x using equation (4.36) and eliminating t_e using equation (4.28):

$$Q_f = 2\pi L \frac{\omega^2}{g} \frac{r_2^2 - r_1^2}{\ln\left\{2r_2^2/(r_1^2 + r_2^2\right\}} v_s$$

$$Q_f = 2 v_s \Sigma \tag{4.38}$$

where∑ this time is the true Ambler Sigma given by:

$$\Sigma = \pi L \frac{\omega^2 L}{g}(\frac{3}{4}r_2^2 + \frac{1}{4}r_1^2) \tag{4.39}$$

Compare ∑ of equation (4.39) with ∑ of equation (4.33). Note the extra numeral 2 in equation (4.39) compared with equation (4.34). This is to be expected if all the particles that have to be separated have the advantage of starting at half pond depth!

It will also be seen in the literature that approximations are sometimes made for the logarithmic term in equation (4.33) to give:

$$\Sigma = 2\pi L \frac{\omega^2}{g}\left(\frac{3}{4}r_2^2 + \frac{1}{4}r_1^2\right) \tag{4.40}$$

With shallow ponds it is sometimes considered that the g-force is constant, in which case equation (4.28) would be rewritten:

$$v_r = v_s \frac{\omega^2(r_2 - r_1)}{2g} \tag{4.41}$$

Substituting equation (4.30) into (4.41) and integrating:

$$(r_2 - r_1) = v_s \frac{\omega^2(r_2 + r_1)}{2g} t_e \tag{4.42}$$

Eliminating $t_e$from equations (4.42) and (4.27) and rearranging gives:

$$Q_f = \frac{\pi L \omega^2 (r_2 + r_1}{2g} \cdot \frac{(r_2^2 - r_1^2)}{(r_2 - r_1)} v_s \tag{4.43}$$

$$= \pi L g_c^{'}(r_2 - r_1)v_s \tag{4.44}$$

$$= \pi L g_c^{'} D_{AV \cdot v_s} \tag{4.45}$$

where $g_c^{'}$ is the mean g-level in the pond; and D_{AV} is the average pond diameter.

Alternatively equation (4.43) may be written:

$$Q_f = \frac{V}{\Delta r} g_c^{'} \cdot v_s \tag{4.46}$$

whereΔr is the pond depth; and *V* is the pond volume.

For this derivation of clarification capacity, it is readily deduced that:

$$\Sigma = \pi L \frac{\omega^2}{g} \frac{r_2^2 - r_1^2}{r_2 - r_1} \tag{4.47}$$

In the graph of Figure 4.7 the various formulae for Sigma developed so far (equations (4.33), (4.39), (4.40) and (4.47)) are compared for various ratios of r_1/r_2.. The common factor $\pi L \omega^2 / g$ is removed and r_2 is taken as unity, for the graphical comparison.

By means of Figure 4.18, a number of observations may be made. The expansion of the logarithmic term to give an easier formula for Ambler's Sigma is a very acceptable approximation. The even simpler formula last developed above is also acceptable for shallow ponds (radii ratio greater than 0.75). However, there is a significant difference for the formula used for deep ponds. Notice that with zero pond radius, the shallow pond versions of Sigma have finite Sigma values while the deep pond version has a zero value.

This is because a particle starting at the centre line will experience no *g* to initiate its fall, while those which by definition start half way into the pond, or are subjected to a mean *g* throughout the pond will always have a finite settling rate. However for practical designs the radius ratio will always be appreciably over zero, generally in the range 0.4 - 0.8.

It will be seen from the various Sigma formulae that increasing the length of the bowl increases Sigma pro rata. Thus, in this respect Sigma is additive. Some like to include the Sigma value of the conical drying zone in their formula , especially when feeding on the conical drying zone. For this, using equations (4.46) and (4.40):

$$\Sigma = \frac{2\pi\omega^2}{g}\left\{L_c\left(\frac{3}{4}r_2^2 + \frac{1}{4}r_1^2\right) + \frac{L_k}{8}\left(r_2^2 + 3r_1r_2 + 4r_1^2\right)\right\} \quad (4.48)$$

where L_k is the wetted conical drying zone axial length; and L_c is the cylindrical length of the bowl.

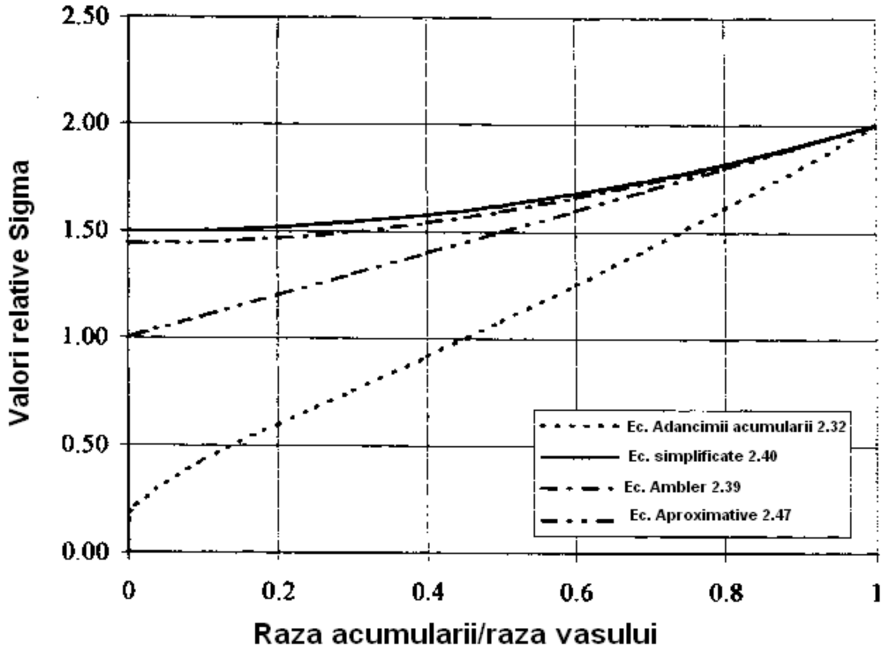

Figure 4.18. Graph comparing the various formulae for Sigma at various pond depths

Considering the assumptions used in the derivation, and the approximations used, one could question whether the use of the simpler equation (4.47) would not suffice, for use with shallower pond machines at least. It is the ratio of Sigma values which is used when scaling from one centrifugal separator size to another. Using equation (4.48), the ratio will be little affected as the extra term will increase by approximately the same ratio with geometrically similar machines.

Another expression, in place of Sigma, uses an empirical formula taking a nominal bowl radius, the f bowl radius, *n* (which equals three quarters of r_2).

This expression [2] is termed the "area equivalent", $Ae_{3/4}$and is determined by:

$$Ae_{\frac{3}{4}} = \frac{2\pi\omega^2}{g}\left(\frac{3}{4}r_2\right)^2\left(L_c + \frac{r_2}{4}\cot\alpha\right) \qquad (4.49)$$

where$a$is the conical drying zone angle (half included).

More often the abbreviated form is used, which ignores conical drying zone volume and uses clarifying length:

$$Ae_{\frac{3}{4}} = \frac{2\pi\omega^2}{g}\left(\frac{3}{4}r_2\right)^2 L \qquad (4.50)$$

$Ae_{3/4}$is then used as a scale-up factor, in place of Sigma. Its use is simply a matter of choice and habit.

All the formulae indicate that a better clarification capacity is achieved at the shallowest pond depth, whereas in practice it is generally the opposite. Therefore, the simple formula is generally considered sufficient for practical purposes. However, when scaling from one machine to another, it is imperative that the same formula is used for both machines. It is also recommended that one should not normally scale between machines of dissimilar geometry.

4.4.2 Using sigma

It is unusual to use any of these formulae to compute the capacity of a single machine. Their most effective and reliable use is in scaling data from one geometrically similar machine to another and assessing relative performances.

Eliminating v_s from equations (4.29) and (4.34):

$$\frac{Q_f}{\Sigma} = \frac{d_c^2(\rho_s - \rho_L)g}{18\eta} \qquad (4.51)$$

or

$$\frac{Q_f}{\Sigma} \propto d_c^2 \qquad (4.52)$$

Taking logarithms of both sides of equation (4.52):

$$\ln\left(\frac{Q_f}{\Sigma}\right) \propto \ln(d_c) \qquad (4.53)$$

It is known from equation (4.28) that the percentage over or under size is a logarithmic probability function of particle diameter. Thus, combining equations (4.30) and (4.53), a logarithmic probability relationship between Q_f/Σ and solids recovery is obtained:

$$\ln\left(\frac{Q_f}{\Sigma}\right) \propto a_1 + a_2 erf^{-1}(2c_w - 1) \qquad (4.54)$$

Plotting Q_f/Σ against solids recovery will thus give a good correlation. Plotting on logarithmic probability paper will produce a straight line. The same straight line is obtained for data from different centrifugal separators, preferably of the same geometry. However, it must be noted that this only applies to process materials with solids exhibiting a skew Gaussian (logarithmic probability) distribution.

When scaling from one machine to another:

$$\frac{Q_{f2}}{Q_{f1}} = \frac{\Sigma_2}{\Sigma_1} \qquad (4.55)$$

where the subscripts 1 and 2 refer to centrifuges 1 and 2, respectively.

When machines of different geometry are used then one needs to take into account the relevant efficiency, £, of each design, when:

$$\frac{Q_{f2}}{Q_{f1}} = \frac{\xi_2}{\xi_1}\frac{\Sigma_2}{\Sigma_1} \qquad (4.56)$$

4.4.3 Sigma enhancement

The use of conical discs, or angled vanes, on the scroll will theoretically enhance the Sigma value, the clarification capacity, of the centrifuge. To estimate the Sigma value of a stack of conical discs, the formula used for disc stack centrifuges may be employed :

$$\Sigma_d = \frac{2\pi n_D}{3} \cdot \frac{\omega^2}{g} \cdot \left(r_3^3 - r_1^3\right) \cdot \cot\theta \qquad (4.57)$$

where $\sum_D$ is the Sigma value for the disc stack; n_D is the number of discs; r_3 is the outside radius of the discs; and θ is the half included angle of the discs.

The total Sigma value for the centrifuge is obtained by adding $\sum_D$ to the Sigma value calculated for the scroll section between the feed zone and the discs.

There is little published on the effect of longitudinal angled vanes, but the equation is derived in a similar fashion to that used for the disc stack centrifuge:

$$\Sigma_V = \frac{n_V L_V \omega^2}{2g \cdot \cot\psi}\left(r_3^2 - r_1^2\right) + \frac{\pi L_V \omega^2}{g} \cdot r_3^2 \qquad (4.58)$$

where $\sum_V$ is the Sigma value for the vanes; L_v is the length of the vanes; n_v is the number of vanes; and ψ is the angle between the vane and a radius.

If the vanes do not extend the full length between the feed zone and the liquids discharge, then the Sigma of the plain section needs to be added to $\sum_V$ to obtain the total Sigma value for the centrifuge.

Caution is needed in using these extended Sigma values, particularly for the angled vanes. This is because flow through the vanes or discs can channel, to take the easiest path. This will reduce the effectiveness of the devices. Good designs, therefore, will endeavour to ensure even distribution of the flow across the vane and disc openings. Even then as liquid flows from the outer edge of vanes or discs, towards the centrifuge axis for discharge at the weir lips, considerable changes in kinetic energy occur. This can cause very complex flow patterns, turbulence and Coriolis effects.

4.4.4 Flocculant requirement

However, it is appropriate at this juncture to mention the need for polymers in some process applications, particularly effluent applications which are a large market for the centrifugal separator. In these applications, without a polymer flocculant, it would not be possible to employ a centrifugal separator. It is clear from equation (4.29), Stokes' law, that the settling velocity of a particle, v_s, is proportional to the square of its diameter. Thus doubling the particle diameter will increase v_s by a factor of four. This results in greater separation efficiency. The objective of flocculants is to change the electrochemical forces on the surface of the particles, so as to bind them together such that they act as one large particle. Once flocculated, these particles must be handled carefully so as not to break them up mechanically. This is especially true when processing them in a centrifugal separator.

