

The Delta dryer

Theoretical and technological development of an energy-efficient dryer for sludge

Proefschrift

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Summary

The annual production of watery waste streams is significant. Considerable fractions are contaminated and cannot simply be disposed off. They must be stored or further treated. Because of problems associated with storage, the latter option is often more attractive. It involves destruction or immobilisation of the contaminants, and renders the residue suitable as a resource. Slurries with high organic content such as sewage sludge thus serve as biofuel, saving fossil fuels and energy crops. Mineral sludge such as dredging spoil can be converted into secondary building materials, replacing primary sand, clay and gravel.

An important step prior to the actual treatment is to dehydrate the wet substance. It usually necessitates a process of thermal drying, since the desired solids contents cannot just be obtained by mechanical means. Unfortunately, the thermal drying step is energy-intensive and contributes significantly to the total costs of sludge treatment. This is partly owing to the shortcomings of existing dryers: they consume more energy than necessary, require extensive exhaust air cleaning, and necessitate additional operations as they are susceptible to the varying consistency of the cake. Besides, the popular techniques show little room for improvements.

The present study focused on a new thermal drying concept that was developed to overcome the disadvantages of existing techniques. The concept is called Delta dryer. It mixes the mechanically dewatered sludge cake with hot steel balls, dries the mixture in a vacuum atmosphere, and then separates the dried product. The steel balls provide exceptionally large heat transferring surfaces and ensure that the rheology of the mixture doesn't change much when the sludge passes through the so-called sticky phase. The absence of air and other non-condensables avoids the problem of air cleaning, and gives the opportunity to recover part of the heat of evaporation by means of multiple stage flash evaporation. It also allows for relatively low drying temperatures, so that low quality heat sources can be applied.

This dissertation describes the work that was done to broaden the knowledge on the Delta drying process, to explore the potential of the concept, and to develop the accompanying technology towards an operational installation. The process was found to rely on the interplay of thermodynamic, kinetic and mechanical aspects, to be influenced by various design details, operational parameters and properties of the feed. Laboratory experiments were done on mechanically dewatered sludge cakes from different sources; detailed physical models were made of the processes inside the dryer; and a full-scale research prototype was developed with the aim to check the technical feasibility, to provide practical evidence on the drying principle, and to explore various implementation details.

The working principle of the Delta drying process was confirmed during this study. The laboratory experiments showed that the characteristics of the investigated sludge cakes are favourable for the feasibility of the concept, and that satisfactory high evaporation kinetics can be obtained. This, together with results from thermodynamic models, proves that the concept is capable of drying at a low heat input: less heat is necessary than the heat of evaporation of the water that is removed. According to estimated exploitation costs, the new technology not only saves energy over existing techniques, but can be economically competitive as well.

Unfortunately however, the process requirements were found to be strict and often conflicting, complicating the design and construction of a practical installation. Many technical difficulties were faced during tests on the research prototype. The responsible defects did not pose a fundamental threat to the concept, but were serious enough to abandon the early configuration. Hence, alternative design directions were explored.

Samenvatting

Jaarlijks komen grote hoeveelheden nat slib vrij als reststromen van industriële en huishoudelijke activiteiten. Een aanzienlijk deel hiervan is verontreinigd en kan als zodanig niet gedumpt worden. Het wordt opgeslagen in depots of verder verwerkt. De laatste optie wordt vaak gezien als de meest aantrekkelijke. Het gaat gepaard met het verwijderen of immobiliseren van de verontreinigingen, waarna het restant op een nuttige manier wordt afgezet. Slib met een hoog organisch gehalte zoals rioolzuiveringsslib kan worden omgezet in brandstof ter besparing van fossiele brandstoffen en biobrandstoffen afkomstig van gewassen. Mineraal slib afkomstig van bijvoorbeeld baggerwerkzaamheden en grondsanering kan worden ingezet bij civiele projecten als vervanger voor de primaire grondstoffen zand, klei en grind.

Het ontwateren van de natte substantie is een veelvoorkomende en belangrijke stap voorafgaand aan de eindverwerking. Hiervoor is men deels aangewezen op thermische technieken, omdat de gewenste drogestofgehalten niet worden verkregen met puur mechanisch ontwateren. Helaas zijn deze thermische methoden relatief duur en is hun energetisch rendement laag. Een analyse van reguliere droogtechnieken toont dat ze meer energie verbruiken dan noodzakelijk; ze vereisen ze vaak specifieke voorbehandelingen van het ingangsmateriaal vanwege de zogenaamde kleeffase van het slib; en ze vergen intensieve nabehandelingen van vrijgekomen processtromen. Hier komt bij dat de betreffende technieken weinig ruimte tonen voor verbeteringen.

Een nieuw droogconcept genaamd Delta dryer is ontwikkeld als antwoord op de genoemde nadelen. Het mechanisch ontwaterde slib wordt bij dit concept gemengd met hete stalen kogels, gedroogd onder gereduceerde druk, en vervolgens gescheiden. De kogels leveren een groot warmteoverdragend oppervlak en zorgen ervoor dat de transporteerbaarheid van het mengsel niet verandert wanneer het slib overgaat van een verpompbare substantie via de kleeffase in een vaste stof. De afwezigheid van lucht in de installatie bespaart een intensieve reiniging van de uitstoot, en biedt de mogelijkheid om een deel van de verdampingswarmte terug te winnen d.m.v. het principe van meertrapsverdamping. Ook kan droging plaatsvinden bij relatief lage temperaturen, zodat laagwaardige warmtebronnen kunnen worden benut.

Het onderzoek dat wordt beschreven in dit proefschrift had tot doel om de kennis van het Delta dryer proces te verdiepen, het potentieel van het concept te verkennen, en de bijbehorende technologie te ontwikkelen richting een operationele installatie. Het droogproces is het resultaat gebleken van een nauw samenspel van thermodynamische, kinetische en mechanische factoren. Het wordt beïnvloed door diverse ontwerpdetails, procesvariabelen en fysische eigenschappen van het slib. Laboratoriumexperimenten zijn uitgevoerd om de slibeigenschappen te beoordelen; natuurkundige modellen zijn opgesteld op basis waarvan invloeden van ontwerp- en procesvariabelen zijn voorspeld; en een onderzoeksprototype is ontwikkeld om de technische haalbaarheid te toetsen, het droogprincipe te demonstreren, en diverse implementatiedetails te onderzoeken.

De studie bevestigt het werkingsprincipe van het concept. De eigenschappen van de onderzochte slibsoorten blijken gunstig voor het proces. Aan de hand van gemeten verdampingskinetiek en thermodynamische modellen kan geconcludeerd worden dat de Delta dryer in principe in staat is om te drogen met minder energie dan de verdampingswarmte van het te verwijderen water. Volgens geschatte exploitatiekosten kan de nieuwe techniek niet alleen energie besparen, maar ook economisch concurreren met commerciële drogers.

Helaas is het lastig gebleken om het concept te vertalen in een praktische installatie. Dit komt door de strenge en vaak tegenstrijdige eisen die gesteld worden aan het ontwerp en de uitvoering. Als gevolg van een aaneenschakeling van technische

problemen met het onderzoeksprototype is besloten van de oorspronkelijke configuratie af te wijken en nieuwe uitvoeringsvarianten te verkennen.

Chapter 1: General introduction

Sludge or slurry is a multi-phase substance consisting of fine solids in a liquid and an amount of gas occupying empty pores. It abounds in nature, but is considered here the by-product of domestic or industrial activities. Characteristic are the significant volumes involved, the presence of contaminants, and its low value. Although it may hold useful constituents as well, sludge is usually regarded as waste.

Until recently, traditional disposal routes for sludge were self-evident, popular options being landfilling, river dumping and disposal at sea. Increasing supplies as a result of growing population and expanding industrialization have raised serious concerns about these options. Environmental problems associated with disposal are generally no longer accepted. As an alternative, controlled storage in basins is problematic as well. It demands large areas in often densely populated regions that must be maintained and monitored for indefinitely long times in order to prevent leaking. Governmental policies, based on the public and political awareness of these issues, are to decrease disposal and encourage the use of sludge as a resource. The latter necessitates some form of treatment that renders the sludge suitable as fuel or converts it into secondary building materials. It has become more interesting with the increasing costs of raw materials.

A necessary first step in sludge treatment is often to separate moisture from the wet substance. It reduces weight and volume of the residual fraction, which can be handled, transported, and undergo further treatment such as high temperature processing in a more beneficial way. As the desired solids content cannot be obtained just by mechanical dewatering, the use of thermal methods is inevitable. The Delta drying technology was developed for this particular purpose.

This introductory chapter presents important sources of sludge, various processing routes, and the role of the thermal drying step herein. It clarifies the motivation for the development of the Delta dryer. Also discussed in this chapter is the outline of the dissertation.

1.1 Production of sludge

The annual production of watery waste streams is significant and a worldwide problem. Sources are communal and industrial wastewater treatments, manufacturing operations, chemical processes, agricultural activities, dredging, forest industries, and mining. (See also Table 1.1) The compositions of these substances depend on their origin and preconditioning. Solid contents range from a few percents for sewage and paper sludge to 50% for chicken manure. Solid fractions vary from purely mineral to primarily organic.

Although the sludge or cake can be a valuable resource, as is the case for wet fines from coal tailings recovery, it often contains unattractive and hazardous ingredients. Trace contaminants such as heavy metals are frequently present. The same counts for organic constituents such as oil, grease, PCB's, PAH's, pesticides from farming, and other types of poison from industries and house holdings. Sewage sludge and manure add pathogenic micro-organisms such as bacteria and viruses to this list.

Table 1.1: Sources of contaminated sludge in the Netherlands, excluding those of manure and digestate. The second column indicates the amounts after mechanical dewatering.

<i>source</i>	<i>dewatered [10⁶ kg/year]</i>	<i>dry solids [10⁶ kg/year]</i>
<i>dredging spoil, class 3/4 [Rienks, 2007]</i>	<i>7000-10000</i>	<i>3500-5000</i>
<i>residue from soil cleaning [RIZA, 1998]</i>	<i>100-150</i>	<i>50-75</i>
<i>sewage water treatment [CBS, 2007]</i>	<i>1500</i>	<i>350</i>
<i>industrial waste water treatment [CBS, 2007]</i>	<i>950</i>	<i>175</i>
<i>paper production / recycling [AOO, 1998]</i>	<i>300</i>	<i>150</i>
<i>drinking water purification</i>	<i>50</i>	<i>25</i>

During the past decades, four important global developments have drawn a growing attention to sludge management.

- Volumes produced have increased substantially.
- Environmental burden and health risks have risen as ongoing urbanization leads to concentrated waste production and limited number of disposal sites.
- Negative effects of human activities on the environment have become more and more recognized and addressed.
- Costs of raw materials have increased.

In line with these ongoing developments, pressure against continuation of traditional disposal policies has risen. Ordinary dumping of contaminated matter is no longer accepted; agricultural application is considered an option only under strict conditions; and storage is not regarded a durable alternative and therefore questioned more and more. Treatment prior to disposal or reuse is recognized as the answer to the sludge problem, generating a demand for cost-effective and sustainable process schemes.

Main drivers for the desired changes are set by governments. Legislative restrictions, taxes and subsidies discourage unfavourable tendencies, support initiatives in preferred directions, and sustain a market for new technologies. A number of recently adopted regulations underline the presence of the sludge issue in the political agenda. In the Netherlands, for instance, tightening environmental standards for land application have banned the use of sewage cake as fertiliser [LNV/VROM, 1998]. Additional requirements and conditions for dumping permits have ruled out direct

landfilling of sludge with high organic contents, and set penalties on landfilling of sandy mineral sludge. Another example is a European directive, which poses upper limits for the amounts of nitrate and phosphate to be applied to cropland [MNP, 2007]. It yields a manure surplus to be treated in one way or the other.

1.2 Drying as part of sludge treatment

Various treatment options exist for contaminated sludge. Depending on the nature of the matter, these involve dehydration, disinfection and immobilization or removal of hazardous compounds. The scheme of Fig.11 shows 10 main routes.

Options 1, 2 and 3 in Fig.1.1 are for mineral streams such as dredged material, residue from soil cleaning, and sludge from drinking water purification. The first option does not involve real treatment, although it may be combined with landfarming to break down mineral oils and PAH's. The other two routes start with sand separation by means of hydrocyclone or sedimentation processes. The recovered sand is usually clean and applicable as building sand. The residual fines, which contain most contaminants, are dewatered by means of filter presses, vacuum filters, or centrifuges, and then either disposed or further treated. The latter necessitates an increase of the solids content up to 80-90%. It is partly achieved by thermal means, as mechanical techniques are capable of dewatering to a maximum of only 50% dry solids. The next and final step involves a process of cold or thermal immobilization. Cold immobilization is based on the addition of binders, such as cement, to chemically fix some of the contaminants and prevent others from leaching to the environment. It yields a civil construction material in which the fines replace primary minerals. The process of thermal immobilization uses temperatures in the range of 1000-1400°C that make the solid particles sinter or melt together. Organic compounds are destroyed while heavy metals are partly evaporated, chemically bound, and entrapped in a hard aggregate such as gravel or bricks. Several commercial initiatives produce these aggregates on an industrial scale [Mushalla, 2004], [Ulbricht, 2002]. Leaching tests on sintered pellets from class 4 dredging spoil have demonstrated that, with carefully controlled manufacture, there are no environmental limitations to their application as a building material [Schouten, 1989].

For sludge types with high organic contents, landfilling without pre-treatment is prohibited and agricultural application is strongly restricted in the Netherlands. Dutch legislation requires the organic fractions to be beneficiated first, which, in many cases, leaves routes 4-7 in the scheme of Fig.1.1. The final destination of the residues is the same as for mineral sludge. They are disposed in a landfill, sintered or cold immobilized. The main operations are [AOO, 2002]:

- Stand-alone combustion in a sludge incineration plant, with or without supporting fuel
- Co-combustion in a municipal waste incinerator (Amsterdam, Monaco)
- Co-combustion in a power plant, generating green energy
- Incineration in a cement kiln, using organic compounds as fuel and retaining inorganic constituents in cement clinker
- Gasification, producing synthesis gas (H_2 , CO) or fuel gas (CH_4)
- Pyrolysis, producing gas, liquid and solid fractions to be used as fuel
- Wet oxidation at elevated pressure

Except for wet oxidation, which is not practiced nor given preference in the Netherlands [VROM, 2003], the benefits of dehydration prior to these operations are evident. It increases the calorific value of the cake and leads to reduced consumptions of fossil fuel. Unfortunately, however, dewatering characteristics of organic sludge are poor. Maximum solids contents around 20-30% are obtained by mechanical means. Further dehydration relies on energy intensive and expensive thermal methods. Still, application of these methods is advisable: it is environmentally and economically favourable in case rest heat from the main operation is used [Stowa, 2005]. For some processes, solids contents around 90% are simply a necessity and thus not open to discussion [Erens, 2003], [Den Uil, 2004].

The remaining treatment routes in Fig.1.1, indicated by numbers 8, 9 and 10, are mainly for manure and sludge from food and beverage industries. Besides incineration and direct agricultural use, these waste streams may be turned into granules to be exported and sold as fertilizer. Thermal drying can be the final treatment step after digestion or the main treatment that simultaneously disinfects the product.