In most applications, the amount of polymer used is just sufficient to flocculate a sample of the feed. The amount necessary, as assessed in the laboratory, is generally the amount used in practice on the centrifuge, plus or minus a small fraction. However, recently there has been considerable development in centrifugal separators and their use in obtaining extra-dry solids from compressible sludges, particularly effluents. In these instances, the consumption of polymer has increased considerably.

The amount of flocculant needed increases as the extent of dryness required in the solid increases, and it increases exponentially. The amount of flocculant required also increases with the feed rate to the centrifuge. In practice, on a "dry solids" application, the polymer used will be two to three times that which would be used on a standard application with the same process material.

There has not been a theoretical formula proposed to quantify polymer demand. However, the available data suggest a format similar to equation (4.59):

$$P_D = k_1 + k_2 \cdot e^{(x_s - k_3)} \qquad (4.59)$$

Where k_1, k_2, and k_3, are constants.

Practical data can be very erratic, as it is easy to overdose when striving for extra dryness. When assessing the minimum polymer requirement, it is necessary carefully to adjust all operating parameters, to ensure performance is at the limit, without contingency levels added.

4.5 CLASSIFICATION

Classification, the fractionation or separation of particles by size, could be considered as merely inefficient clarification. The cut, or desired classification, is adjusted by altering the centrifuge's efficiency. This is most easily done by altering the feed rate or bowl speed. However, adjustment of pond depth or differential may, in certain circumstances, be used.

In a thick suspension hindered settling occurs, when there is a tendency for the larger particles, which should settle, to get held up by the dense concentration of the smaller particles. In these circumstances higher differentials could be used, to agitate the suspension and so release the heavier particles. The disadvantage of this is that the solid or "heavy fraction" tends to be wetter, as a result of the higher differential, and thus entrains larger quantities of the smaller particles. To correct this, a shallow pond is selected to allow release of liquid containing the smaller particles on the dry conical drying zone.

In some classification applications, the required cut point is very sharp and the rheology of both separated phases is such that they remain quite fluid. In this type of application the pond used would be relatively deep, and separation would be akin to a liquid/liquid separation, using a hydraulic balance under some form of baffle.

Very occasionally there will be found a classification application where it is required to separate two distinctly different particles, such as in the refining of minerals. In these cases the two different substances to be separated may have markedly different densities. This is particularly acceptable and quite advantageous when the denser material comprises the larger-sized particles. However, if this is not so, one must consider a combination of density and particle size for the cut point of each of the two substances, in relation to Stokes' law. One could visualize the situation of a large, low-density particle settling faster than a high-density, small particle. Thus for such a process to be feasible:

$$d_{ch}^2(\rho_{sh} - \rho_1) > d_{cl}^2(\rho_{sl} - \rho_l) \qquad (4.60)$$

Where d_{ch} is the required cut point size of the heavy fraction; d_{cl} is the required cut point size of the light fraction; ρ_{sh} is the density of the heavy solids; and ρ_{sl} is the density of the light solids.

Each of the two solid constituents will have their own size distribution from which a cut point size can be chosen to give the desired purity of product and yield.

Poor efficiencies can occur in some classification applications, due to natural agglomeration of particles. In these applications, the use of dispersants is quite common. Dispersants have the opposite effect to flocculants, and can be equally powerful.

4.6 THREE-PHASE SEPARATION

Centrifugal separator three-phase operation involves the separation of two immiscible liquids from a solid. The two immiscible liquids are generally oil and water. This could be a waste oil application or the separation of a vegetable oil, such as palm or olive oil.

To put the centrifugal separator into operation two weir heights or equivalent have to be set relative to the solids discharge level, as illustrated in Figure 4.19.

Firstly the weir height, governed by radius $\mathbf{r_1}$ is set to fix the extent of the dry conical drying zone required before solids are discharged at radius $\mathbf{r_s}$.

The radius $\mathbf{r_h}$ has then to be set, to create a hydraulic balance between the two liquid phases, to maintain the equilibrium line at $\mathbf{r_e}$, where required.

The pressure at any radius, **r**, in a rotating centrifuge is given by $\mathbf{P_r}$ where:

$$P_r = \frac{\rho\omega^2}{2g}\left(r^2 - r_1^2\right) \tag{4.61}$$

Thus, in the three-phase centrifuge, the pressure at the equilibrium line is P_e where:

$$P_e = \frac{\rho_l\omega^2}{2g}\left(r_e^2 - r_1^2\right) = \frac{\rho_h\omega^2}{2g}\left(r_e^2 - r_h^2\right) \tag{4.62}$$

Where $\boldsymbol{\rho_l}$ is the density of the light phase; and $\boldsymbol{\rho_h}$is the density of the heavy phase.

The choice of e-line position depends upon a number of factors. The volume of each phase in the bowl could be chosen in proportion to the volumes of each in the feed. Then approximately the value of Q/∑ for each phase would be the same. However, if the separation of one phase from another is relatively more difficult than vice versa, then extra volume could be given to one phase in the bowl to improve its clarification efficiency. Alternatively, the purity of one phase may be more important than the other, and then bias would be given to the more important phase. Nevertheless, care has to be taken in setting the e-line in order not to allow breakthrough of one phase into the other.

When the flow rate of one or more of the phases is high, cresting over the weirs can move the e-line considerably, and adjustment of the weir heights will become necessary. Back-pressure from a centripetal pump or skimmer pipe will require recalculation of weir settings.

Working with three-phase separation will require revision of the formulae used for performance evaluation. Not only will there be interest in solids recovery and clarity (absence of solids), for both liquid phases, there will be an avid interest in what has happened to the oil.

In some of the three-phase applications, water is added to dislodge the oil from the solids. With water addition there are two input streams, feed and water, and three outlet streams, light liquid phase (oil), heavy liquid phase (water) and solid (solids). Each stream is analysed for the three elements, oil, water and solids. Four of the five streams are monitored for flow rate. The solid rate would be difficult to measure accurately.

To analyze performance, a mass balance across the centrifuge is performed for the three separate elements and the total mass, after which the solid rate is eliminated. Formulae are then developed for pertinent recoveries and purities. It is not necessary to develop these here as the pertinent formulae will depend on the application, and in any case the development is similar to that already shown for two-phase separation (see Section 4.3).

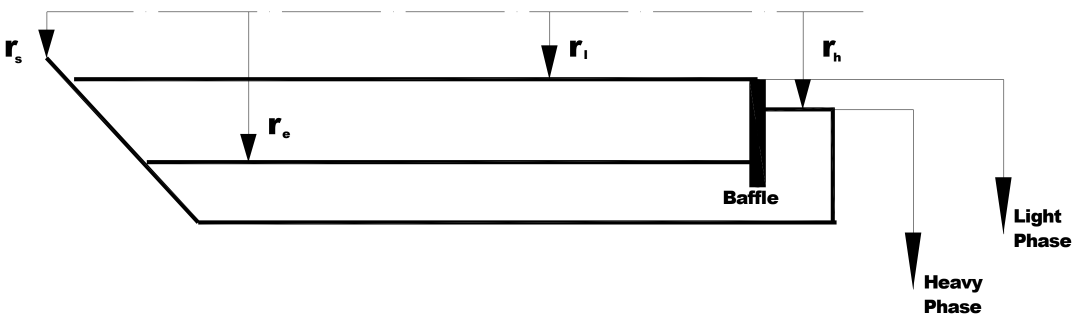

Figure 4.19.Hydraulic balance in three-phase separation.

4.7 FLUID DYNAMICS

By design the centrifugal separator handles very high throughputs relative to the small space it occupies. Moreover, the flow is not simply in one end, and straight through and out the other end. Flow can be under, over and around baffles; it can be a helical path around the scroll flights, or axially through holes in the scroll, or a combination of both.

The axial velocity of the feed into the centrifugal separator has to be converted to a rotational velocity in a very short time. This can cause considerable turbulence, and help is required outside the feed zone to keep the bowl contents up to speed, if not to get it fully to speed. The rotational speed of liquid at the pond surface can slip below that at the bowl wall.

To maintain flow down the bowl and over the weirs, a hydraulic head builds up with a crest.

In this section some of these phenomena will be examined more closely.

4.7.1 Reynolds number

The degree of turbulence in pipes and channels is characterized by the value of the Reynolds number. For a pipe:

$$R_e = \frac{\rho u D_p}{\eta} \qquad (4.63)$$

where D_p is the pipe diameter; and *u* is the velocity in the pipe. In the centrifugal separator with axial flow:

$$u = \frac{Q_l}{\pi(r_2^2 - r_1^2)} \qquad (4.64)$$

In a channel, the pipe diameter, D_p, is substituted by a hydraulic mean diameter, d_m:

$$d_m = \frac{4A}{p} \qquad (4.65)$$

where A is the cross sectional area of the channel; and p is the wetted perimeter of the channel.

Thus, for a non-circular pipe or channel:

$$Re = \frac{\rho u d_m}{\eta} \qquad (4.66)$$

For an annulus:

$$d_m = 2(r_2 - r_1) \qquad (4.67)$$

However the annulus of the pond in a centrifugal separator does not have its inner surface " wetted", and thus the hydraulic mean diameter becomes:

$$d_m = \frac{4\pi(r_2^2 - r_1^2)}{2\pi r_2} = 2\left(r_2 - \frac{r_1^2}{r_2}\right) \qquad (4.68)$$

If the values for d_m from equation (4.68) and velocity from equation (4.64) are introduced into the Reynolds number in equation (4.66), this would imply axial flow. For helical flow:

$$p = P + 2(r_2 - r_1) \qquad (4.69)$$

and:

$$A = P(r_2 - r_1) \tag{4.70}$$

Thus:

$$d_m = \frac{4P(r_2-r_1)}{P+2(r_2-r_1)} \tag{4.71}$$

and:

$$u = \frac{Q_l}{\{P(r_2-r_1)\}} \tag{4.72}$$

Equations (4.71) and (4.72) can be substituted into equation (4.66) to find the Reynolds number for helical flow.

Once the value of the Reynolds number is known, for whatever type of flow is used, the level of turbulence can be assessed. With a Reynolds number below 2000 the flow would be laminar. It will be found that the flow in many, if not most, of all practical cases is in the turbulent regime.

In all centrifugal separators, with solids moving radially out and liquid moving radially in, acceleration and deceleration occur, respectively. Without any mechanical

device to do this, viscous drag of the pond is the only means by which these actions can be accomplished. If the viscosity is low, considerable turbulence can occur, affecting cresting, interface location, stability, and sedimentation and re-entrainment of settled solids.

4.7.2 Moving layer

In a pond of an operating centrifugal separator centrifuge, there often tend to be two, distinct liquid layers. The upper or surface layer, the moving layer, moves rapidly and turbulently towards the discharge weirs. Under this moving layer, the pond is quiescent, allowing solids to settle under a laminar flow regime, and then to compact. This is a simplistic picture, as the shape of the scrolland its movement adds to the complexity.

It is sometimes useful to estimate the depth of the moving layer to know when it is liable to disturb and re-entrain sedimented solids.

It will be appreciated that the thickness of the moving layer will depend upon whether the flow is axial or helical. Research has shown that for axial flow:

$$h_m \propto \frac{\sqrt{Q_l}}{\sqrt[3]{g_1}} \tag{4.73}$$

where h_m is the thickness of the moving layer.

It will be seen that the formula is independent of path length, the clarifying length. It has also been found that moving layer thickness closely follows cresting height (see Section 4.21.3). Thus, the shape, size and number of weir plates used can affect the moving layer thickness. The moving layer thicknesses found in helical flow are greater than those calculated using equation (4. 73).

4.7.3 Cresting

The level difference between liquids and solid discharges can be quite critical when optimising process performance. When the level difference is small, the degree and consistency of the liquids cresting can play an important part in the process performance optimising.

The crest height, the pond surface level above the actual ^weir height, is a function of the liquids rate, the total weir width and centrifuge g-level, as well as physical constants of the liquid such as viscosity and density. Crest height is h_c, given by:

$$h_c = \frac{1}{\sqrt[5]{2r_1}}\left(\frac{Q_l}{c_0 \omega B}\right)^{\frac{2}{5}} \qquad (4.74)$$

where c_0 is a constant and generally approximately 0.415; and B is the total length of weirs.

This equation is derived from the Francis formula .

Experimental data show that, due to the interrupted nature of the discharge weir, the calculated value of crest height needs to be increased by 35% for axial flow and 90% for helical flow. With a 360° internal weir, B would be the full circumference, and thus would cause the least cresting.

4.7.4 Feed zone acceleration

Feed zones are designed to accept the maximum possible feed rate, and bring it up to bowl speed with the minimum of splashing and rejection. Bringing the feed up to the angular velocity of the bowl is not necessarily enough. As the process material flows out of the feed zone to the pond, it has a constant linear velocity fixed by its angular velocity at its point of exit from the feed zone. To maintain its angular velocity extra linear velocity is required as the radius increases [21].

The power required to bring the feed material up to bowl speed at the pond surface is P_P, where:

$$P_p = Q_f \rho_f \omega^2 r_1^2 \qquad (4.75)$$

Power available in the feed stream at the pond surface is P_A, where:

$$P_A = \frac{1}{2} Q_f \rho_f \omega^2 r_1^2 \qquad (4.76)$$

The power lost on entry is thus the difference between equations (4.71) and (4.72), and this is dissipated in heat and turbulence on entry. Thus to minimize turbulence and power loss, it is necessary to design the centrifugal separator with the pond surface as close as practicably possible to the centre line. Nevertheless, other process considerations may require the taking of a different view.

4.8 POWER CONSUMPTION

The total power input required by a centrifugal separator centrifuge comprises a number of separate power components:

$$P_T = P_P + P_{WF} + P_S + P_B \qquad (4.77)$$

where P_T is the total power required by the centrifugal separator; P_P is the power required to accelerate the process material to the bowl speed at the discharge radius; P_{WF} is the power to overcome windage and friction; P_s is the power required for conveying; and P_B is the power for braking. From equation (4.75):

$$P_p = Q_f \rho_f \omega^2 r_d^2 \qquad (4.78)$$

where r_d is the process discharge radius.

Naturally, if solid and liquids are discharged at different radii, then these two power components have to be calculated separately. The windage and friction component is given by:

$$P_{WF} = k_7 + k_8\omega + k_9\omega^2 \qquad (4.79)$$

where k_7, $k_8$and Jc_9 are constants. P_{WF} can be calculated with difficulty but is more generally derived practically in the factory by measuring the power absorbed for different bowl speeds. The conveying component is given by:

$$P_S = NT \qquad (4.80)$$

where N is the scroll differential; and T is the scroll torque. Similarly, the braking component is:

$$P_B = S_P T_P \qquad (4.81)$$

where S_P and T_P are the pinion speed and torque, respectively.

In some types of back drive the braking power can be regenerated, so that the total power used is reduced.

4.8.1 Main motor sizing

The power of the main motor will be based on the calculation of P_T from equation (4.77), while its physical size will be influenced by its starting requirements. Motor manufacturers rate their motors on the basis of the maximum power delivered at the motor shaft, P_M- This has to be greater than P_T to cater for frictional losses in the drive belts and fluid coupling, if used. Thus, the motor power is P_M:

$$P_M \vartheta_D \vartheta_B = P_T \qquad (4.82)$$

where ϑ_D is the fluid drive efficiency; and ϑ_B is the efficiency of the drive belts.

The power used, however, will be greater than P_M, due to losses in the motor itself and losses in some control gear when used, such as an inverter.

Power is lost within a motor due to a number of factors, which include:

* iron losses in the magnetizing material, producing heat in the motor rotor and stator;
* friction in the rotor bearings;
* energy needed to drive a cooling fan, internally attached to the motor shaft;

* windage losses; and
* copper losses (the power lost due to the resistance of the windings.

These five factors combine to give a motor efficiency, ϑ_M, of less than unity.

Extra power is also necessarily supplied to the motor, when the power factor is less than unity. The power factor will never be unity, and is a measure of how much the current lags or leads the applied voltage. It is measured as the cosine of the phase angle between current and voltage. When an induction motor is connected to an AC electrical supply, whether the motor does useful work or not, a current is drawn to excite the motor. This current, instantaneously on start-up, lags 90° out of phase with the voltage, and is reactive current, or so-called idle or wattles current. The power factor increases as the motor accelerates.

When the motor is put to work, it will take in addition to its excitation current, a current according to the amount of work to be done. The power factor will increase and will be maximum when the motor works at its full power rating. Thus the power taken from the mains supply will be P_c where:

$$F_P P_C \vartheta_M = P_M \qquad (4.83)$$

where F_P is the power factor.

To combat the anomaly of a low power factor, the installation of a capacitance bank, ideally directly across the motor windings, causes the motor current to reach its maximum value closer to when the voltage does in the alternating cycle. Therefore, a suitably designed capacitor added to an induction motor will reduce the lag of current, by any desired amount. Generally, in industry, because the cost of small capacitors is high, it is more economical and expedient to install large banks of capacitors at the supply source, and automatically switch in and out various sets of capacitors as the demand fluctuates. Moreover, a leading current, which is possible if the capacitor is too large, increases wattles current as much as a lagging current.

Motor manufacturers supply motors in standard increments of power. Thus, after power demand for the centrifugal separator is calculated, the next larger size is specified. Motor manufacturers can supply tables of efficiency and power factors for ranges of loading. Also available are performance curves for their motors, giving output torque against rotational speed. The selected size of motor, for economic reasons, needs to be as near as possible to the power demanded by the centrifuge.

Details of the installation need to be considered in the motor specification. These factors would include the ambient temperature, whether the installation is indoors or out, and whether any hazards exist, such as flammable materials in use, and whether the motor will need to be hosed down. The installed electrical services need to be assessed to ensure that they are adequate for the method of starting contemplated. It is important that supply cables be adequately sized to minimize voltage drops before reaching the main motor. The power supplied to the motor reduces proportionally to the square of any voltage drop. Nevertheless current will increase to compensate for the drop in voltage, increasing the heating and losses. Moreover regulations restrict voltage drops to a total of 4%.

If reduced voltage starting is used, it is important that the reduced starting torque is never less than the sum of the friction and windage torque. Since the torque available to accelerate the bowl is equal to the difference between the motor torque and friction and windage torque, the motor may not reach full speed in a reasonable time, unless care is taken.

4.8.2 Main motor acceleration

Most centrifugal separator rotating assemblies have high inertias, which can require several minutes' acceleration , or run-up time, t_a. If the run-up time is too short, drive belts will slip and wear out prematurely, or even break. If the run-up time is too long, then the motor could overheat and burn out. The run-up time is:

$$t_a = \frac{\omega_M(J_M - J_P)}{(T_a - T_l)} \qquad (4.84)$$

where J_M is the inertia of the motor; J_P is the inertia of the centrifugal separator at the motor; ω_M is the motor speed; T_a is the motor torque; and T_l is the reactive torque of the centrifugal separator.

The centrifugal separator inertia is given by:

$$J_P = \frac{\omega^2}{\omega_M^2} J_D \qquad (4.85)$$

where J_D is the inertia of the rotating assembly

Both T_a and T_l vary with speed, and not linearly. Examples of motor and centrifugal separator torque/speed curves are shown in Figure 4.20. To use equation (4.84), T_a and T_l are averaged over the speed range from zero to full motor speed.

Given the inertia at the motor shaft, the equations in this section are used to determine whether the torque of the chosen motor is sufficient to accelerate the centrifugal separator bowl smoothly to speed, without slippage of the belts. The thermal limits and the torque limit of the number of drive belts used and their cross-sections have to be checked, with the pre-set diameter of the smallest pulley taken into account. Causing the belts to slip will end in their failure, while producing copious amounts of dust in the belt guard, which could be an explosion hazard.

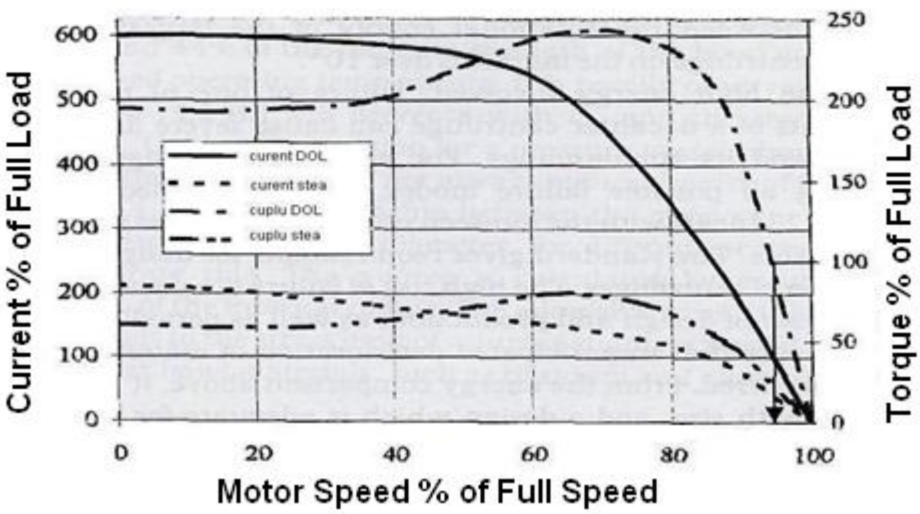

Figure 4.20. Motor and centrifugal separator torque / speed curves

4.9 MECHANICAL DESIGN

The design of a good and reliable centrifugal separator centrifuge requires a thorough knowledge of most mechanical engineering disciplines such as machine dynamics, strength of materials, bearing design, and gearbox design. Some of the more important fundamental aspects of the mechanical design of centrifugal separator centrifuges will now be discussed.

The need for a careful mechanical design can be illustrated by examining the energy accumulated in a centrifugal separator centrifuge in operation. The rotational energy in a medium size centrifugal separator with rotational inertia of 50 kg m^2, rotating at 3600 rpm, will be 3.55 MJ. This energy corresponds to the kinetic energy of a vehicle weighing 9.2 tons travelling at 100 km/h. Furthermore, it can be shown that the rotational energy of a centrifugal separator centrifuge will increase with the fourth power of the diameter, when the centrifugal force at the bowl wall, g_c, and the length/diameter ratio, λ, are kept constant. The ratio of the diameters of the largest to the smallest industrial bowl is over 10. Thus, the ratio between the rotational energy of the largest and the smallest centrifugal separator centrifuge on the market is over 10^4.

With the high energy involved, failure of one of the major rotating components of a centrifugal separator centrifuge can cause severe damage, both to the centrifugal separator and its surroundings. For all centrifugal separator designs a risk analysis, evaluating all possible failure modes, must be carried out. A European standard deals with the foreseen risks for centrifuges in normal operating environments. The standard gives requirements for design, verification, and installation of centrifuges. The high risk of failure of a centrifugal separator requires a high quality, both of design and production, as well as periodic inspection during use, to ensure that unanticipated deterioration of materials of construction has not occurred. From the energy comparison above, it is seen that the risk increases with size, and a design which is adequate for a small laboratory-scale centrifugal separator, may be extremely dangerous on a large industrial scale.

4.9.1 Maximum bowl speed

The bowl shell of a centrifugal separator centrifuge will be subjected to both the pressure loads from the material inside the centrifugal separator and the centrifugal load acting on the material of the bowl shell. The cylindrical part of the bowl shell will be, for normal centrifugal separator designs, the part of the bowl subjected to the highest stress levels. The maximum pressure on the bowl shell is calculated using equation (4.61):

$$P_{2m} = \frac{\rho_M \omega^2}{2}(r_2^2 - r_1^2) \qquad (4.86)$$

where P_{2m} is the maximum pressure at the bowl wall; and ρ_M is the maximum bulk density of process material ever likely in the centrifugal separator.