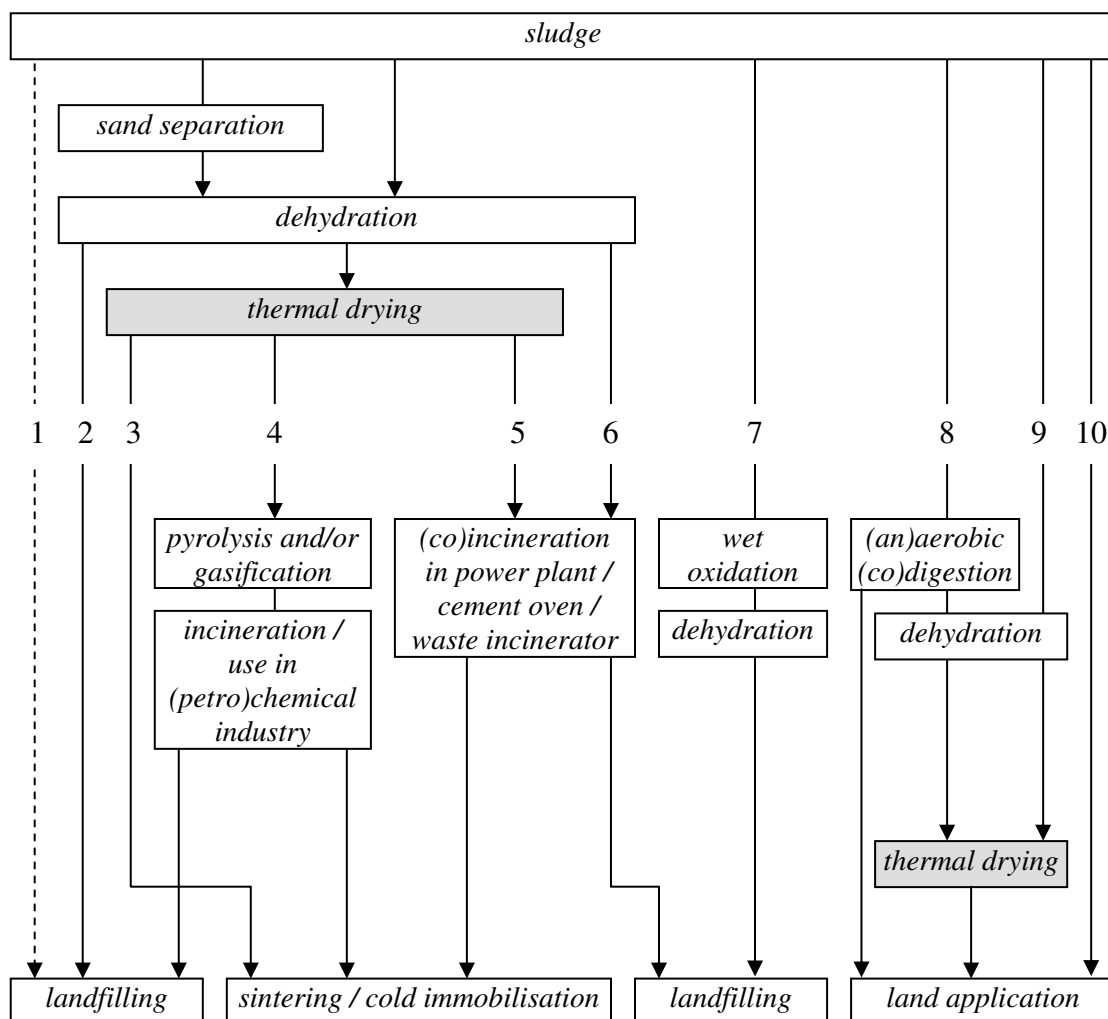


Figure 1.1: Main treatment and disposal routes for sludge. In addition to these process steps, it is often necessary to treat flue gasses from thermal processes and water effluent from dehydration and thermal drying.

1.3 Motivation for this work

Depending on the source and selected processing route, the cost of sludge treatment can amount to 90-140 euros per ton of mechanically dewatered cake [AOO, 2002], [Nielsen, 2003]. Thermal drying is an energy-intensive step, contributing significantly to the total costs. (See also Paragraph 2.4) In order to make the process more attractive, the Delft University of Technology and DHV Environment and Infrastructure worked together and formulated ideas for a new drying technique. The resulting concept, called Delta dryer, was internationally patented [Van Poppel, 1998, 2000]. Its key elements are the use of hot metal balls as heat transferring medium and vacuum drying to reduce off-gas and to recycle part of the heat of evaporation.

The Delta dryer concept is based on an efficient drying scheme. Although this scheme is relatively simple, its success relies on a complex interplay of thermodynamic, kinetic and mechanical aspects. Consequently, its operation is influenced by various design issues, operational details, and sludge properties, some effects being hard to foresee. Knowledge and understanding of the process is essential in order to develop the concept and accompanying drying technology further towards an operational installation. The explorative work described in this dissertation contributes to that. It is based on projects subsidized by Senter BSE (agency of the Dutch ministry of economic affairs), the European Commission, and Dutch engineering company and equipment manufacturer Van Tongeren Kennemer BV. The projects were carried out by the University of Delft, DHV Environment and Infrastructure, and Van Tongeren Kennemer BV, who cooperated as partners.

1.4 Structure of the dissertation

The nine chapters of this dissertation contain results from a literature study, fundamental research, laboratory experiments, design exercises, and prototype tests. Each chapter starts with a brief introduction.

Chapter 2 reflects on the present state of drying technology. Commercially available techniques are reviewed, as well as novel methods and interesting patents that are applicable to large-scale drying of mechanically dewatered sludge cakes. The strengths and limitations of these techniques are discussed; operational costs and energy consumptions are compared; and attention is paid to possible energy saving measures.

Chapter 3 introduces the Delta dryer concept. The starting points for the concept are explained, as well as the principal thermodynamics behind its operation. A very simple model indicates the theoretically maximum potential of the drying scheme in terms of energy efficiency. The last paragraph presents the various process steps for the standard configuration.

The process steps rely on specific properties of the cake. Most sludge types are regarded as waste. Hence, their thermal, rheological, and hygroscopic behaviour (which are of main interest here) are unknown at this point. Chapter 4 explains how the properties were derived empirically, and how the results impinge on the drying process.

Chapter 5 consists of two parts. The first part presents a thermodynamic model, describing the macroscopic heat and mass flows in the core of the Delta drying process. This model accompanies software, which enables the user to simulate conditions inside the dryer and investigate effects of design specifications, operational parameters, and sludge properties on the thermal performance. Results from such simulations are evaluated in the second part of the chapter.

The macroscopic model of Chapter 5 does not take into account every aspect of the Delta dryer. Like any model, it offers a simplified view upon reality. Two shortcomings in particular undermine the reliability of performance predictions. These shortcomings relate to the kinetics of evaporation and condensation processes. They are addressed in Chapter 6.

A full-scale research prototype was developed to check the technical feasibility of the Delta dryer concept. Various implementation details were investigated. Main units of the apparatus were tested separately and then joined to form an operational installation. Some preliminary drying exercises were carried out as well. The hands-on experience is described in Chapter 7.

Some important findings from previous chapters come together in Chapter 8. They show the actual potential of the Delta dryer technology, taking into account some room for improvements. A few limitations appear hard to overcome when sticking to the standard configuration. Two new development directions are therefore suggested. These are based on the same thermodynamic format, yet are to eliminate important limitations of the early design.

Chapter 9, the final chapter, summarizes all main conclusions to this thesis and gives recommendations for further work.

Chapter 2: Existing drying techniques

Thermal drying has applications in many industries, ranging from agricultural, mining, food, chemical, manufacturing to energy sectors. A large number of techniques have been developed over the years. Some of them were tailor made for a particular product while others are more universally applicable. The fundamental working principle of all methods, however, is the same. They involve at least some movement of the material to be dried, heat transfer to the material, and removal of released vapour from the drying scene. Fig.2.1 schematically presents these steps.

This chapter describes the various options to cover the three elementary tasks, as well as their implications for the drying process. It confines itself to methods that are suitable for large-scale deep dehydration of mechanically dewatered sludge cakes. The first paragraph examines the different modes of heat transfer, i.e. radiation, convection, and conduction. Movement of the cake involves transport through the drying stage and usually some form of agitation. Depending on the dryer type, the wet substance may be poorly mixed to intensively mixed, which is the point of interest in the second paragraph. The third paragraph focuses on the released vapour. More specifically, it examines systems that bring the vapour back to the process with the intention to recover latent heat (indicated by the dashed line in Fig.2.1).

The final configuration determines the potential of a drying installation, the economy of which is captured in two parameters: energy consumption and exploitation costs per unit of evaporated water. This chapter concludes with an overview of these two primary parameters for several drying techniques.

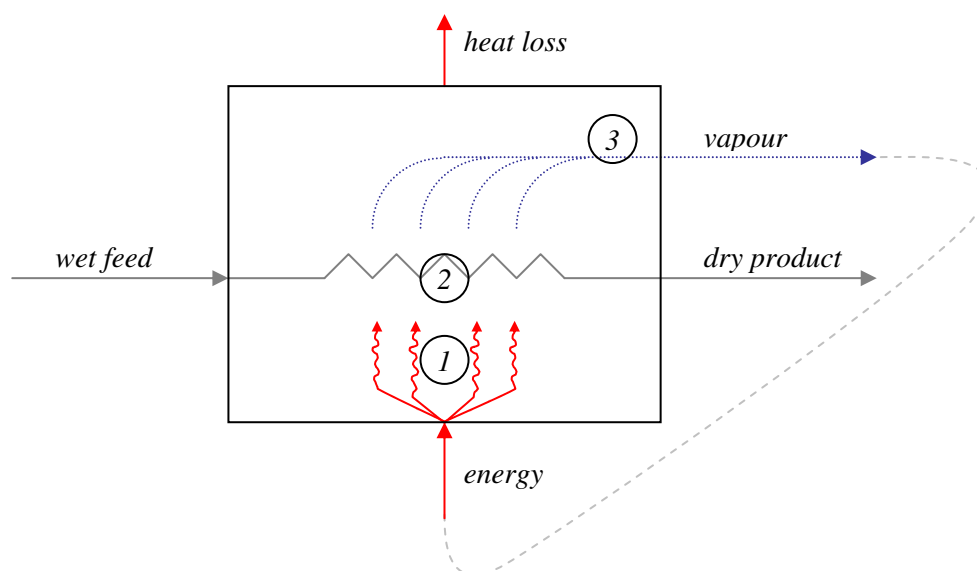


Figure 2.1: Basic features of a thermal dryer: heat supply to the surface of the cake (1), transport of the material (2) and removal of released vapour (3).

2.1 Modes of heat transfer

2.1.1 Radiation

Heat transfer occurs through radiation, convection, conduction, or a combination of these. The use of solar radiation to heat the wet product and evaporate moisture is without doubt the oldest of all drying techniques. The classical variant, which is the open-air bed, uses a minimum of technology by letting natural convection carry away the vapour. This method is not suitable for products like sewage cake or manure, since it has the disadvantage of emitting bad odours. The bed easily becomes a breeding ground for insects and germs as well. Greenhouse tunnel plants avoid hygiene problems by providing full shelter for the process. The plants run in an automated batch mode. Solids handling equipment feed and empty the bed, and move and aerate the cake in the time between. Controlled ventilation takes away the vapour. The air stream is occasionally preheated in a collector or by auxiliary means, so that the cake heats up by convection as well.

The equipment of a greenhouse tunnel lends itself for small-scale, decentralized dewatering of sludge cakes. It is relatively cheap and energy efficient. The standard solids handling instruments consume typically 40-110kJ of electric energy per kilogram of evaporated water [Ist-Anlagenbau, 2006], while fans add an equal amount to that. As a downside, however, the method exhibits low evaporation rates that decrease further and further as the cake dries. A greenhouse plant therefore requires large areas to be built. According to literature, a capacity of 500kg of water per square meter per year would be feasible in the West European climate [Warmerdam, 2001]. Still, solar sludge drying offers little prospects in the Netherlands because of the high relative humidity, low solar intensity, nature's unpredictability, and space required in densely populated areas.

Besides the exploitation of a natural resource, the use of artificially generated radiation is also a possibility. Most interesting from this perspective is dielectric drying, which subjects the product to electromagnetic radiation in the microwave or radio frequency spectrum. The oscillating radiation causes rapid changes in the orientation of dipole molecules such as water, and hereby produces heat. Unlike other methods that rely on heat flows from the product surface inwards to the wet bulk, the heat is directly generated in all wet parts throughout the material. This leads to a more beneficial mode of mass transfer than in surface heating [Strumillo, 1986], and is particularly interesting for poor thermal conductors such as moist particulate matter.

Dielectric dryers are unfortunately not energy efficient. It costs approximately 3.6MJ in the form of microwaves or radio waves to evaporate one kilogram of water [Van Loock, 1987]. Because of the low efficiency of electric power conversion into microwaves (50-60%), the inefficiency of a power station, and losses in the electricity network, the primary energy consumption easily amounts to 15MJ/kg. This is unacceptable for large-scale dehydration of low-value sludge cakes.

2.1.2 Convection

Convective dryers blow hot, dehumidified gases through a chamber against the surface of the cake. The drying agent is usually a mixture of air and combustion gases, with an inlet temperature that varies between 100°C and 800°C depending on the type of installation. Cooling of the gas stream provides the heat of evaporation. The air stream itself takes up the vapour and carries it away from the surface.

Heating of the gas can be direct, using the exhaust from a burner, or indirect by means of a heat exchanger. The most straightforward method is to let the burner fire directly into the drying chamber. This is not advisable for every feed type, but the optimum way to supply heat, bringing more than 90% of the primary energy to the product. Unfortunately, the unfavourable thermal properties of gases affect the amount of heat that actually reaches the surface of the product. The low heat capacity of the medium necessitates large volumes per unit of evaporated moisture, whereas the poor thermal conductivity raises the need for high flow velocities to intensify contact with the product.

The use of volumetric gas streams automatically brings significant expenditures for exhaust air purification, since the flue gas together with collected vapour and dust cannot just vent off to the atmosphere. Cyclones, bag filters, wet scrubbers, condensers, biological filters, and thermal oxidizers are applied to remove fine particles, vapour, heavy metals, volatiles and odours. A second drawback of volumetric air streams is the heat loss involved as the warm gas leaves the installation. The losses affect the overall efficiency of the drying plant unless large fractions of it are recovered. A popular method is to transfer the low-grade exhaust gas to other operations. An alternative is to reintroduce part of it to the inlet stream, or to recirculate an indirectly heated and repetitively dehumidified gas. For products with high organic content, the use of an inert medium running in a fully closed circuit is advisable for safety reasons. The product would otherwise bear the risk of fire or dust cloud explosions.

Superheated steam dryers are special types of indirect convective dryers that, instead of an inert gas, circulate superheated steam to heat the product. As the cake dries, the agent cools and shifts towards saturation. Excess steam is removed from the stream. The rest is brought to higher temperature and reintroduced to the process.

Thermal properties of steam are superior to those of air. The medium is a better thermal conductor, exhibits higher specific heat capacity, and shows no resistance to diffusion of evaporated water. Unfortunately, these superior properties do not always yield increased evaporation rates compared to convective air-drying. This is because the temperature of a wet surface in superheated steam (i.e. the saturation temperature at the operating pressure) is higher than the wet bulb temperature in hot air drying at comparable operating conditions [Van Deventer, 2004]. Hence, the temperature difference between the medium and the product, which is the driving force for heat transfer, is lower. The gas temperature at which superheated steam drying yields enhanced evaporation rates is called the inversion point. It is somewhere between 160°C and 200°C [Kudra, 2002]. As the process often takes place well above atmospheric pressure, superheated steam dryers need to be leak proof and equipped with special feed and discharge operations.

2.1.3 Conduction

The method referred to as contact drying is based on heat supplies via conductive surfaces. The surface can be a wall, tray, cylinder, band or a series of hollow blades, dishes, etc. (or a combination of those), designed to offer as much contact area as possible. It receives energy from steam or hot oil in turn. A small sweep of air is normally present to carry away the water vapour. Unlike the gas stream in convective methods, this air stream has no other function than to serve as a carrier. Hence, the required flow velocities are much lower, exhaust air treatments are less expensive, and less heat emits to the atmosphere.

Another potential advantage of contact drying over convective drying is a fundamentally more beneficial mass transfer mode in the product. Moisture transports from the bulk to the surface may run anti-parallel to the temperature gradient whereas for convective drying they always point in the same direction. Pressure gradients, which are the result of the temperature gradients, thus assist these moisture transports. Contact drying also allows special installations to operate under vacuum conditions, i.e. in absence of noncondensables. These so-called vacuum dryers show several advantages: they display increased evaporation rates at lower process temperatures; they allow for advanced energy saving measures (described in Paragraph 2.3); and, like inert gas dryers, they eliminate the risk of spontaneous combustion of the product.

The use of contact techniques is not profitable on every occasion. Compared to direct-heat convective appliances, the heating medium usually has higher losses at its point of generation and in its transport to the installation. Generation of steam through burning of gas or oil, for instance, has thermal efficiencies of typically 80%. It is not very rare when another 20% of the primary energy is lost in the distribution system [Mercer, 1994]. Moreover, steam and oil do not allow high temperatures, so that the capacities of contact dryers are limited. Up scaling is also less effective than for convective methods, as the amount of heat transferring areas per unit of volume decreases.

2.2 Transport and agitation of the cake

Thermal dryers often show great structural similarities even though they may rely on different modes of heat transfer. Convective rotary dryers, for instance, have much in common with their contact version, the steam tube rotary dryer. A number of appliances operate in a dual-mode as well, e.g. both convective and conductive at the same time. Characteristic in those cases is the state of the product and the way it moves through the installation. Depending on the used technique, the drying bed varies from stationary to agitated, suspended and highly turbulent. Every mode has its own strengths and weaknesses. More turbulence generally results in increased evaporation rates.

2.2.1 Stationary bed dryers

Stationary bed dryers provide minimum movement of the product. The moist particles remain mainly at rest throughout the process. This approach results in low machine and maintenance costs, but also in low thermal efficiencies.

Typical stationary bed dryers are tray dryers. They traditionally operate atmospherically in a batch mode. Wet material is spread onto trays, which are then

stacked and placed in an oven with forced air circulation. The residence time in the oven is very long, and that is why this procedure is suitable for small-scale applications only.

More advanced, (semi)continuous tray dryers do also exist. These have larger capacities and higher drying rates. They contain stacked circular trays with a feed inlet at the top. A vertically rotating central shaft with horizontal arms and scrapers spreads the material in thin layers and moves it from one tray to the tray below. Meanwhile, heat is transferred to the surface of the cake by convection, and to the bottom of the cake by conduction; or, as in vacuum tray dryers, heating is purely conductive via indirectly heated hollow trays. The dried product is eventually collected from the bottom tray.

Another well-known stationary bed type is the band dryer. This method requires granulation of the mechanically dewatered cake. The preconditioned feed is subsequently distributed over the surface of a perforated metal belt that passes through an oven. The cake warms up and dries while hot air flows over the bed and through the openings of the belt. The exposure time is not short but the width of the belt and furnace is easily adaptable to allow capacities up to 10 tons of water per hour [Andritz, 2006].

2.2.2 Centrifugally agitated bed dryers

The common tactic to increase drying rates is to stir up the wet substance. Centrifugally agitated bed dryers, better known as rotary dryers, rely on this tactic. The core of these machines consists of a revolving tube, or of a stationary cylindrical housing with coaxially rotating shaft (Fig.2.2). The cylinder or shaft contains a series of moving peripheral vanes that break and agitate a continuous feed of wet material, uncovering fresh product surface. The vanes also convey the material across the length of the housing, sometimes assisted by a slight inclination of the machine to the horizontal.

Heat transfer in rotary dryers is based on convection, conduction, or a combination of both. Convective types are constructed and controlled in such a way that the granulated cake is well exposed to the hot gas stream. The granules are lifted, finely distributed and showered in the stream, while moisture evaporates and moves along with the gas.

Conductive rotary dryers transfer heat to the cake via hot surfaces. They can resemble almost standard screw conveyors with a double walled hull in which hot oil or steam circulates. Specialized installations contain means for improved contact between the product and heat transferring surfaces. Disc dryers use a number of indirectly heated, rotating hollow discs to enlarge the heat transferring areas. The high-speed paddle dryer comprises a rotor fitted with paddles that move at high peripheral speeds, intended to increase the turbulence in the bed. The paddles break the product into small pieces and centrifuge them in thin layers against the hot inner surface of the hull.

A difficulty shared by many practical installations, including rotary dryers, is their susceptibility to the consistency of the cake. Granulation, which is an essential process step, can be a simple preconditioning operation. However, it is problematic for feed types that exhibit a so-called 'sticky phase'. This sticky phase expands over a range of moisture contents at which the substance tends to adhere to metal surfaces and agglomerate (see also Section 4.1.2). It leads to the formation of incrustations on

hot surfaces of contact dryers or the formation of large lumps in convective dryers, in either case resulting in reduced heat transfer rates.

To avoid the sticky phase, the wet feed is often premixed with solids from other sources or with a re-circulated part of the dried product. This can be an intensive process. Another remedy described by several old patents is the addition of inert coarse-grained particles to the wet feed. These particles may be steel balls that, at the end of the process, are separated from the dried material and recycled. Some patents propose to preheat the particles with exhaust gas from a burner and to use them as a heating medium [Wilmot Engineering Company, 1967], [Bearce, 1977]. It is hard to find any practical example of this idea though.

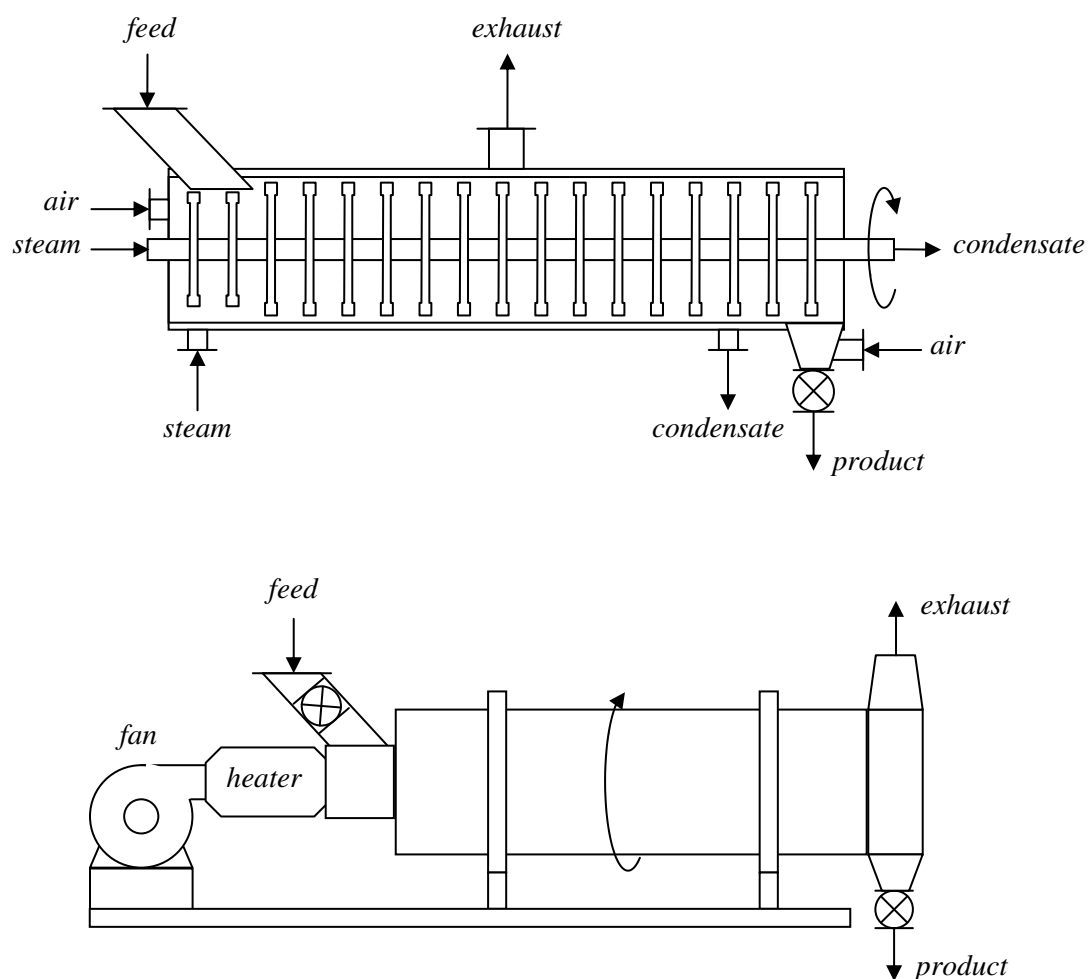


Figure 2.2: Schematic cross-section of a conductive rotary dryer (top), and outline of a convective rotary dryer (bottom).

2.2.3 Fluidised bed dryers

Fluidised bed dryers use hot gas streams for multiple purposes. The streams take care of drying, agitation and movement of the wet material all in once. Hence, there is only very little contact between the product and moving parts, which makes the equipment more wear resistant than e.g. rotary dryers.

The fluidised bed process starts with cutting the wet cake into small pieces. These pieces are fed from the top into a chamber, while gas flows in from the bottom through a perforated distributor plate. The flow velocity is high enough to suspend the weight of the particles in a free-floating state (Fig.2.3). The extensive contact between the gas and the so-created fluidised bed makes the moisture evaporate relatively fast. Batch type fluidised bed dryers are emptied after completion of the process. Continuous feed and discharge operations are based on the natural behaviour of moist particles to migrate towards the bottom of the fluidised bed. Particles containing less water are located in the upper parts, whereas dry pieces automatically move over a partition and leave the chamber.

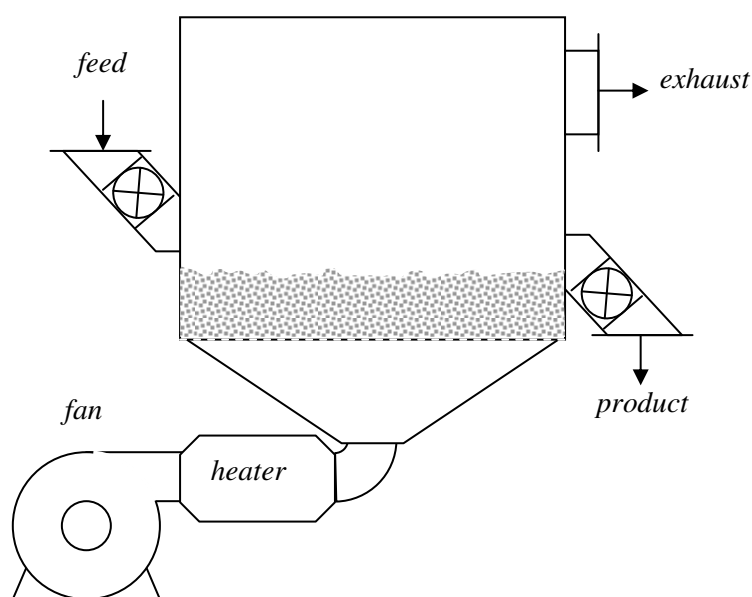


Figure 2.3: Central part of a fluidised bed dryer.

The gas velocities that are necessary to maintain the fluid state require high-pressure fans. They result in significant electric power consumptions and gas consumptions per unit of evaporated water. The agent can just be fresh air mixed with combustion gases, but as the gases leave the drying chamber at relatively high temperature, recycling of the medium is profitable.

From a performance point of view, the optimum design suspends the whole bed with the lowest volumetric airflow at the highest possible velocity [Cook, 1991]. This yields the best ratio between exposure time and gas consumption. Unfortunately, it also restricts the wet feed to more isometric particles with narrower particle size distributions: coarse and sticky fractions may block the bed; large quantities of fines

travelling along with the gas may overload the dust collection equipment; and mixtures containing needles, flakes and fibres are more difficult to suspend in a controlled way.

There are several ways to improve the performance of the fluid bed dryer for unfavourable feeds. A pulsed gas stream for instance, can increase the stability of the bed for larger distributions of sizes and shapes. Another option is to add a large number of inert particles to the bed, which is known to make fluidised bed drying applicable to slurries. The most adopted remedy, however, is the use of a vibrating gas distributor. A vibrating grid increases motion in the lower regions of the bed, preventing incrustation of sticky material. As the fluid bed itself also acts as a back mixer, recirculation of dried product can become redundant; even when processing mechanically dewatered sewage cake [Erens, 2003].

2.2.4 Highly turbulent bed dryers

Like fluid bed dryers, pneumatic dryers do not rely on mechanical means to transport and agitate the product. The core of a standard machine is depicted in Fig.2.4. It consists of a 10m to 30m long vertical pipe, through which a finely dispersed feed moves under influence of a hot gas stream. Meanwhile, the moisture evaporates from the small particles within seconds. (Hence, the technique is called flash drying.) Cyclones recover the dried product from the medium. The gas is further cleaned using bag filters, wet scrubbers etc., which, because of the large volumes, forms a crucial part of the process.

The main advantage of pneumatic dryers is their simple construction. They are compact, can reach large capacities, contain few moving parts and require little maintenance. Standard configurations may just comprise a burner, fan, pipe and cyclone to take care of the basic features. Vortex dryers do not even use separate cyclones. They apply a rotating duct to dry, convey and centrifuge the product from the stream, all within a single unit.

The working principle of pneumatic dryers does make the installation susceptible to overloading and requires proper feed conditioning. The feed must be homogeneous to begin with; the particles should be smaller than a few millimetres in order to avoid unacceptably high gas velocities and large pressure drops in the pipe; and the material to be dried should not tend to agglomerate. Back mixing of dried product is self-evident even for simple feeds.

The short residence time of the material has consequences as well. It requires high evaporation rates - thus hot gases - and makes the medium leave at relatively high temperature. This is why it is worth to recycle the exhaust stream and reduce losses to the atmosphere. Moreover, the short residence time restricts the process mainly to removal of free moisture. Some designs aim to solve this problem and make the technology suitable for more thorough drying. They let the product move in loops or sweep back and forth in opposing gas streams [Kudra, 2002].

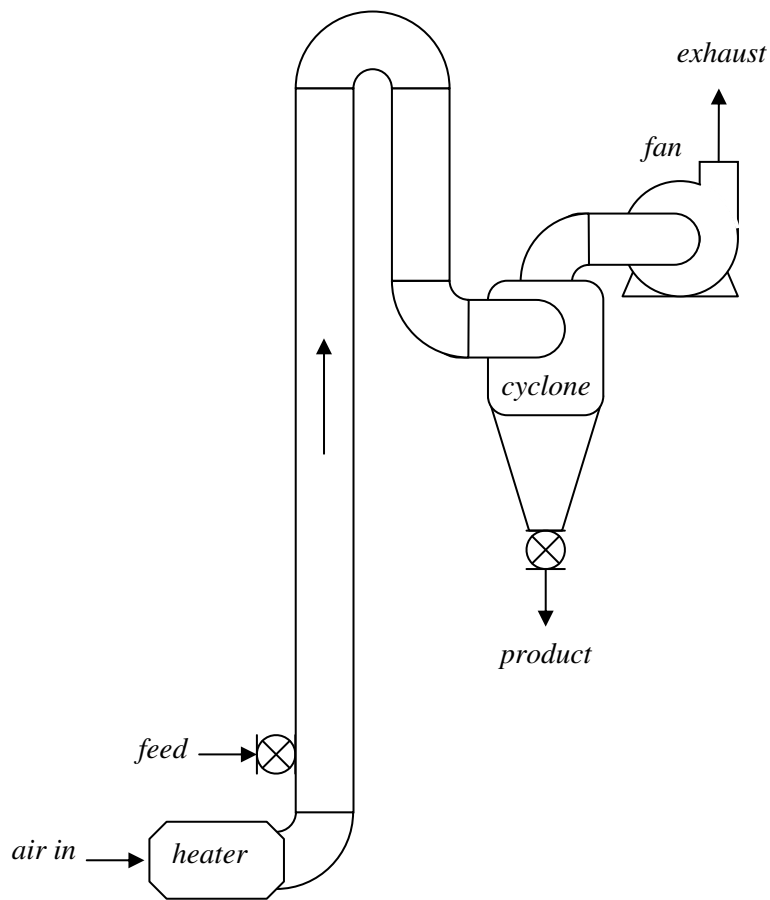


Figure 2.4: Schematic illustration of a flash dryer.

2.3 Reuse of released vapour

The significant energy consumptions in thermal drying are to provide the enthalpy of evaporation, to raise the temperature of the product, and to compensate losses from the system. Modern installations aim to reduce the latter, using indirect heating and proper thermal insulation. Because of many evolutionary improvements, the losses have been making smaller and smaller contributions to total energy consumptions.

This, however, does not mean that the thermal efficiencies are approaching their upper limit. The theoretically minimum amount of heat to be supplied is not the enthalpy of evaporation. At least three techniques allow additional savings by recovery of part of the latent heat of the vapour. These techniques are the process of vapour recompression, heat pump drying, and multiple stage drying.

2.3.1 Mechanical vapour recompression

Drying in a pure water vapour atmosphere offers the best opportunity to recover the heat of evaporation. The absence of noncondensables as in vacuum contact drying or superheated steam drying allows the exhaust stream to condense relatively easy in a heat exchanger or to be brought to higher temperatures by means of thermal or mechanical vapour recompression. The latter method requires a steam compressor. The instrument sucks away released vapour and compresses it to higher temperature. The gas subsequently moves to a heat exchanger, where it condenses and indirectly heats the cake.

Mechanical vapour recompression can virtually substitute all external heat supplies. The work to run the compressor is less than the thermal energy recovered, even if the losses in the power plant and the electricity network (on average a little below 60%) are accounted. The method can therefore be cost-effective. The savings, however, are partly annihilated by the costs of electric energy and suitable compressors. Besides, the method has some limitations to overcome. The performance of heat exchangers, for instance, is susceptible to surface contamination as well as the presence of noncondensables that leak in or form during the drying treatment. Moreover, the process gas usually contains dust particles and fibres, which are detrimental to mechanical pumps. Finally, practical compression ratios (and thus the temperature rise of the vapour) are too low for conventional drying techniques, requiring large heat transferring surfaces and ditto investments.