Defining t_w as the wall thickness of the centrifugal separator bowl shell and ρ_B as the density of the bowl material, the average tangential stress in the bowl shell can be expressed, for a straight cylinder, as:

$$\sigma_t = \frac{r_2}{t_w} P_2 + \omega^2 \rho_b \left(r_2 + \frac{t_w}{2}\right)^2 \qquad (4.87)$$

where P_2 is the actual pressure at the bowl wall; and $\boldsymbol{\sigma_t}$ is the mean tangential stress in the bowl wall.

The formula is equivalent to the well-known pressure vessel formulae for thin cylindrical shells. To ensure safety against failure, the tangential stress must be below a certain allowable stress. According to the European Engineering Directive, the tangential stress shall be kept below 66% of the yield strength and 44% of the ultimate strength of the bowl material, at the maximum allowed operating temperature. It is readily observed that the first term of equation (4.87) will decrease with t_w, and the second term will increase with t_w. Unlike the situation for a pressure vessel, simply increasing the thickness of the bowl shell will not always reduce the risk of failure. On the graph of Figure 4.21, the relationship between the maximum obtainable g-force, and the centrifugal separator internal diameter, for different process densities is shown to illustrate this. The g-force is calculated by assuming a bowl thickness of 10% of the internal radius. The allowable stress is set to 240 MPa, which corresponds to the stress limit of a duplex stainless steel at 100°C.

Of course, other bowl materials, such as titanium and aluminium, will give different values.

The pressure inside the bowl will also create an axial force, acting on the end hubs. The maximum axial force, F_x, is found by taking the mean pressure in the pond as half the pressure calculated from equation (4.87), and multiplying by the cross-sectional area of the pond. Thus:

$$F_x = \frac{\pi}{4} \omega^2 \rho_M (r_2^2 - r_1^2)^2 \qquad (4.88)$$

On a large centrifugal separator, the axial force on the end hubs will be greater than 10^6 N. The end hubs, and axial fixings, on on the rotor must therefore be designed to withstand this force with a sufficient safety margin. Components of the centrifugal separator may be subjected to several other design-dependent static loads, which must be considered by the designer. One example is the axial load, acting on the scroll, caused by conveying the solids.

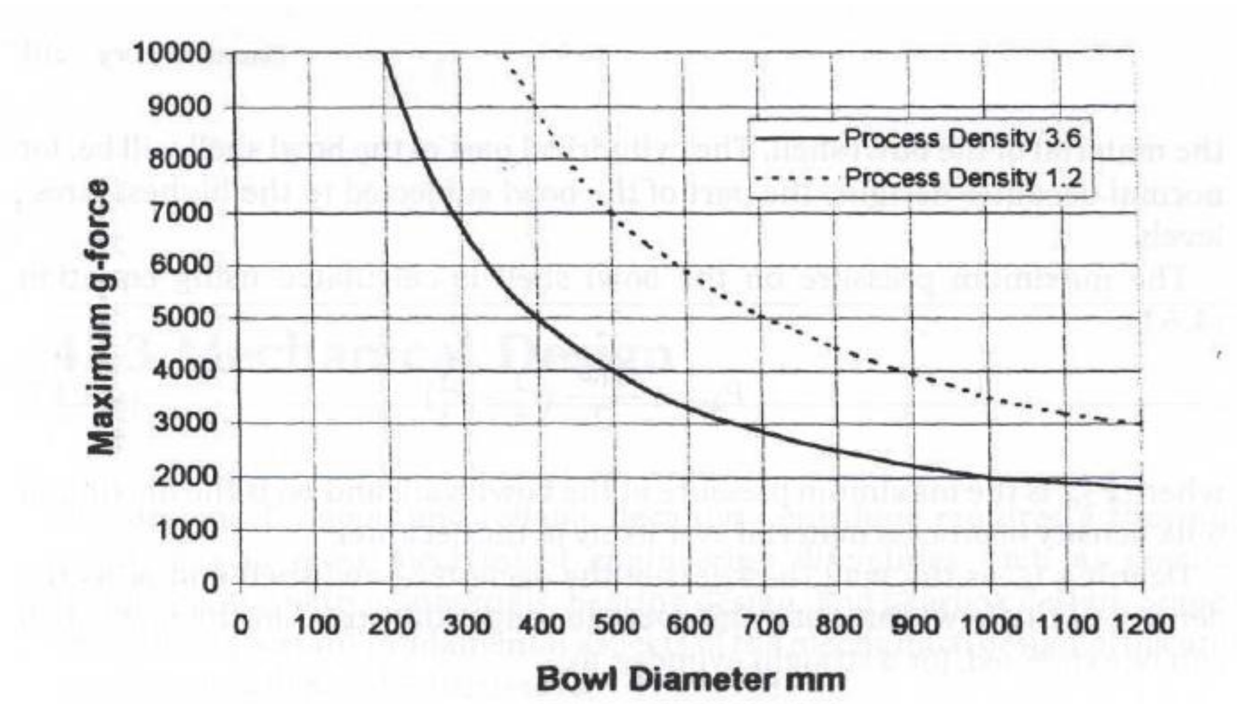

Figure 4.21 Example of the relationship between bowl radius, max g-force and solid density for one material and one relative bowl shell thickness

The centrifugal separator will also be subjected to cyclic loads, which can cause mechanical fatigue damage on both the rotating assembly and on the stationary parts. Among the cyclic loads which must be considered by the designer are the bending forces on the shafts, caused by the weight of the rotor, the loads from belt drives, unbalance forces from the rotor, and cyclic loads from frequent starts and stops, or intermittent loading with process material. On complicated hub geometries, often it will be necessary to make a finite element calculation of the stresses to make a proper fatigue evaluation. The notch sensitivity and ductility of the material must be considered.

Quality assurance procedures during manufacturing, such as X-raying of critical welds and die penetrant testing of castings, must be maintained.

4.9.2 Critical speeds

The natural frequencies and critical speeds of a centrifugal separator will depend on its actual configuration. A conventional centrifugal separator centrifuge consists of a frame holding the double rotor — scroll and bowl — in rigid bearings. The main motor can be attached to the rotor frame either by a rigid connection or flexibly through vibration isolators. Further the motor can be attached to a sub-frame or to another part of the supporting structure.

At speeds below operating speed the main frame and the rotor can be considered as one rigid body. If the centrifugal separator frame is mounted on soft vibration isolators the centrifugal separator assembly will have six natural frequencies and associated vibration modes below the operating speed. The natural frequencies are determined by the spring stiffness of the vibration isolators and the mass and inertia of the total system. When the main motor also is supported on the centrifugal separator frame by vibration isolators, it will have six additional natural frequencies below the operating speed.

The important critical speed for a centrifugal separator is the lowest speed at which there is significant flexible deformation of the rotor. This speed is called the first rotor critical speed. Centrifugal separators will always have a certain unbalance, both due to the handling of solids from the process and due to wear on the rotor. Operating the centrifugal separator close to, or just above, the flexible critical speed of the rotor will result in high vibration levels and very high stresses in the rotor components. The critical value of the rotor speed will therefore be an upper limit for the operating speed, and the centrifugal separator must be operated below this speed with a safe margin.

The first rotor critical speed will mainly be a function of scroll geometry, bowl geometry, gearbox weight, main bearing stiffness and scroll bearing stiffness. The first rotor critical speed will decrease with the length of the centrifugal separator. The critical speed of a centrifugal separator can be calculated by using a finite element method and verified by measurements. It is normal practice to test centrifugal separators at a speed 15-20% above operating speed, to verify the design integrity, and such an over-speed test can also reveal if the operating speed is close to a critical speed. These factors influencing the first rotor critical speed have been more fully covered by Madsen.

4.9.3 Liquid instability problems

Often very large vertical and horizontal vibrations are seen in some speed intervals on centrifugal separators when they are started and stopped with liquid inside the bowl. The vibration frequencies in the instability intervals correspond to rigid body natural frequencies of the centrifugal separator, but the vibrations are not caused by unbalances. Rather, they are due to interaction between the liquid inside the bowl and the centrifugal separator.

The vibrations which occur in some instability speed ranges are sub-synchronous, i.e. the vibration frequency is a fraction (normally about 0.7) of the actual operating speed. If, for example, the centrifugal separator is vibrating at an operating speed of 1000 rpm, the vibration frequency will be around 700 rpm.

The vibrations are usually harmless, but as very large forces could be acting on the foundations of the centrifugal separator, the manufacturer must supply information on the magnitude of these forces, and the foundation must be designed to withstand these forces. By having a constant flow of water to the centrifugal separator during starting and stopping, the instability vibrations can be suppressed.

The complicated dynamic phenomenon, which is related to all rotating cylinders with an internal annulus of liquid, has been dealt with in a number of publications.

4.9.4 Length/diameter ratio

In general the bowl strength, the first rotor critical speed, and the maximum permissible speed of the main bearings control the maximum speed at which a' centrifugal separator can be operated.

It is argued that a long, slender centrifugal separator centrifuge will give advantages with respect to overall economy, power consumption and process performance. For a centrifuge with the *L/D* ratio above 3, the critical speed will often be the main factor controlling the maximum obtainable speed and it can therefore be desirable to increase the critical speed in order to obtain a high *L/D* ratio without sacrificing the maximum operating speed.

When a centrifugal separator bowl, for calculation purposes, is approximated to a beam, its natural frequency is inversely proportional to the square of its length. In that g-force is proportional to the square of bowl speed, and it is necessary to keep resonance frequency above bowl speed, the maximum bowl speed is proportional to its length to the fourth power. To obtain g-forces in the range 2000-3000, generally required for commercial centrifugal separators, the maximum length-to-diameter ratio, for the most frequently used designs, has to be restricted to a little over 4.0.

In order to increase the critical speed of the rotor a number of different modifications can be made to the rotor system. By supporting one or both main bearings in a flexible pillow block the first critical speed of the rotor can be turned into a low speed rigid-body motion for the rotor. It can then be operated super critically with respect to this critical speed. Other modifications are the floating scroll and the separately supported gearbox.

These sorts of modifications have been utilized by Alfa Laval in producing a centrifugal separator with an *L/D* of over 5 which can operate with up to 10 000 g.

How these modifications extend the possible *L/D* ratio and clarification capacity was graphed by Madsen, for 250 mm diameter bowls, and reproduced in Figure 4.22.

4.9.5 Bearing life

One of the most frequent reasons for breakdown of centrifugal separators is failure of one of the main bearings. The operating conditions of centrifugal separators are often very arduous, and there can be a high load on the bearings. The failure of a main bearing on a properly designed centrifugal separator will not lead to a dangerous situation, but it can cause damage to other parts of the centrifugal separator, and expensive downtime.

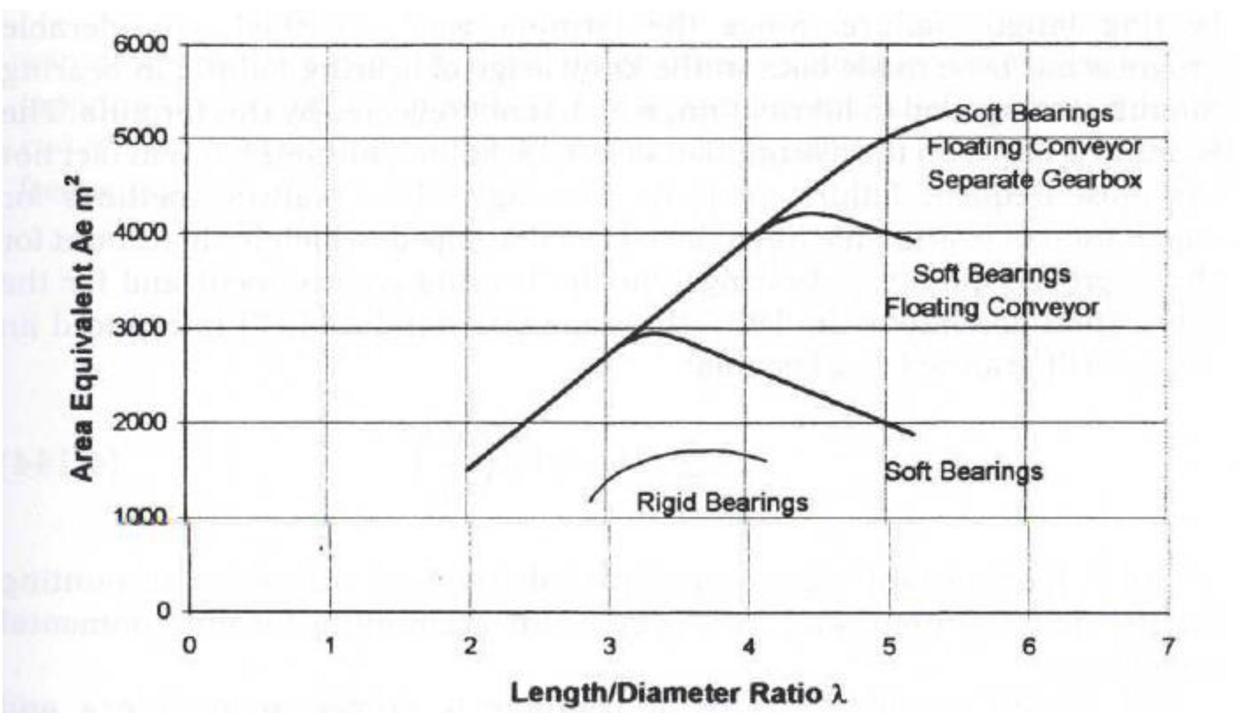

Figure 4.22. Maximum area equivalents &L/Dfor various 250 mm diameter bowl

The bearing life is defined as the number of revolutions or number of hours at constant speed a bearing will operate before it fails. According to the international standard, and based on the assumption that the bearing will fail by fatigue, the expected life of a bearing is calculated by the simple formula:

$$L_{10} = (\frac{C}{C_E})^w \quad (4.89)$$

where L_{10} is the expected life measured in 10^6 revolutions; C is the dynamic load capacity of the bearing, a characteristic figure for the bearing, determined by the manufacturer in accordance with the ISO standard; C_E is the equivalent dynamic load, calculated from the dynamic and static loads; and *w* is a number depending upon the bearing type (e.g. for ball bearings w = 3 and for roller bearings w = 10/3).

Both C and C_E are expressed in a unit of force. The L_{10} life is also sometimes referred to as the B_{10} life.

For a machine rotating at a constant speed, *n,* in revolutions per minute, the expected life can be expressed in expected hours of operation, L_{10h}:

$$L_{10h} = \frac{10^6}{60n}(\frac{C}{C_E})^w \quad (4.90)$$

where L_{10h} is the expected life in hours; and *n* is the number of revolutions per minute.

This simple formula was developed around 1950 and was based on data for bearing fatigue failure. Since the formula was published, considerable progress has been made both in the knowledge of bearing failure, in bearing manufacturing, and in lubrication, which is not reflected by this formula. The formula is based on the assumption of fatigue failure, although it is in fact not the most frequent failure mode for bearings. More realistic methods for calculation of bearing life have since been developed, which both account for the improved quality of bearings, for the bearing environment and for the ' lubrication conditions. In 1977 the same ISO standard introduced an adjusted life-rating L_{10ah} formula:

$$L_{10ah} = b_1 b_2 b_3(\frac{C}{C_E})^w \quad (4.91)$$

Where b_1 is a constant accounting for reliability; b_2 is a constant accounting for the material used; and b_3 is a constant accounting for environmental conditions.

The key to avoiding failure of bearings is proper maintenance and lubrication. By monitoring and analysing vibrations measured with sensors directly on the bearing housings of a rotating machine, bearing faults can often be detected before they lead to failure. Several systems for detection of bearing faults, by continuous vibration monitoring, are available, and some centrifugal separator manufacturers offer their own specialised systems. For critical installations, and installations with several centrifugal separators, such monitoring systems can be a good investment, to avoid inconvenient bearing failures, damage to the machine, and unnecessary downtime.

4.9.6 Gearbox life

The centrifugal separator manufacturer will often quote the expected life of the gearbox. This will be based on the fatigue life of the gear teeth, which is proportional to the ninth power of the torque encountered. Thus, one has to be extremely careful not to overload the gearbox above its torque rating. An 8% increase of torque over its rating will halve the expected life of the gearbox.

4.9.7 Feed tube

Each component of the centrifugal separator has its own natural frequency, even the stationary components, which could resonate sympathetically if this frequency is close to the bowl speed. The feed tube is a good example, being a long, thin tube. Apart from the inverse square relationship with length, resonance frequency is also proportional to the fourth power of diameter in its simplest form. Unless care is taken the feed tube can be caused to resonate like a tuning fork.

The design engineer thus endeavors to maximize diameter and minimize the length of the feed tube. Other techniques employed include tapering the feed tube and making it of lighter materials. Of course the double concentric tube used, when flocculant is added, helps to increase the natural frequency.

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4.10 SIMULATION OF FLUID FLOW IN CENTRIFUGAL TRICANTER

Nomenclature

D_I	kinematic diffusivity [*m*^{2}/*s*]
h	specific thermodynamic enthalpy [*J/kg*]
k	specific turbulent kinetic energy per unit mass [*m*^{2}/*s*^{2}]
w	molecular weight [*kg/kmol*]
p	pressure [*N/m*^{2}]
R	universal gas constant [*J/kgK*]
S_I	is the source term due to chemical reaction rate involving component I
T	temperature [*K*]
u	velocity [*m*]
P_k	turbulence production due to viscous forces
t	time [*s*]
R_K	elementary reaction rate of progress for reaction k
W_I	molar mass [*kg/kmol*]
Y	mass fraction [-]
C_{S1}	model constant [-]
C_{S2}	model constant [-]

Greekcharacters

δ	Kronecker delta [-]
ε	dissipation rate of the turbulent kinetic energy per unit mass [*m*^{2}/*s*^{3}]
μ	dynamic viscosity [*kg*/(*ms*)]
ν	kinematic viscosity [*m*^{2}/*s*]
ρ	density [*kg/m*^{3}]
ν_{kI}	stoichiometric coefficient for component *I*
τ_{ij}	turbulent stress tensor [*m*^{2}/*s*^{2}]
σ_k	model constant [-]
σ_ε	model constant [-]
Γ_i	diffusion coefficient of component I [m^2/s]

Dimensionlessnumbers

Pr Prandtl number

Statistical quantities

$\bar{\phi}$ time or ensemble average of variable ϕ

ϕ' fluctuating part of variable ϕ

$\tilde{\phi}$ filtered variable ϕ

4.10.1 Introduction

Development of high speed computers has a significant impact on how the principles in fluid mechanics and heat transfer are applied in modern engineering design process. Design issues pertaining to this area can be solved in a short time with current computers, problems that have required years of calculation methods and computing power twenty years ago.

From the progress of methods and computing power have benefited primarily research laboratories and specialized industry where solving the problem in shortest time was a priority. Also the implementation of specialized schools helped form specialists who know how to use new methods before graduation.

In recent years we have seen the development of new methods to solve complex problems in fluid mechanics and heat transfer. This new method has been called CFD (Computational Fluid Dynamics). It is characterized by the fact that the governing flow equations, usually written as partial derivatives are solved numerically.

This method was developed in the past as a third method of design of equipment besides the theoretical and experimental methods, in terms of fluid mechanics and heat transfer.

Although it is very costly the experimental method is still very important especially in complex flows, but the focus begins to be increasingly more on CFD, where solutions are validated using experimental methods. The use of CFD in design a can be explained in two different aspects:

1. Economic aspect (Chapman 1979) - As the years past computing speed increased much more than the cost of computing. As shown in Figure 1 for a specific problem of computing costs have fallen 10 times in the course of eight years.

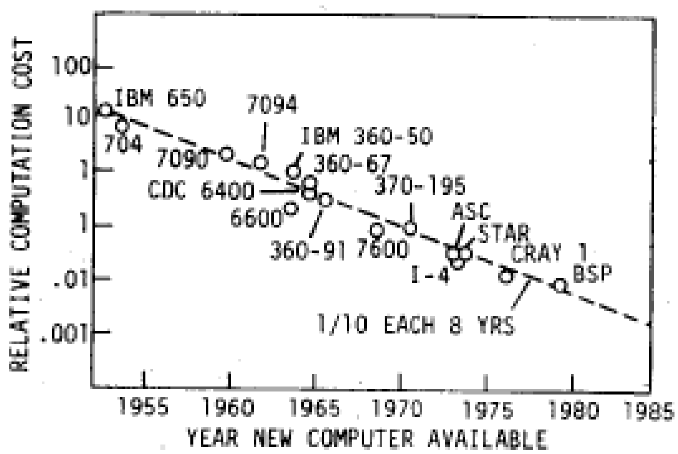


Fig 1 The evolution of the costs account for a certain flow and algorithm (Chapman 1979)

An example is that described by Chapman in 1979 he found that numerical simulation of flow over an airplane wing using Navier Stokes equations, Reynolds averaged, can provide a solution in less than half an hour with about $ 1,000 costs today.

If this calculation had been tried 20 years ago, with the computers (e.g. IBM 704-class) and algorithms known at the time, computing costs would have been approximately $ 10 million and results for this flow were hardly available in about 30 years.

2. Aspect of computing speed - as computers grow their computing speed increases which leads to problem solving in shorter and shorter time.

Method	Advantages	Disadvantages
Experimental	1.The most realistic	1. Specialized equipment 2. Scaling problems 3. Test bench corrections 4. Measuring difficulties 5. Operating costs
Theoretical	1.exact, general information	1.restricted to simple phsical Problems and simple geometries 2. restricted to linear problems
Numerical	1. No restriciton concearnig liniarity	1. Truncation errors

2. Complex roblems can be studied 2. Problems with boundary conditions
3. Time dependent solution of the flow

Table 4.1

As you can see from the table 4.1 and the most viable design and calculation methods remain the experimental and the numerical method. But between them remain some differences that deserve to be mentioned. The experimental method is usually used on smaller scale models due to high costs.

Another feature is the inability to accurately simulate operating conditions of various equipment and data sampling areas and obtain certain results. The main experimental method has several constraints which numerical method does not have but on the other hand this one has other drawbacks like limited data storage space and computing speed.

And another thing worth mentioning to the numerical method is that misunderstanding of certain phenomena and their impossible mathematical transcription. But of all these disadvantages of the numerical method is not insurmountable.

Progress of CFD in the past 50 years is impressive, therefore experimental method in recent years began to play a secondary role, namely to validate the numerical methods in aerodynamic problems.

In this paper we present a fluid dynamics study of three-phase separation applicable to centrifugal separator (tricanter) designed.

Due to the complex geometry flow analysis will be performed in 3D.

4.10.2. Numerical Method

For this study, the flow was assumed compressible, the ecuations that govern the flow, written in Reynolds averaged form, time and mass averaged [4] , being, in the repeated indices summation convention:

The Continuity Equation**:**

$$\frac{\partial \overline{\rho}}{\partial t} + \frac{\partial \overline{\rho} \widetilde{u}_j}{\partial x_j} = 0 \qquad (1)$$

The Momentum Equations:

$$\frac{\partial \overline{\rho}\tilde{u}_i}{\partial t}+\frac{\partial \overline{\rho}(\tilde{u}_i\tilde{u}_j)}{\partial x_j}=-\frac{\partial \overline{p}}{\partial x_i}+\frac{\partial}{\partial x_j}\left(\overline{\tau_{ij}}-\overline{\rho u_i' u_j'}\right) \tag{2}$$

, where

$$\overline{\tau_{ij}}=\mu\left[\frac{\partial \tilde{u}_i}{\partial x_j}+\frac{\partial \tilde{u}_j}{\partial x_i}-\frac{2}{3}\delta_{ij}\left(\frac{\partial \tilde{u}_k}{\partial x_k}\right)\right]$$ represent the stress tensor.