The commercial Econdry technology addresses some of these problems. It makes the vapour condense in a separate heat exchanger to evaporate clean water in a secondary circuit. The pure vapour in this secondary circuit is subsequently compressed and used as heating agent in the dryer. Although this approach is successful, it does not solve the problems of surface fouling. Regular cleaning of heat exchanging surfaces remains necessary.

2.3.2 Heat pump drying

Condensation of vapour from an air stream is a much slower process than condensation of pure vapour. If the stream is unsaturated, the process also takes place at lower temperatures, i.e. below the dew point. This makes it less interesting to recover the heat of evaporation. Nevertheless, a method exists that may be applied to the gas mixtures involved in convective drying. This method is based on a heat pump system. It comprises an evaporator, compressor, condenser and expansion valve, through which a secondary working fluid runs in a closed circuit. The evaporator allows thermal contact between the working fluid and the low-grade process stream. Initially, the process stream just transfers heat by cooling, but as the relative humidity reaches the point of saturation, condensation of vapour takes place. Meanwhile, the working fluid changes its state from liquid to gas by absorbing the released energy. The next step involves compression of the working fluid to higher temperature. The condenser, which is a second heat exchanger, transfers the heat to the drying medium. The working cycle closes by letting the fluid expand to low pressure.

The theoretically maximum efficiency of a heat pump cycle is the Carnot coefficient of performance, which is the condenser temperature in Kelvin divided by the temperature gain. Although practical efficiencies are typically 40-50% of the maximum values [Palandre, 2003], the amount of heat recovered can still be much larger than the electric energy consumed. However, just as the method of mechanical vapour recompression, heat pump drying requires clean heat-transferring surfaces.

The overall performance drops significantly when the surfaces collect dirt. Hence, the value of the recovered heat is not always worth the cost of electric energy and additional equipment.

2.3.3 Multiple stage drying

Another method that exploits the latent heat of the vapour is based on a system of multiple drying chambers. A patent entitled “Method or installation for the treatment of sludge or press cake” describes the idea. According to the patent, a facility comprising a pre-heater, pre-dryer and dryer can dewater sludge at only half the usual energy needs [Klein, 1992]. The three units operate at different temperature levels in a pure vapour atmosphere. Released vapour from the dryer is carried to the pre-dryer, where it condenses in a double-walled conveyor hull and indirectly heats the cake. Evaporated moisture from the pre-dryer is transferred to the wet feed in the pre-heater.

The principle of multiple stage drying is the same. It just involves more units operating at a wider range of temperatures. Heat is supplied via conductive surfaces, as the product passes the stages and gradually dries. The stages interconnect in such a way that released vapour from one stage serves as a heat source for the next. Vapour that leaves the last stage dissipates or is compressed to higher temperature.

The technique of multiple stage evaporation is well-known and widely used for the production of fresh water out of salt water. Despite the fact that it was patented for slurries as well [Willis, 1987], [Greig, 1993], it is not easily applicable to non-pumpable substances. Transportation between the physically isolated stages and heat transfer to the partially dried solids form the primary obstacles.

The Carver-Greenfield concept is an exception that does use multiple evaporators to dry sludge to low moisture contents. The key to its success is a special form of preconditioning. The feed remains in a fluid state after grinding and addition of large quantities of water immiscible oil with high boiling temperature [Holcombe, 1992]. The oil-solids-water slurry is pumped through the evaporators. Removal of the carrier oil after the last evaporation stage occurs by mechanical and thermal means. It is claimed to yield a product with solids contents up to 98% and less than 1% oil [Huybrechts, 2001]. Along these lines, the Carver-Greenfield concept can reach high thermal efficiencies. Its limitations are obvious though: it involves considerably more equipment and corresponding processing costs.

2.4 Energy consumption and cost comparison

The ideal technique dries at the highest capacity and the lowest possible energy consumption. These two optimums do not go together well: higher drying rates generally lead to lower efficiencies, and vice versa. The most economical situation is therefore a compromise between throughput and efficiency, yielding an acceptable balance between fixed expenses and running costs per unit of evaporated water.

The chosen configuration determines the maximum performance of an installation. As discussed in previous paragraphs, the heat transfer mode influences the fraction of primary energy that is effectively applied. Agitation of the bed increases drying capacities, be it at the cost of extra investments and electric power consumptions,

while recovery of latent heat can bring significant savings. This paragraph gives an overview of the energy consumptions and exploitation costs of various drying techniques.

2.4.1 Energy consumption

Despite efforts put into the reduction of heat losses and recovery of heat, thermal drying will always be significantly more energy intensive than mechanical solid-liquid separation. (Industrial techniques spend 100 to 500 times more energy per unit of removed water.) It therefore pays to maximize the potential of mechanical dewatering first, before the cake is further dehydrated by thermal means.

Energy is primarily required as heat. An additional contribution is needed for the feed and discharge operations, transport of material through the installation, and fans or pumps. Table 2.1 gives an overview of thermal, electric, and total primary energy consumptions of various techniques. All dryers use significantly more energy per kilogram of evaporated water than the heat of evaporation (2.3 MJ/kg).

Table 2.1: Consumptions of heat, electric energy, and air for various dryer types, according to TNO/KEA-Consult, 1998. The total primary energy consumptions in the last column were derived using conversion factors of 0.95 for gas, 0.8 for steam and 0.4 for electric power.

<i>dryer type</i>	<i>thermal [kJ/kg H₂O]</i>	<i>electric [kJ/kg H₂O]</i>	<i>air [kg/kg H₂O]</i>	<i>primary [kJ/kg H₂O]</i>
<i>pipe</i>	3200/steam	35	0.25	4100
<i>disc</i>	3000/steam	35	0.25	3850
<i>paddle</i>	4400/steam	38	5-10	5600
<i>drum</i>	3500/gas	20	3-10	4200
<i>fluidized bed</i>	4800-6500/gas	100-200	25	5300-7350
<i>fluidized bed/indirect</i>	3000/steam	100	closed syst.	4000
<i>Carver-Greenfield</i>	930/steam	720	vacuum	2950
<i>Carver-Greenfield/MVR</i>	465/steam	900	vacuum	2850
<i>Vortex</i>			40	5600
<i>flash [Strumillo, 1986]</i>	4000-4600/gas	70-180		4400-5300
<i>opposing jet [Turoskiy, 2006]</i>	3400-3800/gas	180-290		4050-4750

To compare the various techniques, the primary energy consumptions in Table 2.1 were calculated using fixed conversion factors. The factors are based on the assumption that heat is produced on-site from combustion of fossil fuels, and that electricity is generated in a conventional power plant. Actual energy consumptions may differ from the presented numbers if an installation happens to operate in unusual settings. The dryer may benefit from energy sources with different conversion factors, such as biogas from digestion reactors, flue gas from incinerators, steam from combined heat and power plants, rest heat from cement kilns, or geothermal energy. If the dried product has a high calorific value, (autonomous) thermal disposal in an integrated dryer-burner system is also an option. Whether or not an alternative energy source is applicable depends on its local availability, the net savings, and the type of dryer. Low-quality rest heat, for instance, cannot always be beneficiated, as most dryers operate at too high temperatures.

Just as rest heat from thermal processes can be profitable for a neighbouring drying installation, the by-products from the drying process are sometimes used in other operations down the line or for less related purposes. Some dryers produce excess steam for advanced applications. The exhaust gases of others are treated using wet scrubbers, after which the water serves to warm buildings or to speed up the digestion process in wastewater treatment. Whenever exhaust heat is reused in a sensible way, its value should in all fairness be settled in the economy calculations of the system.

2.4.2 Operating costs

The operating cost per unit of evaporated water is usually the decisive argument for a choice between suitable candidate dryers. When considering the total lifetime of a facility, energy consumptions are responsible for the largest part of the expenses, followed by initial capital investments, interest, maintenance and labour. Other important issues are the drying capacity, life expectancy of the equipment, number of operating hours per year, space requirements, value of by products, and product quality.

Fig.2.5 displays the total costs and the contributions to these costs for various commercial installations. The graph is based on data from manufacturers and a few assumptions. It does not take into account the necessary infrastructure at the site and the peripherals, labour and space that may be required for preconditioning of the feed, aftercare of dried product, and cleaning of exhaust gases. This is important to note since the cleaning equipment alone can contribute up to 30-35% of the total cost of the drying facility [Flaga]. If a building is to be erected for the dryer, another 20-35% over the installed costs needs to be factored in [Mujumdar, 2006].

It is wise to consider implementation of improved dryer designs if the savings can compensate the additional investments. In practice, real system innovations do not show themselves that often though. It is partly because of technical limitations that come along with the advantages of new techniques, but it is also due to conservatism originating from the size of the investments and length of payback periods. New, potentially superior concepts bring more risk, whereas the choice for a long proven technique is a safer one.

The common trend is thus to reduce costs by optimising conventional drying methods. Heat losses are reduced wherever possible, energy is recovered in all possible ways, and processes are modelled, fine-tuned and controlled using advanced systems [Yliniemi, 1999]. This trend, however, will not always continue. Eventually, it will come to a point where system optimisation reaches its limits set by physical, practical or economical constraints. If these asymptotic limits are approached closely enough, the ongoing strive for higher efficiencies and capacities will boost the development and implementation of revolutionary new concepts.

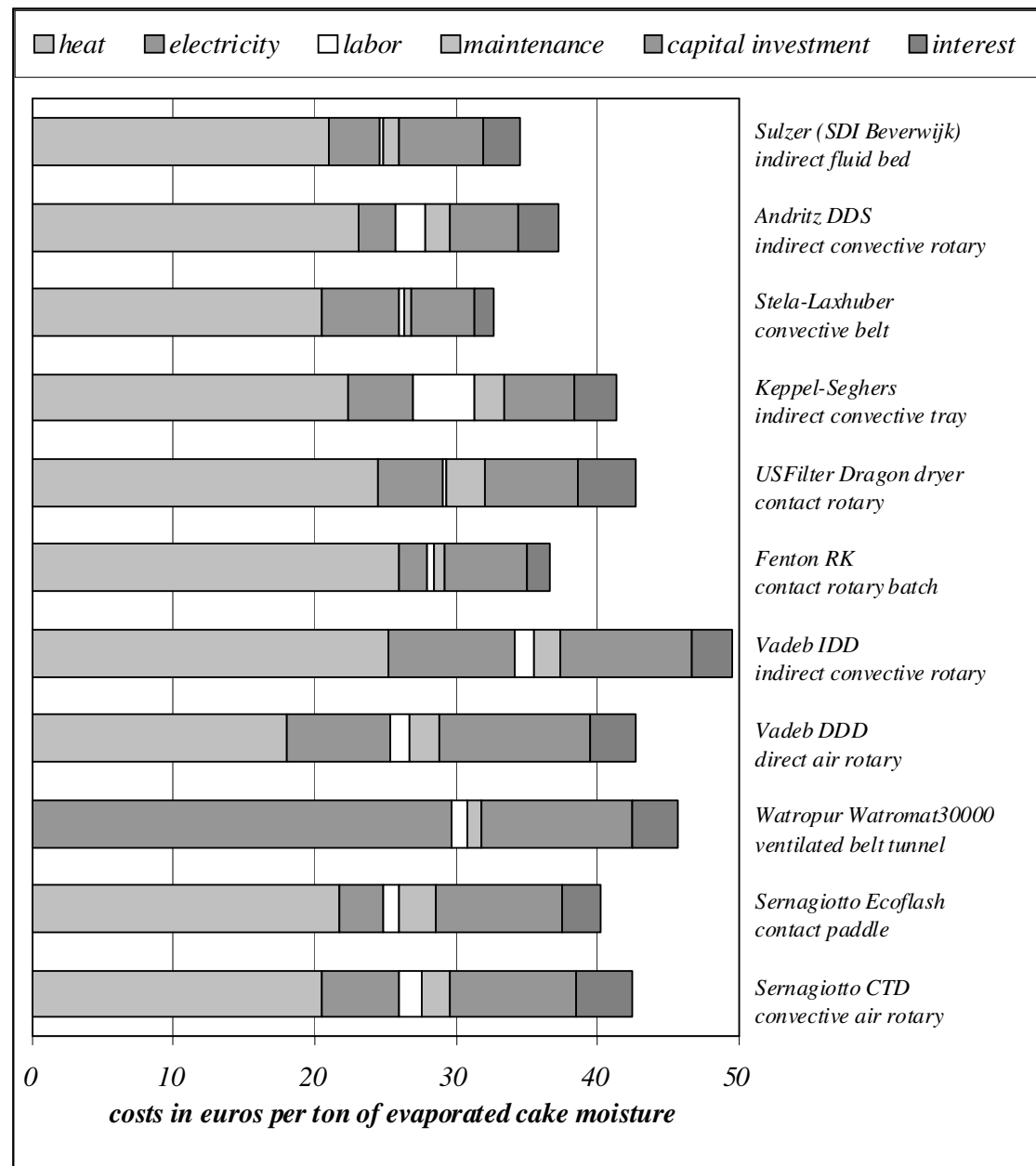


Figure 2.5: Cost comparison between dryers [Zonneveld, 2005], [Kiezwetter, 2002]. The costs are based on a gas price of € 5,60 per GJ, electricity price of € 0,09 per kWh, labour cost of € 50.000,- per person per year, and 7500 operating hours per year.

Chapter 3: The Delta dryer concept

The technology review in Chapter 2 showed that most dryers use significantly more energy than is actually needed to evaporate water from the cake. In addition, they often use air to carry away the water vapour, which necessitates considerable effort to separate the volatiles, odours and fine dust that move along as well. Finally, several dryers are susceptible to the consistency of the paste, which depends on the type of cake and strongly varies during the drying process. The installations in question require extensive preconditioning operations in order to achieve proper characteristics of the feed.

The shortcomings make the processes less cost effective, which is an issue due to the large expenditures involved. Novel concepts appear now and then, trying to overcome the disadvantages. Unfortunately, these usually result in technical difficulties, additional operations and unacceptable investments as well.

The Delta dryer concept was developed as a low-cost, energy efficient alternative to existing techniques. This chapter starts with a description of the concept, showing its elementary features and intended operation. The thermodynamics laying at the basis of the concept are those of a process called 'multiple stage flash evaporation'. The theoretical working principle of this process is analysed for the most optimum situation as well as for less-ideal variants. A 'very simple model' subsequently elaborates on the gained insights, predicting the maximum efficiency of the drying concept as a function of the number of flash stages and a few operating variables. The outcome of the model is not realistic. However, it reveals the theoretical upper limit of performance and is therefore a good reference. This chapter concludes with a description of the principal configuration of the Delta dryer in terms of process units.

3.1 Description of the Delta dryer concept

3.1.1 Starting points for a novel drying technology

The Delta dryer concept combines two principles: the use of hot, metal balls to be mixed with the wet feed and acting as the drying medium; and the use of a vacuum atmosphere to increase evaporation rates and to carry away the water vapour. This unique combination brings several potential advantages over existing techniques.

- In mixing with the balls, the cake is exposed to large heat transferring surfaces. The excellent thermal conductivity of the metal balls results in large drying capacities within installations that are relatively compact in terms of dimensions.
- The use of mobile heat exchanging surfaces allows different stages of the process (mixing, drying, and recovery of dried product) to take place in separate parts of the installation. It also enables additional operations such as a cleaning step.
- As large quantities of balls are mixed with the cake, the consistency of the mixture depends merely on the balls and does not change much when the product transfers from a 'pumpable' material, through a sticky phase, into a solid. Neither is the consistency of the mixture influenced by the composition of the cake.
- The almost pure water vapour atmosphere avoids the problem of exhaust air cleaning simply because large air volumes are absent. Small amounts of noncondensables that may be present can be carried away in a concentrated form and treated with little effort.
- The vapour atmosphere gives the opportunity to recycle part of the heat of evaporation by making the vapour condense on a heat exchanger. In this specific situation, the evaporated product moisture serves to preheat clean balls that are mixed with fresh feed at a later stage.
- The vacuum and the large heat-transferring areas result in higher evaporation rates at lower drying temperatures compared to existing techniques. This makes it possible to exploit low-grade heat from other thermal processes or from alternative sources.
- A low discharge temperature means less aftercare for cake with high organic contents, as the dried product leaves the installation with only little smell and without the risk of spontaneous combustion. Hazardous situations like fires and dust explosions are no issue inside the installation.