The Total Energy Equation:

$$\frac{\partial}{\partial t}\left(\overline{\rho}\tilde{h}\right)+\frac{\partial\left(\overline{\rho}\tilde{u}_j\tilde{h}\right)}{\partial x_j}=\frac{\partial \overline{p}}{\partial t}+\frac{\partial}{\partial x_j}\left(\overline{\frac{\mu}{\Pr}\frac{\partial h}{\partial x_j}}\right)+\frac{\partial}{\partial x_j}\left(-\overline{\rho h' u_j'}\right), \tag{3}$$

where h is the enthalpy.

Ideal Gas Equation of State:

$$\tilde{\rho}=\frac{w\left(\tilde{p}+p_{ref}\right)}{R_0\tilde{T}} \tag{6}$$

, where w is the molecular weight

To close the correlation type terms that appear in the above equations, the k-ε two-equation turbulence model is employed. The model uses the gradient diffusion hypothesis to relate the Reynolds stresses to the mean velocity gradients and the turbulent viscosity. The turbulent viscosity is modeled as the product of a turbulent velocity and turbulent length scale.

In the two-equation class models, the turbulence velocity scale is computed from the turbulent kinetic energy, which is provided by numerically solving its transport equation along with the governing equations presented earlier. The turbulent length scale is estimated from two properties of the turbulence field, in this case the turbulent kinetic energy, k, and its dissipation rate, ε. The dissipation rate of the turbulent kinetic energy is also provided by numerically solving its transport equation.

$$\frac{\partial(\rho k)}{\partial t}+\nabla(\rho U k)=\nabla\left[\left(\mu+\frac{\mu_t}{\sigma_k}\right)\nabla k\right]+P_k-\rho\varepsilon \tag{7}$$

$$\frac{\partial(\rho\varepsilon)}{\partial t}+\nabla(\rho U\varepsilon)=\nabla\left[\left(\mu+\frac{\mu_t}{\sigma_\varepsilon}\right)\nabla\varepsilon\right]+\frac{\varepsilon}{k}\left(C_{S_1}P_k-C_{S_2}\rho\varepsilon\right) \qquad (8)$$

Where C_{S1} , C_{S2} , σ_k , σ_ε are model constants, and $P_k$is the turbulence production due to viscous forces which is modeled using the following formula [3]:

$$P_k=\mu_t\nabla U\left(\nabla U+\nabla U^T\right)-\frac{2}{3}\nabla U\left(3\mu_t\nabla U+\rho k\right) \qquad (9)$$

These equations are discretized using a second order upwind scheme.

4.10.3. Setup and Boundary conditions of the numerical simulations

Aerodynamic computational model implies certain modifications from the real model. Basically what is kept is the path of the working fluid inside our tricanter centrifuge.(Fig. 2)

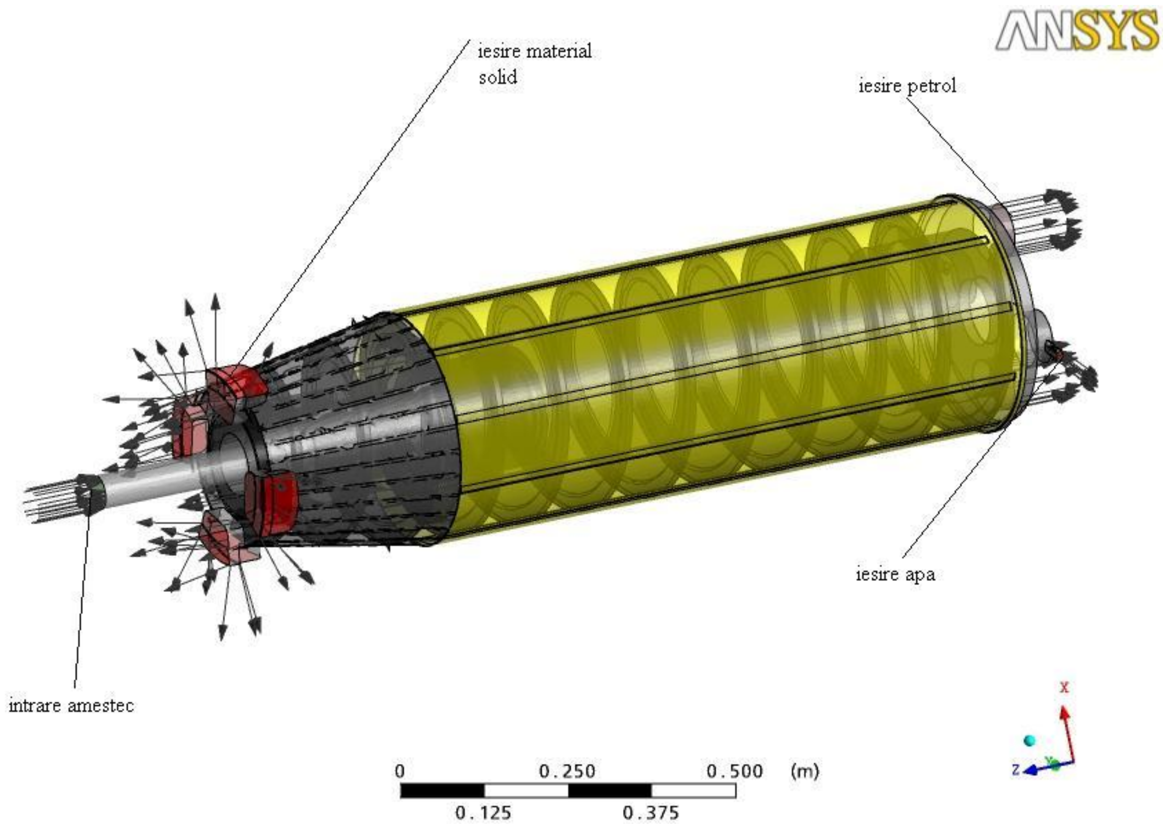


Fig.2 Computational domain

Initial conditions are the following:
Mass flow: 20 m^3/h
Inlet:
mixture: water + oil in liqiud form
Water density: 1000 kg/m^3
Oil density: 800 kg/m^3
Temperature : 293 K
Total pressure: 5 bar
Turbulence intensity: 5 %
Oil volume fraction: 0.7116
Water volume fraction: 0.2884

Solid particles:
Density: 2011 kg/m^3
Inlet speed : 10 m/s
Mass flow : 68 g/s
Particle minimum diameter: 20 de microni
Particle maximum diameter: 60 de microni

Outlet:
Mass flow : 20 m^3/h

Casing:
Speed: 4000 rot/min
Centrifugal rotor:
Speed: 3960 rot/min

The solid walls were considered adiabatic (no heat transfer), impermeable and no-slip(zero velocity at the wall).

For this case an unstructured grid had been used because the complexity of the tricanter makes the problem too computationally expensive for a structured grid. Also, in order to be able to control the total numer of cells, local refinments have been used .

Results

In the follwing we will present to you the results of the aerodynamic analisys of the tricanter centrifuge.

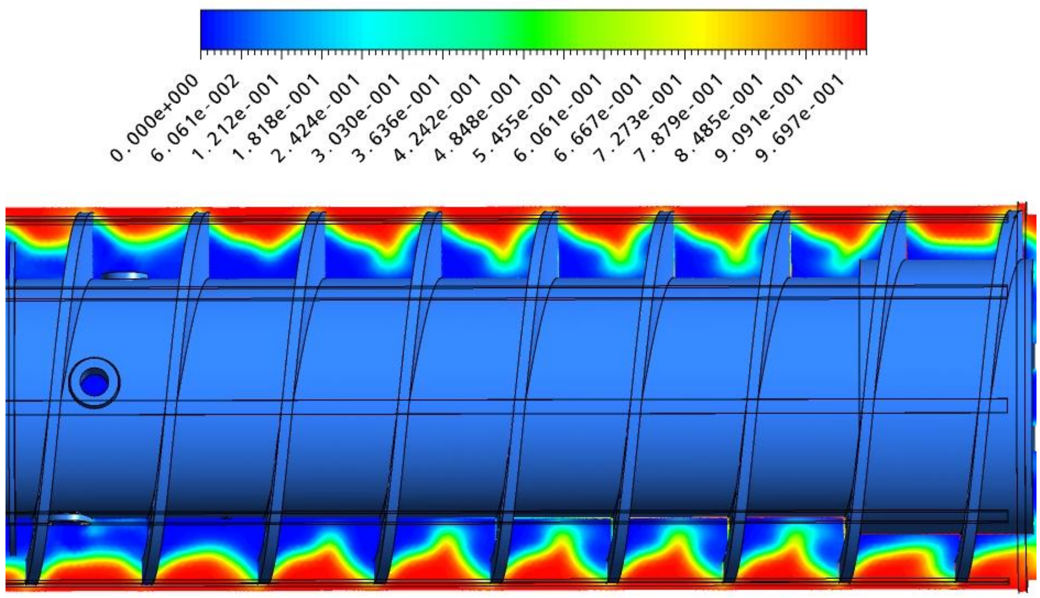


Fig. 3 Water volume fraction around the centrifugal rotor

As you can see in fig.3 the biggest water concentration is near the casing, and this is due to higher water density than the density of oil. Also the water concentration is zero or almost zero close to centrifugal rotor body.

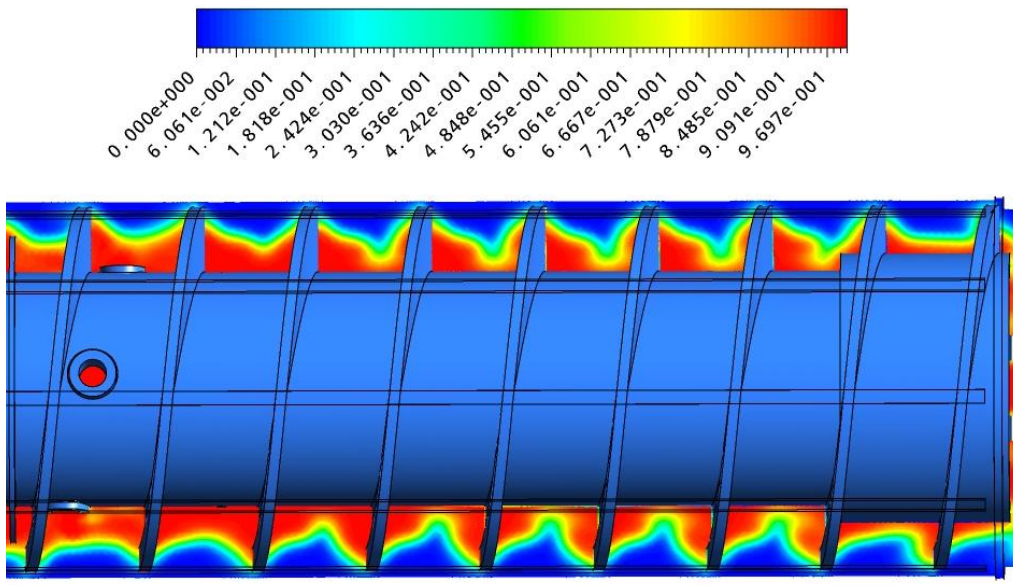
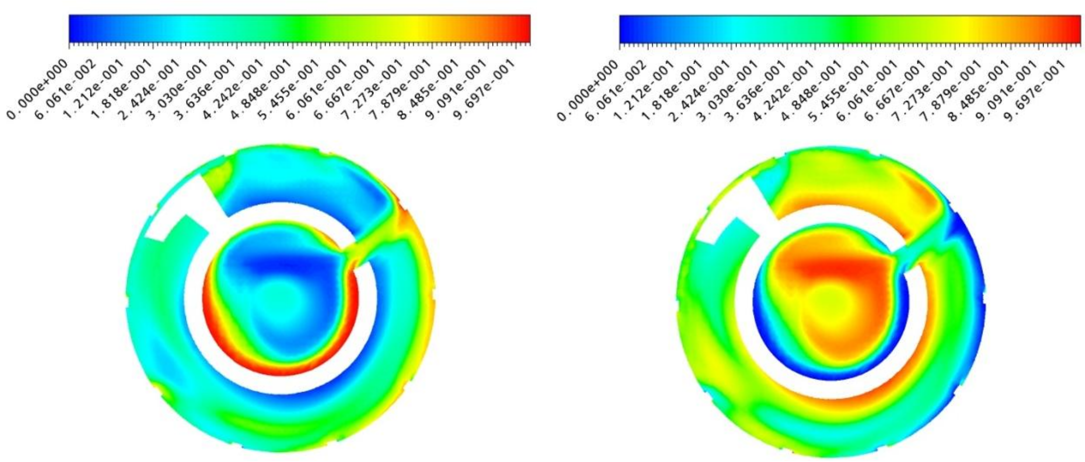


Fig. 4 Oil volume fraction around the centrifugal rotor

In figure 4 is it possible to see that oil concentration is almost zero or zero exactly where the water concentration is the biggest (fig. 3). This shows that the separation of the mixture starts around the centrifugal rotor.



a b

Fig. 5 Water (a) and oil (b) volume fraction at the first entrance in the centrifugal rotor

In figure 5 it can be observed the irregularity of the flow due to complex geometry. Also it can be seen that the mixture separation starts before entering the centrifugal rotor, as it can be observed the water is concentrated on the walls and the oil in the middle.