3.1.2 Elementary features of the Delta dryer

The Delta drying process is based on the natural phenomenon of the free water vapour pressure to rise with temperature. If, in absence of air or other non-condensing gasses, two wet compartments at different temperatures are atmospherically connected, water will evaporate from the hot region, flow to the colder side and condense. This mode of heat and moisture transfer is known to be highly effective. In case of free water, evaporation continues until the hot compartment has dried or the system has reached a uniform temperature.

The thermodynamics of the Delta dryer process are comparable to those involved in multiple stage drying (see Section 2.3.3). Likewise, a series of contacts is arranged between evaporation and condensation chambers at different temperature levels. Vapour from the drying stages is carried away to heat exchangers operating at lower temperature. However, whereas the released heat of evaporation is normally reused directly by letting the product dry at the other side of the heat exchanger, the novel

concept makes use of a different strategy. It brings a compartmentalised, hot mixture of balls and cake repetitively in contact with counter currently moving colder sets of clean balls that yet do not actively take part in the drying process. While passing through the stages, the balls-sludge mixture gradually dries up and cools off, whereas the temperature of the clean return balls rises. The finished product is removed after the last drying stage. An extra heating and dewatering step renders the preheated balls suitable to be mixed with fresh feed and to participate in the drying process themselves.

The flow sheet of Fig.3.1 shows the elementary process steps of the Delta dryer. The individual steps are explained below.

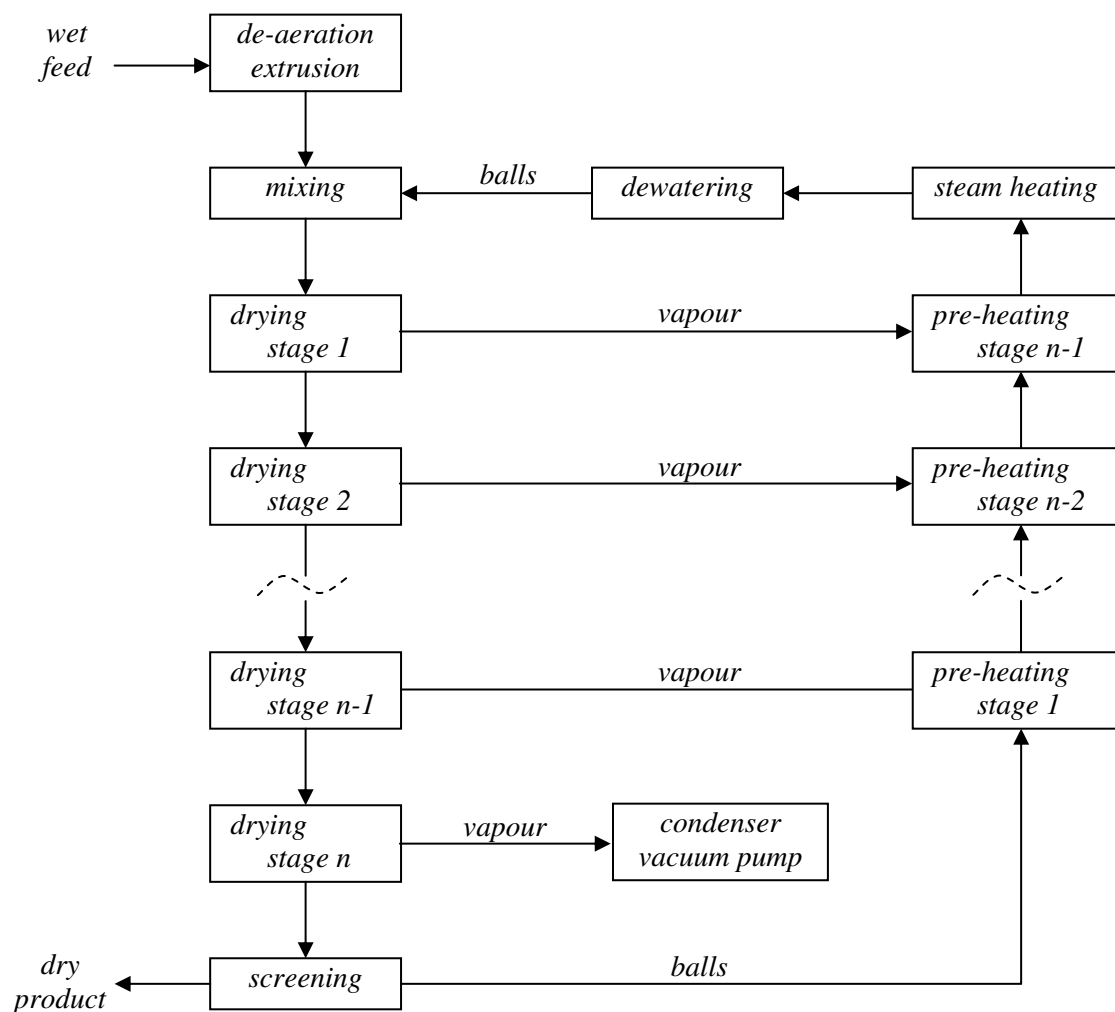


Figure 3.1: Flow sheet of the Delta dryer concept, showing the elementary process steps [Van Poppel, 1996].

de-aeration and extrusion

The presence of air inside the installation can be detrimental to the process. Air does not condense onto the clean return balls. It will build up at the points of lowest pressure and level down pressure differences between interconnected flash compartments. The installation is sealed to prevent significant amounts of air from entering the process. The wet feed is injected by an extruder or pump, which degasses the cake and acts as a seal between the air atmosphere of the feeding operation and the water vapour atmosphere of the process.

mixing

The wet cake is mixed with hot balls. It spreads out over the surface of the balls, forming a thin, preferably homogeneous layer.

drying

The balls-cake mixture is compartmentalized into a train of packages that stepwise dry up and cool off as they move along the evaporation stages. Vapour pressure gradients carry released vapour from the drying stages to colder packages of clean return balls, and to a condenser.

condenser and vacuum pump

The condenser is the point of lowest pressure, removing the moisture that is still present near the end of the drying process. It is also the point with the highest air to water vapour ratio, since it eventually collects all air that has leaked into the system. A connection to a vacuum pump prevents excessive build-up of air inside the installation.

screening

The dried mixture moves to a screening stage where the product separates from the balls. The temperature in this part of the installation is relatively low, and so is the pressure. A sluice system controls the discharge of dried material from the low-pressure atmosphere, while preventing air leaks from outside. An additional cleaning step may be necessary in order to remove residual solids from the balls.

reheating of balls

Released product steam condenses onto the clean balls, heating them up. Pressure increases as the temperature of the balls rises.

steam heating

The vapour from the drying process will not bring the balls entirely back to their starting temperature. Hence, an extra heating step is required.

removal of condensate

The condensate that forms in the reheating stages must be removed before the balls are mixed again with fresh feed.

3.2 The principle of multiple stage flash evaporation

The Delta drying process is characterised by the various evaporation and condensation stages, and the high drying rates resulting from the vacuum and large heat transferring areas. The term ‘multiple stage flash evaporation’, abbreviated MSFE, covers these features. The general working principle of an MSFE-process is analysed to investigate the thermal potential of the Delta dryer concept. This is done for the optimum situation as well as for less-ideal variants.

3.2.1 The ideal MSFE-process

The ideal MSFE-process exhibits no losses to the environment. All compartments have the same, constant heat capacity, and each flash brings two compartments of a stage in perfect thermodynamic equilibrium. This implies that after each flash, the temperature on both sides of a stage is the same. The common temperature forms the starting point for the cooling section of the next stage and the heating section of the previous stage. As a result, a linear temperature profile along the stages should establish itself automatically. This profile can be reconstructed from any two known temperatures and the number of stages, similar to the example in Fig.3.2.

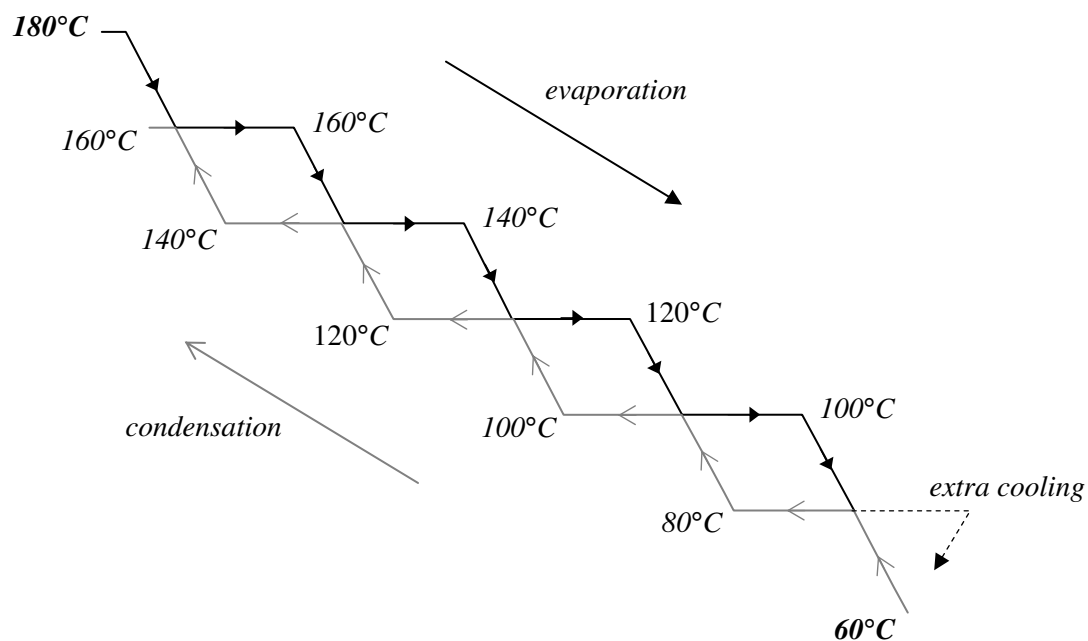


Figure 3.2: Equilibrium temperature profile over 5 stages of an ideal multiple stage flash evaporation process. Heat is lost in the extra cooling step only.

The MSFE-process loses some heat to an ‘extra cooling stage’ or heat sink. This creates the temperature gradient (the driving force for heat and moisture transport) in the flash stages, keeping the process going. In the ideal situation, the heat loss in the extra cooling stage equals the amount of heat that is necessary to raise the final temperature of the last heating section to the starting temperature of the first cooling section. Hence, the theoretically minimum energy consumption is a simple function of the total evaporation rate Φ_{vapour} , the heat of evaporation H_{evap} , and the number of heat

recovery stages n (Equation 3.1). The theoretically minimum energy consumption is nil for an infinitely large number of flash stages.

$$P_{\min} = \frac{\Phi_{\text{vapour}} \cdot H_{\text{evap}}}{n + 1} \quad [3.1]$$

3.2.2 Less-ideal MSFE-processes

The conditions in a real installation will never meet the strict demands of the process described in Section 3.2.1. The flash stages may exhibit different heat capacities as a result of fluctuations in the balls throughput or due to changing contents of the sections. A condensation section with lower heat capacity is undesirable as it will take away less vapour from the evaporation sections, and bring less heat back to the start of the process. Vapour that does not condense onto the balls disappears to the heat sink. It yields non-linear temperature profiles as in Fig.3.3.

The evaporation sections in a real Delta dryer comprise mixtures of steel balls and wet cake, whereas the condensation sections contain balls and condensate. The heat capacity of the solids is small, so that the heat capacities of the stages differ less than a few percent as long as the balls throughput is constant. The real issue is whether the enthalpy of the hot condensate is reused in the end, or eventually lost anyway.

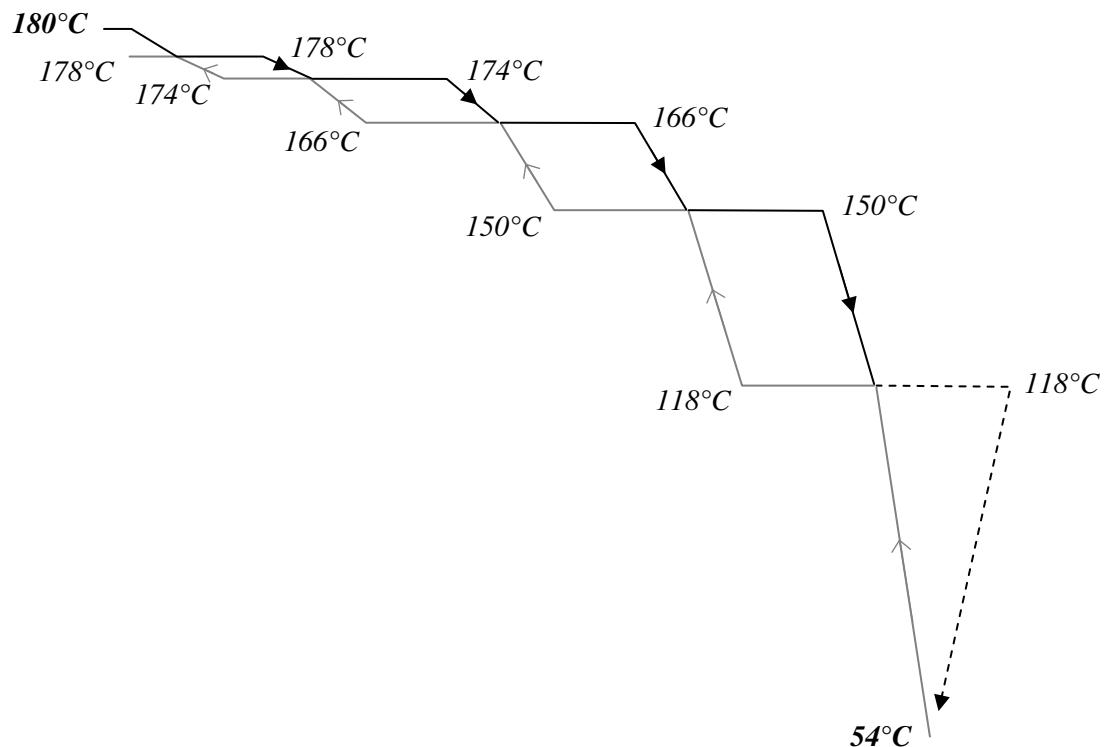


Figure 3.3: Equilibrium temperature profile of an MSFE-process. The heat capacities of the evaporation sections are twice as large as the heat capacities of the condensation sections. Although the final temperature after the last heating stage closely approaches the starting temperature of the first cooling section, most heat actually disappears to the heat sink.

A second distinction between ideal and real installations is that the latter lacks perfect insulation. It is simply impossible to eliminate all losses from the system and to prevent thermal contacts between adjacent flash stages. The temperature profile in Fig.3.4 belongs to an MSFE-process in which small amounts of heat escape. The heat losses are equally large for all 5 stages, equivalent to a temperature drop of 2 degrees. The end temperature after the last condensation step (150°C) is obviously 10 degrees lower than the end temperature of an identical but perfectly insulated process (Fig.3.2). More surprisingly, the heat losses also influence the shape of the temperature profile. The temperature gradients are initially large, but become smaller and smaller towards the final stages. This can adversely affect the efficiency of the last couple of stages, as together with the temperature gradients, the driving forces for heat and mass transfer decrease.

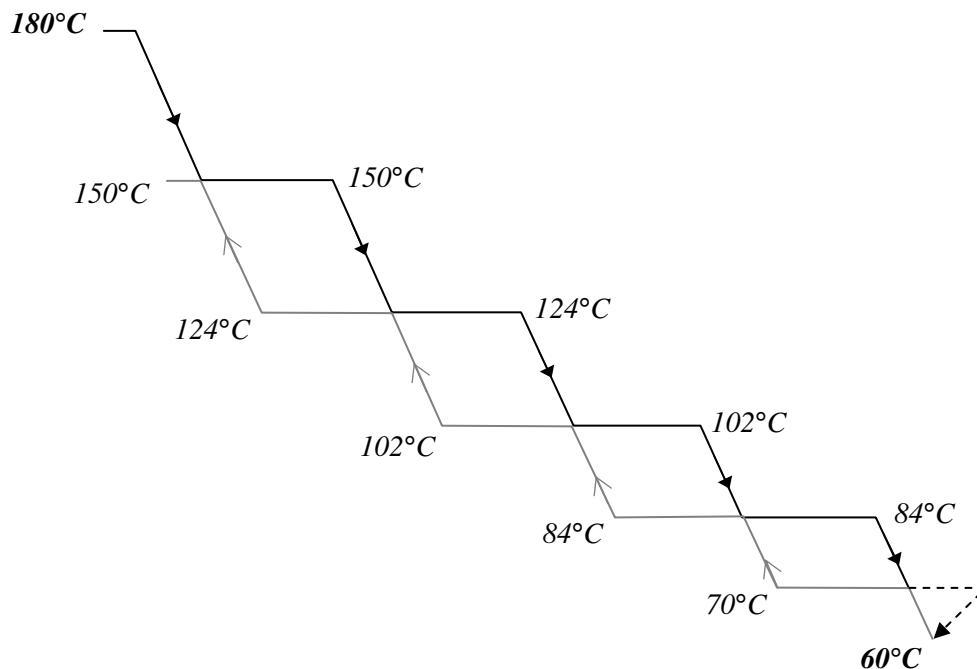


Figure 3.4: Equilibrium temperature profile over 5 stages of a multiple stage flash evaporation process. A fixed amount of heat is lost in every stage.

It was assumed until now that each flash brings two sections of a stage in perfect thermodynamic equilibrium, which is unrealistic as well. The two facing sections will not entirely reach a common temperature in practise, because of the limited duration of the flash. Certain pressure and temperature gradients must be present in order to generate acceptably large heat and mass flow rates. Fig.3.5 schematically displays what happens to the temperature profile in case thermodynamic equilibrium does not establish in two of six flash stages: the end temperature after the last heating section is definitely lower than in a situation with six ideal stages, and higher than in a situation with only four stages.

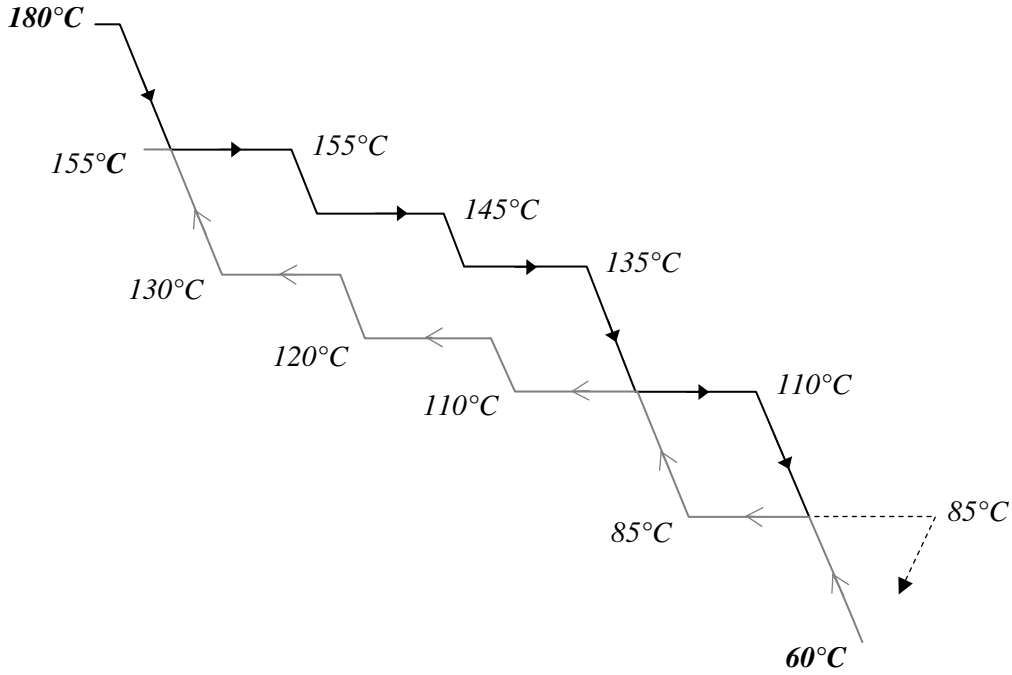


Figure 3.5: Temperature profile in case no thermodynamic equilibrium establishes between the sections of two flash stages. No heat is lost to the environment.

The foregoing general analysis clarifies that the total efficiency of the MSFE-process depends on the performance of all individual stages, and that efficiency losses in one stage cannot fully be compensated down the route. The power consumption of a ‘less-ideal’ MSFE-process follows from the equation below. eff_i is the efficiency of stage i , defined as the quotient of actual heat transfer and optimum heat transfer between two facing sections. P_{loss} represents the additional losses from the system.

$$P'_{min} = \frac{\Phi_{vapour} \cdot H_{evap}}{1 + \sum_{i=1}^n eff_i} + P_{loss} \quad [3.2]$$

It may be tempting to adapt the design and operational parameters of a Delta dryer installation with the aim to steer the MSFE-process into a certain direction; for instance if some stages happen to exhibit higher evaporation rates than others. In such cases, one may easily decide to reduce the contributions of the first stages with the intention to raise the efficiencies of next ones. Considering the foregoing however, this will not give positive results, but will reduce the overall efficiency of the drying process instead.

It is more worthwhile to find a balance between the drying temperature and drying capacity in order to minimise heat losses. Design changes that reduce vapour leaks between adjacent flash stages are beneficial as well. Such vapour leaks influence the efficiency of the drying process in a similar way as obstructions to vapour flows within the stages: the flash process is frustrated in either case. A certain amount of heat is kept from flashing in the right stage, and shifts towards a next stage.

3.3 A very simple model of the Delta dryer

Equation 3.1 formulated a relationship between the theoretically minimum energy consumption and the number of vapour bridges. For n evaporation stages, the minimum energy consumption per kg of evaporated water is $1/(n+1)$ times the heat of evaporation. Fig.3.6 extends the theory for a situation in which heat flows are present due to inputs and outputs of solids (m_s) and water (m_w). All sections still share the same heat capacity. No additional heat and mass flows exist between the process and the outside, or between flash stages.

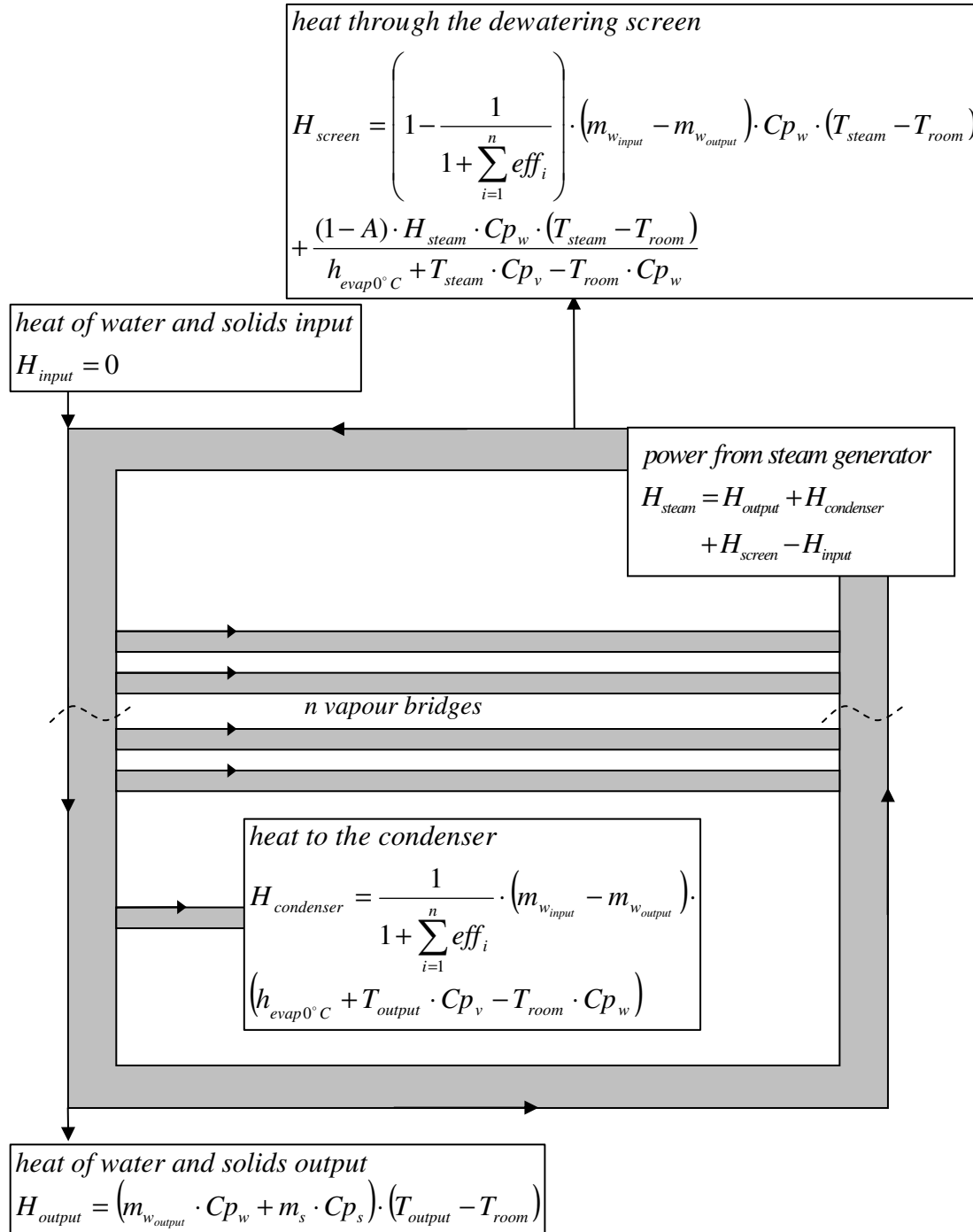


Figure 3.6: Process scheme of the very simple model, with T_{room} as reference temperature.

One of the equations in Fig.3.6 describes the heat loss through the dewatering screen. The equation contains a factor A , representing the fraction of condensed steam (from external supplies) to be recycled. A is nil if all heat through the dewatering screen is lost. The factor equals unity in case the hot condensate is fully reused.

Apart from the heat balance in Fig.3.6, a relationship exists between the output temperature of the dried product, the initial ball temperature (T_{steam}), the balls-cake ratio, and the amount of water that evaporates from the cake. Equation 3.3 presents this relationship.

$$T_{output} = \frac{m_{balls} \cdot Cp_{steel} \cdot T_{steam} + m_{w,input} \cdot Cp_w \cdot T_{room} + m_s \cdot Cp_s \cdot T_{room} - \Delta H}{m_{balls} \cdot Cp_{steel} + m_{w,output} \cdot Cp_w + m_s \cdot Cp_s} \quad [3.3]$$

with: $\Delta H = (m_{w,input} - m_{w,output}) \cdot (h_{evap0^\circ C} + Cp_v \cdot T_{output})$

Typical values for the parameters in Equation 3.3 are: $T_{steam}=105^\circ C$, $T_{room}=20^\circ C$, $m_{balls}=110kg$, $m_s=1kg$, $m_{w,input}=1kg$, $m_{w,output}=0.1kg$, $Cp_{steel}=500J/kgK$, $Cp_s=1000J/kgK$, $Cp_w=4180J/kgK$, $Cp_{vapour}=1900J/kgK$, and $h_{evap0^\circ C}=2.5MJ/kg$. These values yield an output temperature of $62^\circ C$. Substitution of values into the equations of Fig.3.6 leaves a single relationship between energy consumption and the number of vapour bridges. Fig.3.7 displays the results for $A=0$ and $A=1$.

According to the graph, the minimum energy consumption of an installation with 12 vapour bridges is 0.6MJ ($A=1$) to 0.7MJ ($A=0$) per kilogram of evaporated product moisture. This is three to four times higher than the theoretically minimum power consumption, which is represented by the lower curve in the graph. Although the used model is oversimplified and process efficiencies will still be lower in reality, the results give an idea of the potential of the concept in relation to those of existing drying techniques (Section 2.4.1).

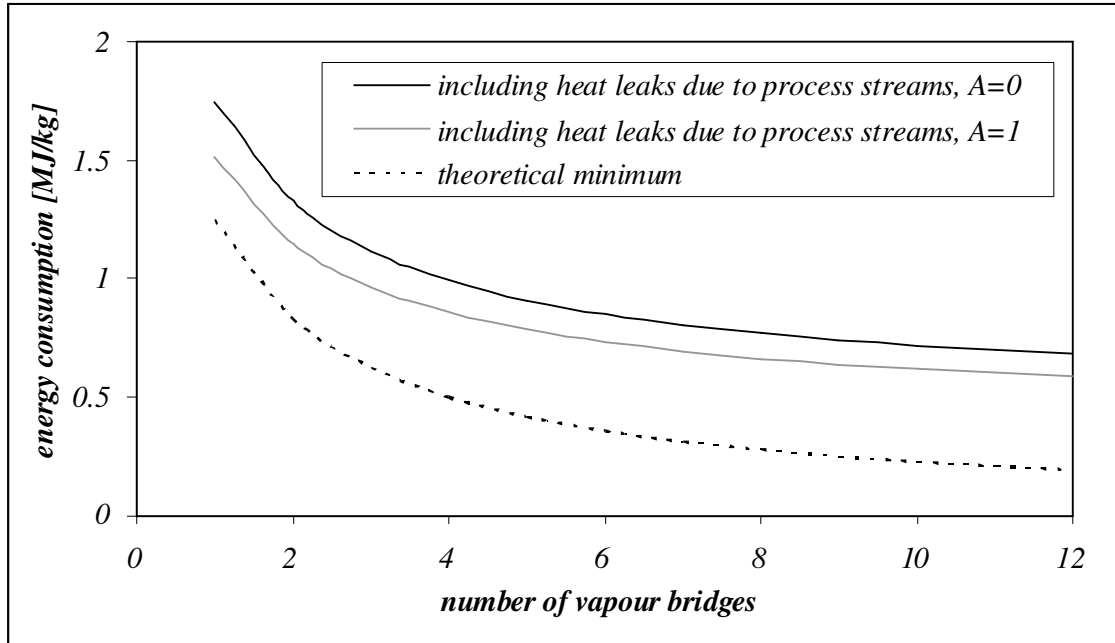


Figure 3.7: Energy consumptions per kg of evaporated water as function of the number of flash stages.

3.4 Implementation of hardware

3.4.1 Process units and description

The basic Delta dryer concept can be rendered into several design variants. Possible configurations principally differ in the way the ball bed is compartmentalised into a train of packages. This can be accomplished either by separating the packages using physical barriers, or by making use of the resistance of the packed bed itself. For reasons of simplicity, the latter option was preferred.

Fig.3.8 visualises the process units that were chosen to take care of the essential features. The installation comprises two parallel screw conveyors interconnected by means of vapour bridges, furthermore a steam vessel, sluice systems, and an elevator. The balls stream runs through these units in a closed circuit. Feeding equipment, a condenser, a vacuum pump and other peripherals complete the apparatus.

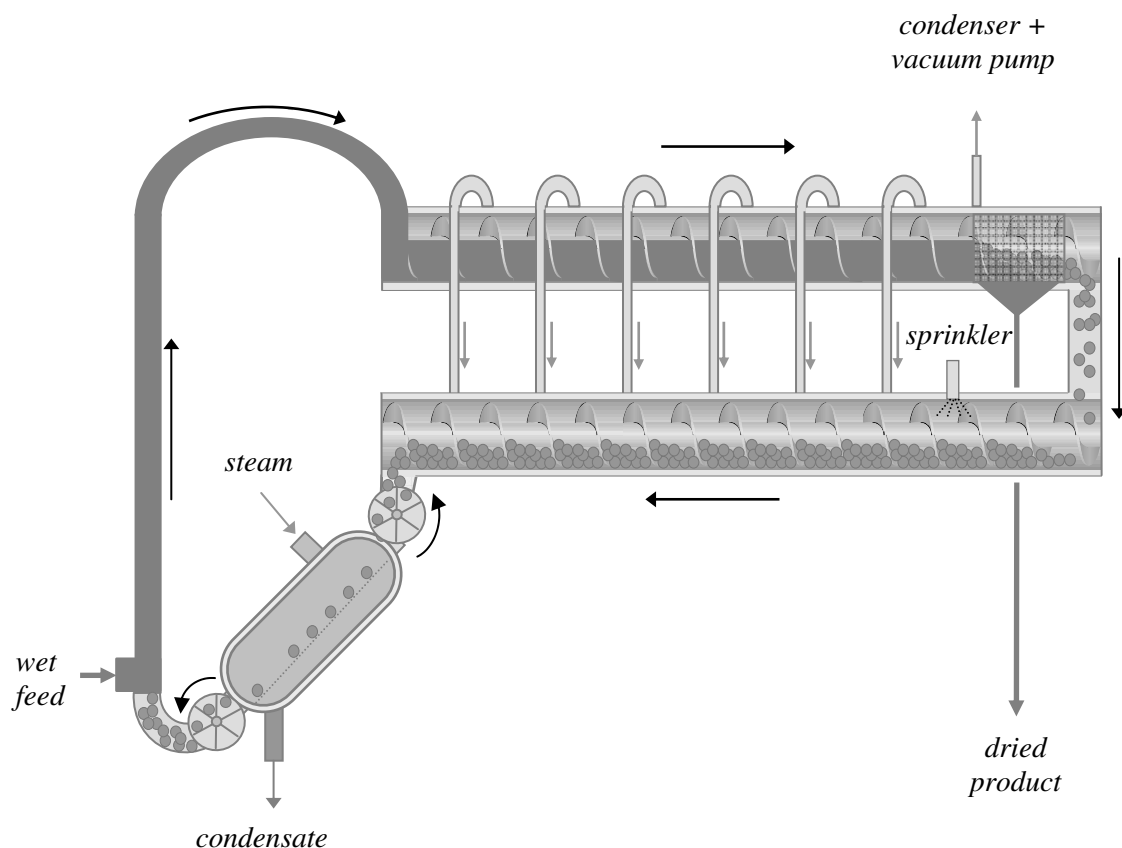


Figure 3.8: Drawing of the Delta dryer, showing a closed circuit of metal balls running through the process units.