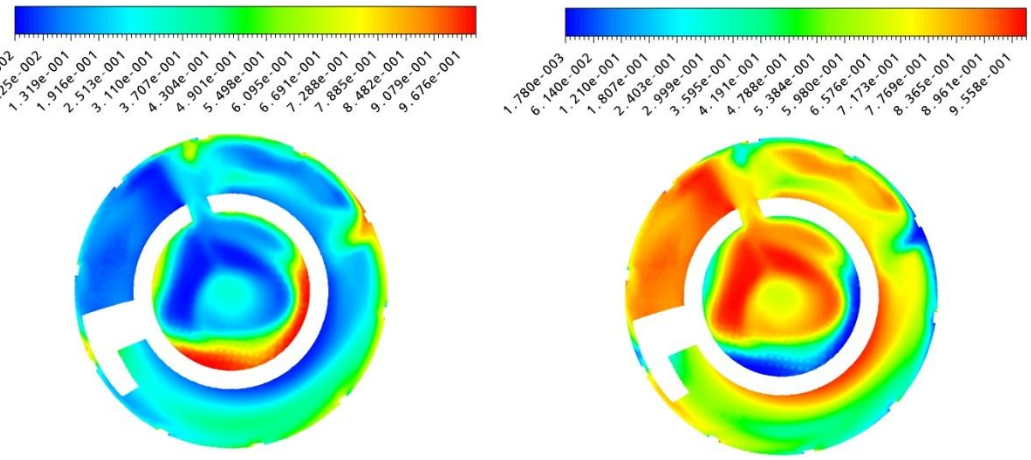


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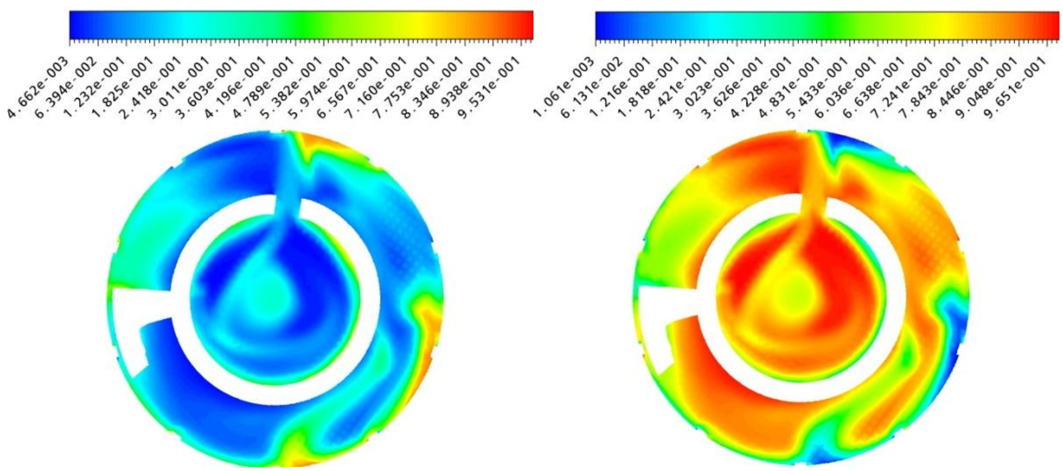
Fig. 6 Water (a) and oil (b) volume fraction at the second entrance in the centrifugal rotor

Also in figure 6 we can see the same characteristic of the flow.



a b

Fig. 7 Water (a) and oil (b) volume fraction at the third entrance in the centrifugal rotor

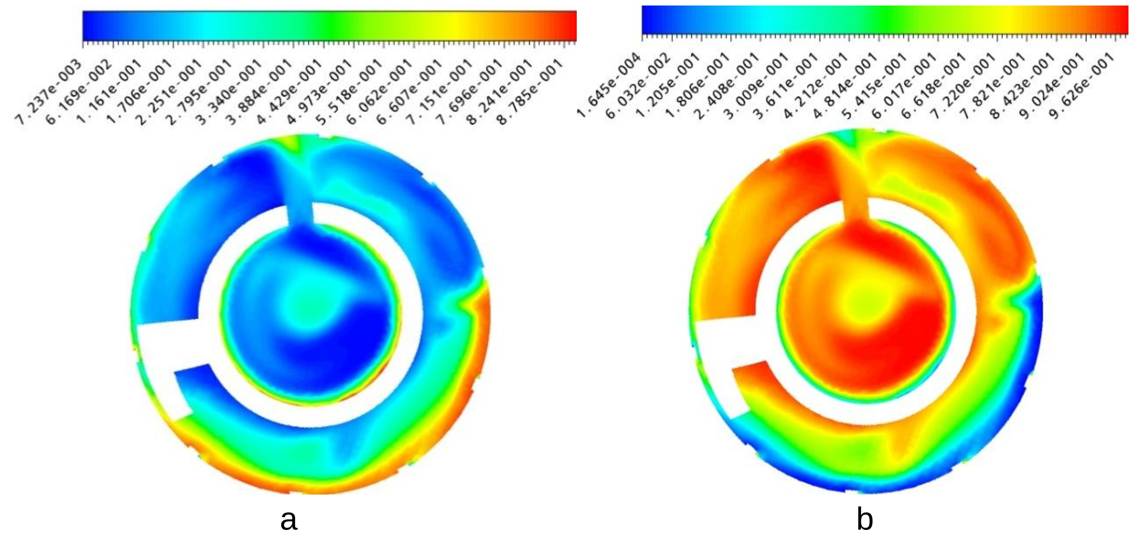


Fig. 8 Water (a) and oil (b) volume fraction at the fourth entrance in the centrifugal rotor

In figures 5-8 it can be observed how the water concentration is decreasing as the mixture aproces the fourth entrance. A possible explanation is that the water enters the centrifugal rotor faster then the oil through the first entrances.

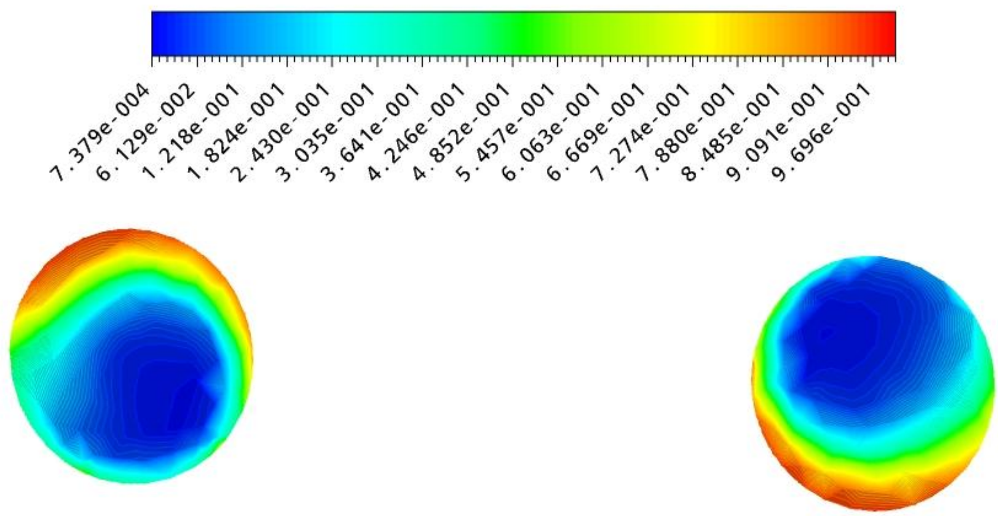


Fig. 9 Water volume fraction at oil outlet

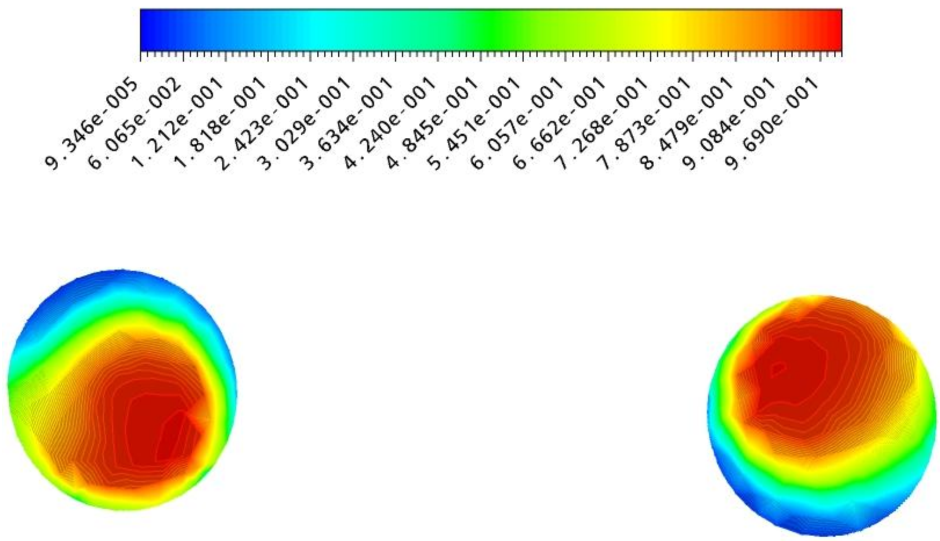


Fig. 10 Oil volume fraction at oil outlet

In figure 9 and 10 it can be observed that on the oil outlet the dominant fluid that exit through this outlet is oil.

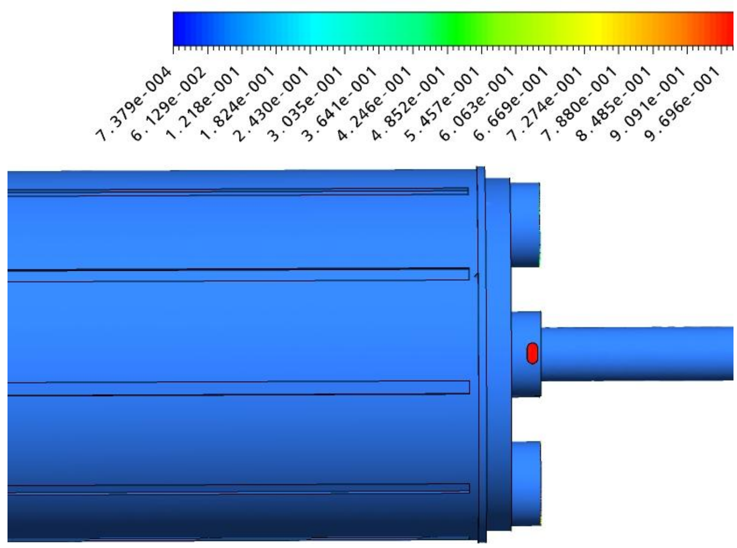


Fig. 11 Water volume fraction at water outlet

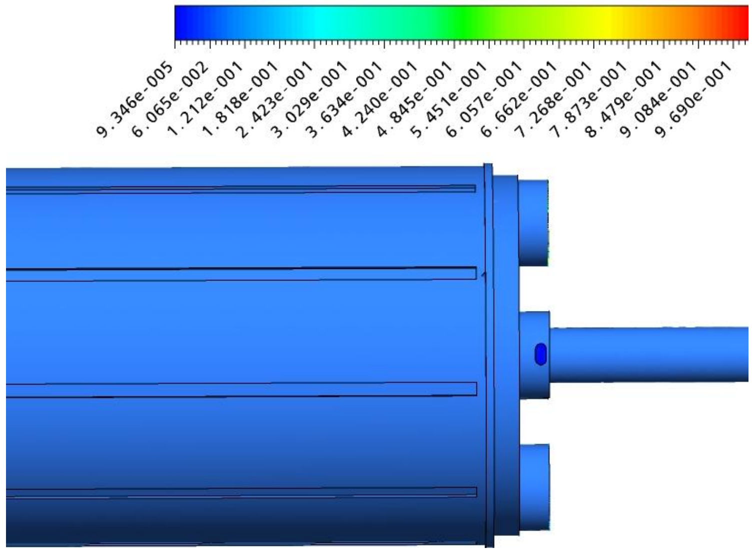
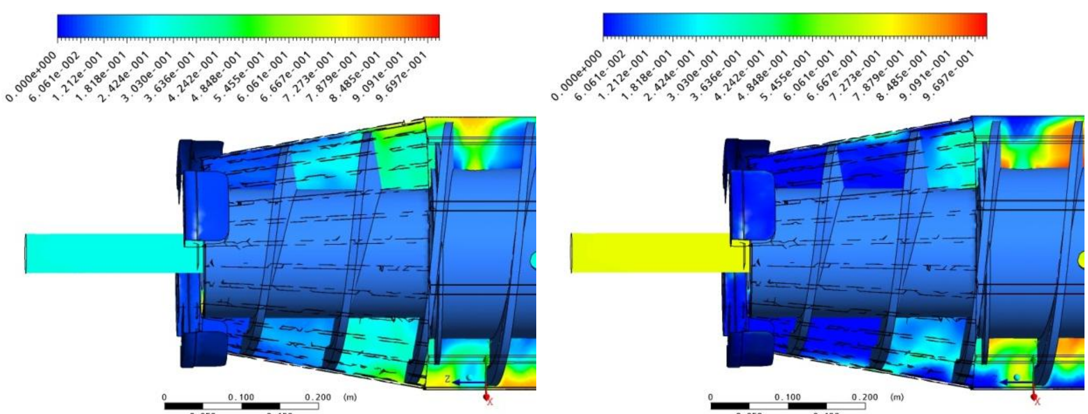


Fig. 12 Oil volume fraction at water outlet

In figures 11 and 12 it can be observed that on the water outlet exit only water.



a b

Fig. 13 Water (a) and oil (b) volume fraction in frontal part of the trincanter centrifuge

In figure 13 it can be seen that the water and oil concentration is low due to the fact that solid particles partially blocks the flow channel.

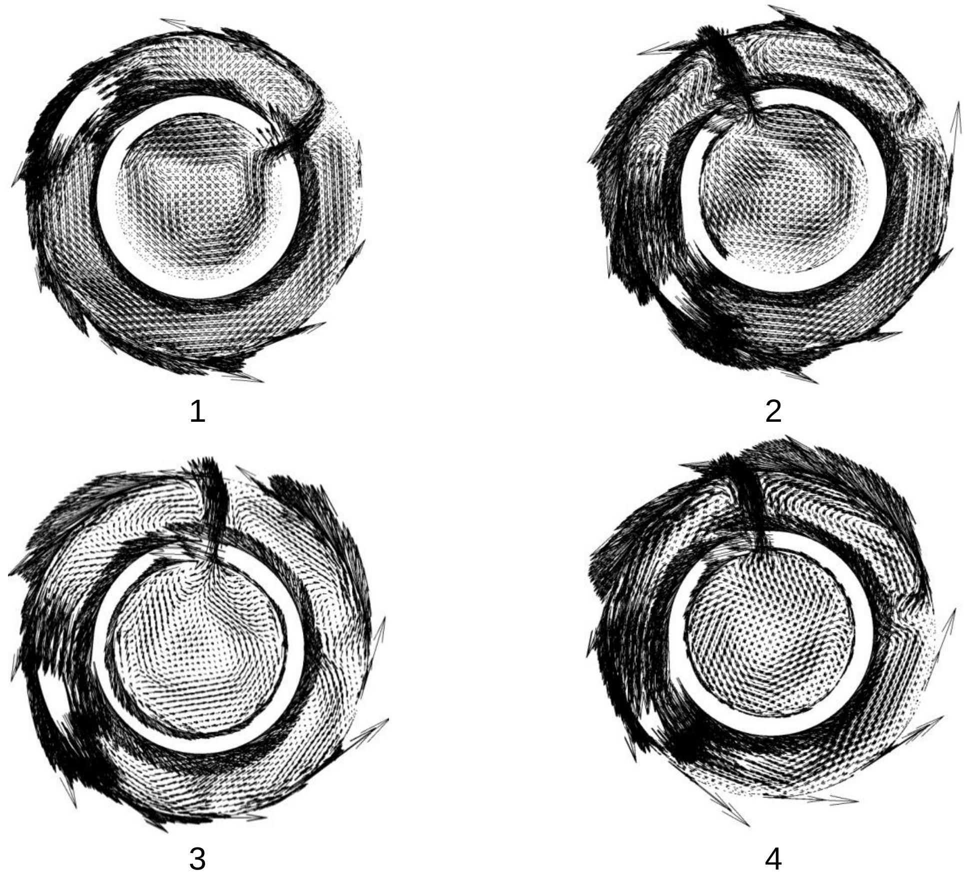


Fig. 14 Vector field at the entrances inside the centrifugal rotor

To understand how these solid particles behaves we have to observe first the vectorial field, fig. 14.

Here it can be observed the recirculation zones that produce before entering the centrifugal rotor and also inside the centrifugal rotor.

The recirculation zones affect not only the flow but also the solid particles behavior.

In figure 15 it can be observed where is the highest concentration of particles at the four entrances. This evolution show that the particles are cought inside the trincanter centrifuge between the recirculation zones.

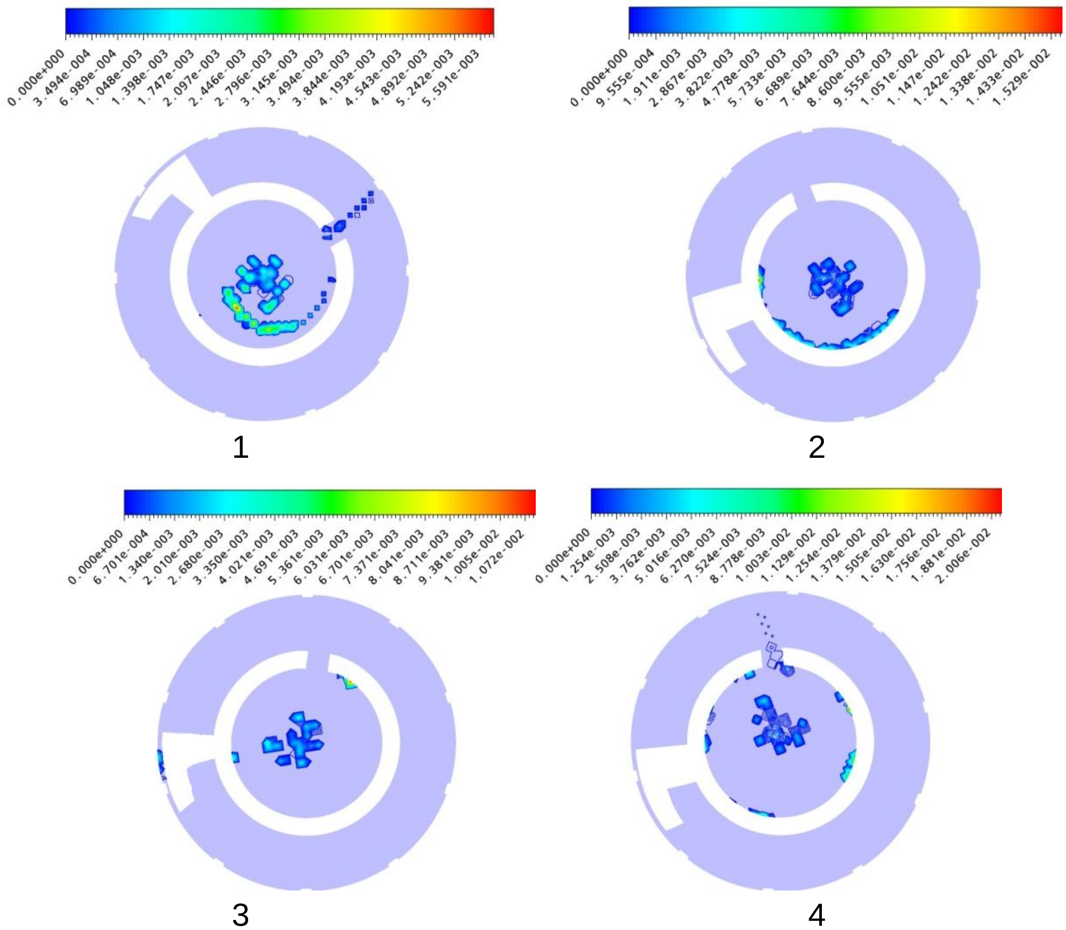


Fig. 15 Volume fraction of the solid particles in the domain

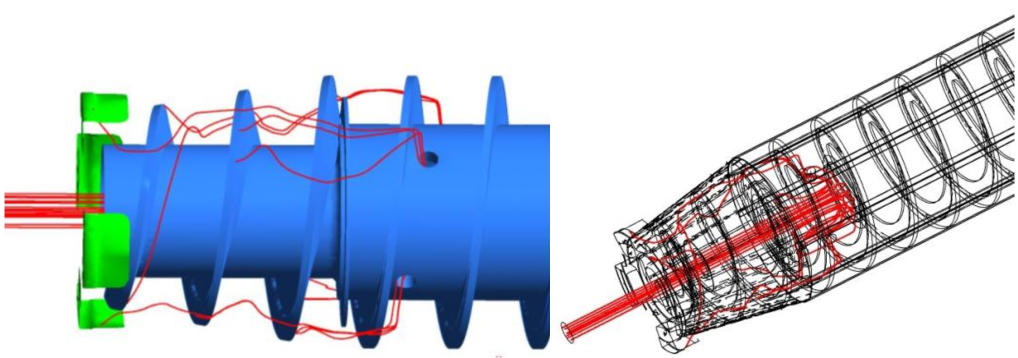


Fig. 16 Solid particles behavior in the domain

Particles evolution it is correct because the flow has an inverse sense of rotation than the casing and centrifugal rotor (fig.14). So the flow directs the particles that passes through the space between centrifugal rotor and casing (fig. 15) towards the outlet situated in the frontal part of the tricanter centrifuge.

4.10.4. Conclusion

1. The study was done using ANSYS CFX.
2. It was done the flow analysis of the flow inside the tricanter centrifuge and during this analysis has been consider the influence of oil, water and solid particles.
3. Inside the tricanter centrifuge the separation of water, oil and solid parts is done efficiently.
4. At the outlet it is possible to see that on the water outlet is comes out water, on the oil outlet comes out oil and on the solid particles outlet it comes out solid particles mainly.
5. In the next stages is proposed to run the of simulation program with other initial conditions both functional and technological.

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Anexa 4.1
Assembly drawings