The balls heat up inside the steam vessel to a starting temperature of typically 105°C. It occurs by means of steam injection. The elevated pressure involved is the reason why two sluices are located at the entrance and exit of the vessel. The sluices shield the internal atmosphere from the rest of the installation. The vessel has another function as well. It houses a sloping dewatering screen, which is essential to get rid of the condensate. As the balls revolve down the screen, their rotational velocity reduces the adhered water film to an acceptable thickness.

After passing the lower sluice, the hot balls enter the elevator and meet wet feed that is pressed into the installation by a clay extruder or pump. The elevator accounts for the vertical transport to the upper screw conveyors where the actual drying process takes place. Extensive mixing of the balls and the cake spreads the material over the large heat transferring areas. It requires some form of agitation, generated in this same elevator, in the first part of the screw conveyor, or in an additional appliance.

Inside the upper screw conveyor, the balls-cake mixture occupies at least half the volume of the pitches, forming alternating packages of solids and vapour along the spiral void. A high water vapour pressure causes the product moisture to evaporate into steam. The steam is carried away via vapour bridges as the mixture proceeds through the drying stages. As a result, the cake gradually dries up and cools off during its transport.

The last part of the screw conveyor connects to a spray condenser and vacuum pump. The spray condenser functions as a heat sink, removing remaining moisture from the product. The vacuum pump takes away the air that has entered the installation, as well as small amounts of volatiles that may release from the cake. At this point, the mixture reaches a temperature between 50°C and 60°C. The dried cake is separated from the balls by a screen, collected, and discharged via a sluice system to big bags or the like.

The balls now run into the second screw conveyor and move in the opposite direction parallel to the first conveyor. A sprinkler installation may be present to wash off the dirt. The balls then meet the other ends of the vapour bridges and repetitively get in contact with the vapour atmospheres belonging to the warmer packages of balls and cake in the upper conveyor. The temperature differences between facing sections of both conveyors induce vapour pressure gradients, allowing evaporated product moisture to flow through the bridges and condense at the colder side. As a result, the return balls are reheated step by step. At the end of the cycle, the preheated balls leave the second conveyor, pass through a sluice, and enter the steam vessel again.

3.4.2 Diameter of the balls

The heat content of the steel balls is to provide the heat of evaporation of the cake moisture to be removed. Given that the desired temperature range at which the MSFE-process takes place (100°C-50°C) is not widely variable, the mass ratio between balls and evaporating cake moisture is fixed around 100:1. The size of the balls, however, is not necessarily fixed. It is an important parameter in the various process steps of the Delta dryer.

A smaller ball diameter is advantageous for multiple reasons. First of all, it results in larger heat transferring surfaces per kg of steel, so that the cake is exposed in thinner layers. (The average layer thickness is reversely proportional to the ball diameter, provided that the cake spreads out equally uniform.) The heat transfer rate through the layers enhances, thus improving the kinetics of the evaporation process.

Secondly, the internal heat resistance of a ball drops proportionally with its diameter. This does not significantly influence the evaporation kinetics, since the internal heat resistance is very small compared to the heat resistance of the cake layer. However, its effect may be noticeable, letting a smaller ball heat up faster inside the steam vessel.

Finally, a smaller ball diameter is more favourable as it increases the resistance of the ball bed in the screw conveyors. The vapour leaks along the axial direction of the screws will be smaller, so that more heat can be recovered.

The aspects mentioned are clarified in the next chapters, which are dedicated to the evaporation kinetics (Paragraph 4.3), the heating process inside the steam vessel

(Paragraph 6.1), and the vapour leaks through the ball bed in the screw conveyors (Chapter 5 and Section 6.2.2). The ball diameter, which frequently shows up in the equations, is itself not a subject of investigation. This is because a minimum diameter exists below which the balls are too difficult to handle: transport becomes problematic, the required mesh size of the screens becomes too small, and the degree of recovery of dried product becomes too low. For practical reasons, the ball size of choice was set at 12mm.

Chapter 4: Physical properties of sludge cakes

The process steps of the Delta dryer make specific demands upon the characteristics of the cake.

- The feed must be able to spread out over the balls and adhere to the metal surface.
- The material should allow rapid transfer of heat and moisture, resulting in evaporation rates high enough to sustain a pressure gradient along axial direction of the screw conveyors.
- The equilibrium vapour pressure above the cake must approximate the isothermal free vapour pressure above liquid water in order to make efficient recovery of heat possible.
- The dried product should break loose from the metal surface to be recovered from the balls stream.

This chapter focuses on the most important physical properties of sludge cakes. Samples from various sources are analysed to check whether their characteristics are favourable or unfavourable for the Delta dryer concept. The sludge types of investigation use to be regarded as waste streams, which is why no properties are guaranteed and why very few data are available from literature. The samples include:

- Sludge from drinking water purification, dewatered by a belt filter press.
- Wet coal floatation concentrate from the Augusta Victoria mine in Germany, dewatered by a centrifuge and filter press.
- Residue from dredged material after sand separation and mechanical dewatering.
- Sludge from wet MSWI bottom ash processing, produced in a pilot-plant in Amsterdam, January 2004 [Muchova, 2004].
- Wet cake from the municipal sewage sludge dehydration plant (SDI) in Beverwijk, the Netherlands.

The properties of these samples are discussed here, starting with the consistency of wet fines as a function of moisture content. The second paragraph concentrates on the hygroscopic behaviour of the cake, which arises from the ability of the solids fraction to bind water. The third paragraph describes the drying process at microscopic scale, and pays attention to the kinetics of evaporation at sudden depressurisation. The chapter concludes with the kinetics of liberation of the dried material during the screening step.

4.1 Rheology of sludge cakes

4.1.1 Deformability

Because of their composition and microstructure, moist fines exhibit different physical states depending on their moisture content. These states are important for the Delta drying operation. They influence the mixing behaviour with the balls and the ease of separation at the end of the drying process. Insufficient mixing results in inefficient transfer of heat, reducing the kinetics of the evaporation process. Poor separation increases energy consumptions per kilogram of recovered product. Since the balls and cake mix at mass ratios of 100:1 to 100:5 (depending on the moisture content), the rheology of the mixture does not depend much on the properties of the cake. Hence, the consistency of the cake is less relevant for its transport through the drying stages, as long as the material holds to the balls.

The varying consistency of moist fines is a popular subject of investigation in the ceramic industry and in the field of geotechnical engineering. Four zones are usually distinguished for soils. They are presented in the graph of Fig.4.1. Although the transitions between the zones are gradual, more or less arbitrary boundaries are defined. These are the so-called Atterberg limits [Atterberg, 1911].

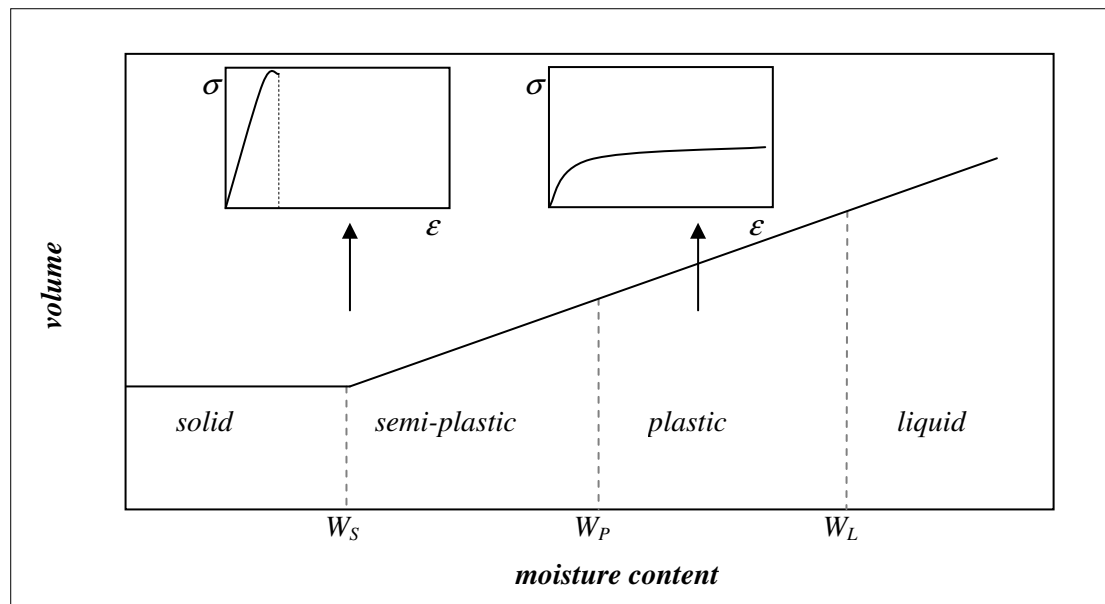


Figure 4.1: Schematic presentation of the volume of clayey soil as function of moisture content, showing four areas of different physical states.

A soil sample that belongs at the left side of the graph contains little water. Its pores are mainly filled with gas. This results in a brittle structure that can develop considerable strength as result of cohesive forces between the solid particles.

As the moisture content of the sample increases, additional liquid will replace the gas phase. Initially, this does not have any effect on the sample's volume. When more and more water is added, a point is reached at which the pores become saturated. This point has been defined as the shrinkage limit (W_S). From now on, the volume of the mixture increases proportionally with the volume of added water.

As the soil sample shifts to the right side along the horizontal axis, its mechanical properties change. Strength and hardness drop and maximum elongation increases. As the material passes the plastic limit (W_P), it attains the ability to deform continuously without rupture. This plastic zone extends towards the liquid limit (W_L). For moisture contents above the liquid limit, the sample behaves as a viscous fluid.

The geotechnical limits that were originally defined to characterise soils are also identifiable for the five sludge types introduced at the beginning of this chapter. The liquid and plastic limits of these samples were determined using test procedures prescribed by ASTM D4318 and BS 1377. Table 4.1 presents results of the tests.

Table 4.1: Plastic limit and liquid limit of various feed types. The quoted moisture levels are based on wet solids: $W=m_l/(m_s+m_l)$. No plastic state could be determined for coal, due to lack of coherence between particles.

<i>source</i>	W_P	W_L
<i>sewage cake</i>	0.51	0.82
<i>drinking water sludge</i>	0.46	0.56
<i>dredged material</i>	0.37	0.56
<i>MSWI bottom ash treatment</i>	0.36	0.46
<i>coal fines</i>	-	0.25
<i>silt</i>	0.17-0.20*	0.23-0.30*
<i>clay</i>	0.20-0.33*	0.30-0.70*

**[Budhu, 2000]*

4.1.2 Adherence to metal

Plastic behaviour alone does not guarantee optimum heat transfer between the balls and the cake. The feed should also hold to the metal surface. This is fortunately one of the characteristics of wet fines: within a range of moisture contents, they tend to adhere to metal. The adhesion forces increase when the material is being dried from above the liquid limit. This is because the lubricating effect of the abounded water gradually diminishes. After reaching a maximum, which for clays is said to be at moisture contents near the plastic limit [Thewes,1999], the adhesion forces decrease as the material becomes too dry to form liquid bridges.

The lowest moisture content at which the cake adheres to metal has been defined as the sticky limit [Terzaghi, 1948]. It depends on the composition of the cake (in particular the organic content), on the grain size distribution, and to some extent on the metal's surface roughness at micro-scale [Zimnik, 1990]. The cake cannot establish full contact with the balls at moisture contents below the sticky limit. The fraction that breaks away beyond this point will drop from one ball to another, move towards the lower regions of the ball bed, and exhibit low drying rates.

The sticky limit can be determined by gradually reducing the moisture content of a sample pat until a clean, dry spatula can be drawn over the surface without picking up any material [Head, 1982]. Table 4.2 presents results of such tests for the five investigated cake samples. The last column of the table contains typical moisture contents obtained after mechanical dewatering. These indicate the points of departure for the thermal drying process. Apparently, the substances are well deformable and sticky during the first period of this treatment step.

Table 4.2: Sticky limit of different feed types. The last column shows typical moisture contents obtained after mechanical dewatering.

source	sticky limit	$W_{\text{dewatering}}$
sewage cake	0.40	0.7-0.8
dredged material	0.40	0.6
drinking water sludge	0.38	0.5
MSWI bottom ash treatment	0.30	0.5*
coal fines	0.13	0.2-0.3**

* [Muchova, 2004]; ** [Le Roux, 2003]

Despite being well deformable and sticky, the wet cake will not fully spread out and uniformly distribute over the heat transferring surfaces. Intensive mixing activities yield average layer thicknesses that are two to three times larger than the theoretical optimum (see Fig.4.2). The actual layer thickness depends on the ball diameter and on the balls-cake ratio, which follows from the moisture content of the feed, the moisture content of the product, and the starting temperature of the balls. It is much larger for wet coal fines, for instance, than for sewage cake, simply because the coal does much better respond to the preceding dewatering step. The upper photographs in Fig.4.2 display mixtures of balls and mechanically dewatered cake samples. The mass ratios between balls and cake moisture are a constant 100:1. The lower photographs show the mixtures after being dried in an oven.

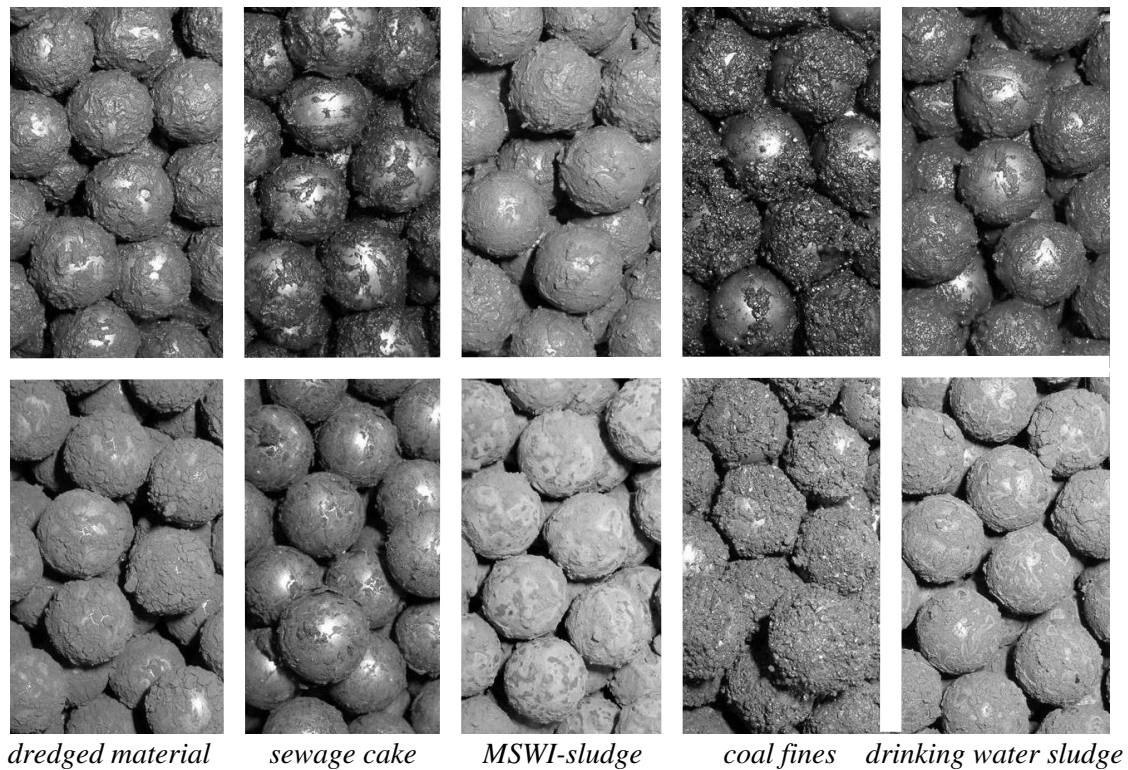


Figure 4.2: Mixtures of balls and various types of cake. The mass ratios between balls and initial cake moisture in the upper pictures are 100:1. The diameter of the balls is 12mm. The lower photographs show the mixtures in an oven-dry state.

4.2 Hygroscopic activity

4.2.1 Vapour pressure above bound water

Liquid placed in a sealed empty container will evaporate and raise the vapour pressure until the evaporation rate matches the condensation rate. The final pressure above a flat liquid surface is the free vapour pressure. It progressively rises with temperature.

The equilibrium vapour pressure above moist fines equals the free vapour pressure provided that free water is present. This is the case for mixtures of water and coarse-grained materials of which the surfaces of the grains do not interact with the water, as well as for finer fractions that are sufficiently wet. For the latter however, the final pressure reaches lower values if the fines dry up below critical moisture contents. This is due to the ability of the solid matrix to bind water. The heat of evaporation of bound water is higher than of free water, which is why the equilibrium shifts towards lower concentrations of molecules in the gas phase.

Sludge contains various types of bound water. These types are distinguished by the nature and intensity of their bonding to the solids [Kopp, 2000].

- Capillary water resides in cavities between the sludge particles or within a particle. It is held by adhesion forces between the liquid and the solid, and by cohesion forces in the liquid. Finer fractions form smaller cavities, yielding larger capillary forces.
- Surface water refers to layers of oriented molecules covering the particles. The interaction between the two phases arises from the (negative) surface charges of the fines. These charges, resulting from unsatisfied valency bonds, attract the positive side of dipole molecules. Although surface water is believed to extend from the solid surfaces only several molecular diameters into the liquid [Israelahvili, 1989], the total area of the particles is usually large enough to adsorb a considerable amount of water.
- Cellular water is enclosed in the cells of organic substances such as sewage sludge. In order to free this water, the cell walls must be destroyed.
- Water of hydration is chemically bound.

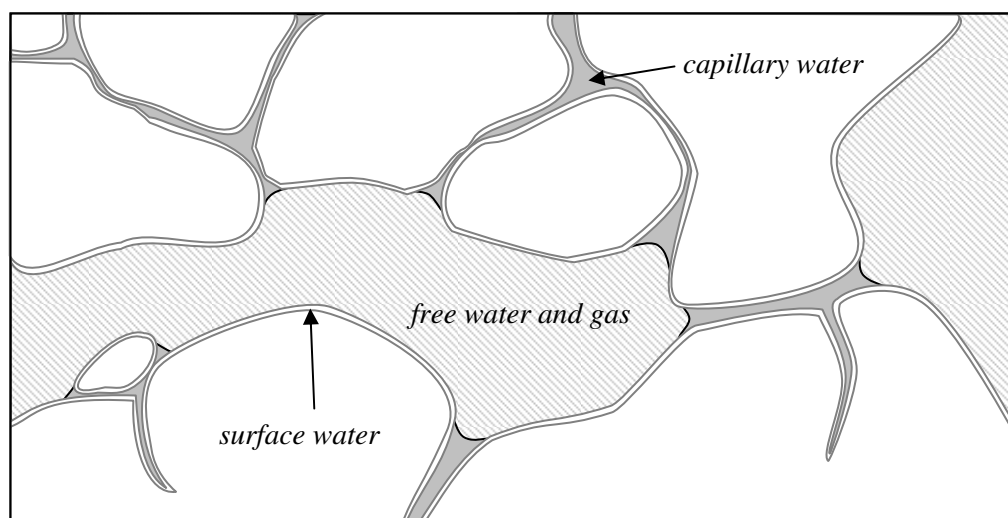


Figure 4.3: Schematic two-dimensional magnification of a sludge sample, showing the solid phase (white), macropores with free water and gas, capillary water (grey), and surface water.

Whereas free water can be removed relatively easily by means of a centrifuge or filter press, bound water is more difficult to extract. It requires thermal means. Especially for the Delta dryer, the hygroscopic nature of sludge cakes can adversely affect the process efficiency. It cuts down the pressure gradients within the MSFE-stages as it starts to act upon the vapour pressures at the evaporation side. Consequently, less vapour is transferred to the condensation side of the flash stages, so that less heat is recovered.

Since the solid sludge particles can be made of almost anything, the hygroscopic effect varies over the wide range of possible feeds to be dried. The specific surface area of the grains is the principle factor determining the amount of bound water; the geometry of the pore structure is responsible for the weaker bonds, referring to liquid captured by capillary forces; and the chemical composition of the solid matrix determines the intensity of the stronger binding forces.

The Polanyi adsorption potential theory suggests a relationship between the vapour pressure at which bound water releases, the free vapour pressure, the adsorption potential (ϵ) and the absolute temperature. This relationship is given by Equation 4.1 [Chiou, 2002]. A range of adsorption potentials normally exists within a single material due to its heterogeneous composition. When the substance is exposed to gradually decreasing vapour pressures, all free water filling the pore volumes and covering the surface will evaporate first; then capillary water in regions of low adsorption potentials; and finally the moisture in regions of progressively higher adsorption potentials. Chemically bound water will not release at the relatively low temperature of the drying process.

$$P = P_{free} \cdot e^{-\epsilon/RT} \quad [4.1]$$

4.2.2 Empirical determination of hygroscopic properties

Hygroscopic activities of different types of cake were determined using the laboratory set-up shown in Fig.4.4. This set-up comprises cylindrical containers, filled with a layer of wet material. The containers are pulled vacuum, sealed, and fully immersed in an oil bath at 80°C, while their internal pressure is registered. The final pressure is a function of the oil bath temperature, the moisture content, and the hygroscopic nature of the sample. It also depends on the amount of non-condensing gasses that are present from the start or that form during the experiment. Hence, the cylinders are quenched in cold water at the end of the experiment. The resulting pressure drops make it possible to calculate the partial pressures of the noncondensables as well as the true vapour pressures from the earlier stage. The samples are subsequently dried in an oven at 70°C to determine their moisture contents.

The described experimental procedure was repeated for several types of cake at a range of moisture contents.

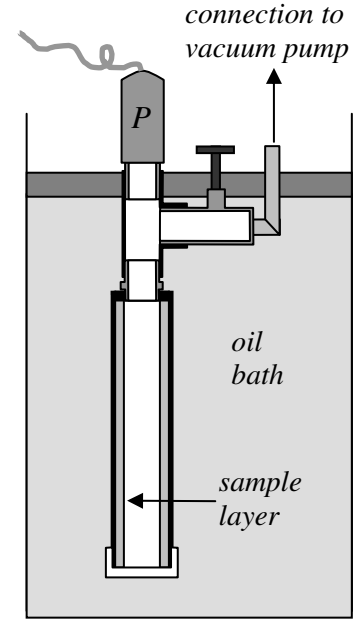


Figure 4.4: Equilibrium vapour pressures above moist fines were determined using cylindrical containers. These cylinders were filled with cake, pulled vacuum, and immersed in a hot oil bath.

The graph of Fig.4.5 presents experimental results together with literature data for silt, loam, clay, and peat. The vertical axis displays the relative humidity, i.e. the vapour pressure above the cake divided by the free vapour pressure. This makes it easy to compare data from slightly different oil bath temperatures.

The hygroscopic nature of a sample can in a simplified form be captured in a single parameter *Hygr*. This parameter is defined as the moisture content at which the relative humidity amounts to 90%. For moisture contents above *Hygr*, the vapour pressure (P_v) approximates the free vapour pressure ($P_{v,free}$). Below the threshold, the vapour pressure is assumed to be proportional to the moisture content. Equation 4.2 formulates this relationship.

$$\begin{aligned} \text{If } \frac{m_l}{m_s + m_l} > Hygr \text{ then } P_v &= P_{v,free}(T) \\ \text{If } \frac{m_l}{m_s + m_l} < Hygr \text{ then } P_v &= \frac{m_l}{m_s + m_l} \cdot \frac{1}{Hygr} \cdot P_{v,free}(T) \end{aligned} \quad [4.2]$$

The legend of the graph contains values for *Hygr*. Strictly speaking, these values are valid only for the temperature at which the tests were performed, which was around 80°C for practical reasons. More interesting than this temperature is the range between 50°C and 60°C, since the impact of the hygroscopic nature of the cake will be most apparent near the end of the drying process when the material reaches its lowest moisture content. The theory of Polanyi, however, suggests that the temperature dependence of *Hygr* is small. According to Equation 4.1, the relative humidity would decrease from 90% to 89% as a response to a temperature drop from 80°C to 50°C. The experimental results would thus shift only little towards higher moisture contents if the tests were repeated at lower temperatures.

The hygroscopic nature of most tested samples does not seem to be problematic when bearing in mind that the desired moisture content of the product is usually 10-15%. (This could be different for peat, which is known to be strongly hygroscopic.) A detailed prediction of consequences is not possible at this point though. It requires an advanced drying model such as the one described in the next chapter. Section 5.3.5 reveals effects of different values of $Hygr$ on the thermal performance of the Delta dryer.

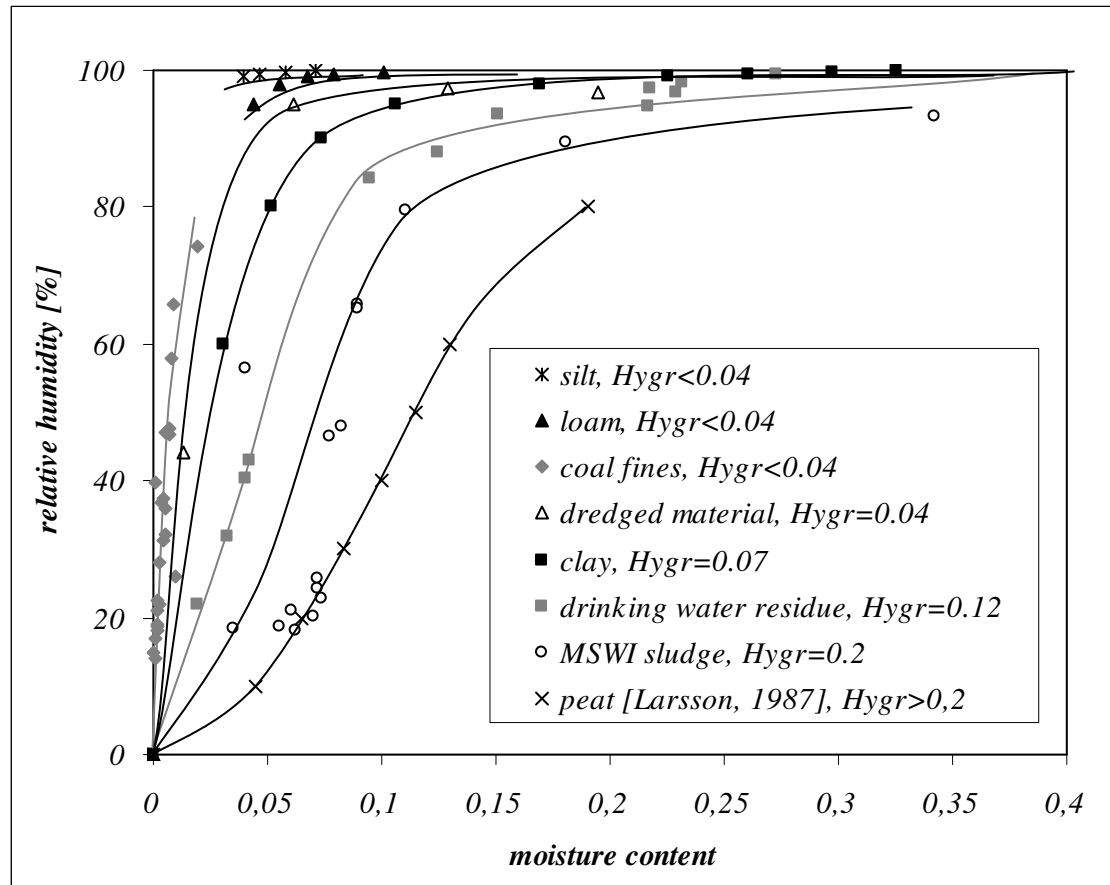


Figure 4.5: Relative humidity above samples as function of moisture content. Data for clay, loam and silt are from literature [Farouki, 1986]. The presented moisture contents are on a wet base.

4.3 Evaporation kinetics

4.3.1 Heat and mass transfer mechanisms

Thermal drying of moist fines relies on three physical transport phenomena: transfer of heat to the liquid, movement of water in the liquid or gas phase through the porous matrix, and removal of vapour from the surface. The driving forces are gradients in temperature, pressure and moisture content. They give rise to multiple mechanisms that are interacting in a complex way.

Numerous attempts have been made to predict the resultant of the interacting mechanisms for conventional drying of mechanically dewatered sludge cakes. The developed theories, however, are of limited use here. The conditions in the Delta dryer, characterised by large heat transferring surfaces, exceptionally thin cake layers, and a reduced pressure atmosphere, derive far from those in standard convective and contact dryers. They are comparable with the conditions in high-intensity contact drying of paper, in which the temperature in the paper sheet is raised in excess of the boiling point corresponding to the ambient pressure. Available knowledge on paper drying is applied to describe the Delta drying process at microscopic scale.

In analogy to high intensity paper drying, the elevated temperatures during the Delta drying process will make liquid evaporate not just from the surface but also in the bulk. An internal pressure build-up causes a convective outflow of vapour and liquid. This yields two distinct zones in the cake: a region close to the heat transferring surface in which pores are partly filled with water; and a saturated region that extends from the unsaturated area to the free surface. The boundary between the zones advances towards the exterior as drying proceeds. In addition to these zones, two regions will form in the course of the process: a dry region next to the ball-cake interface, which gradually expands at the cost of the unsaturated region; and an area close to the free surface, exhibiting reduced moisture levels as result of a receding drying front (see Fig.4.6).

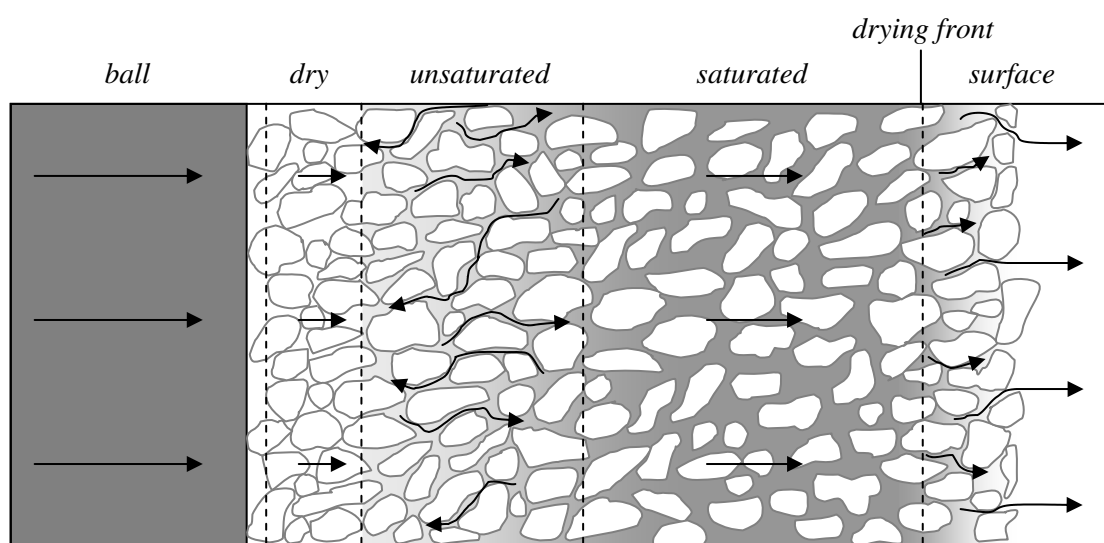


Figure 4.6: Schematic, magnified cross-section of a ball surface, thin contact layer and cake layer with dry, unsaturated, and saturated zones. The arrows indicate the directions of heat and mass transport.

Since the convective mass transfer mode is efficient, enough water is supplied from the bulk to the drying front. Removal of vapour from the free surface is neither an issue, as the absence of air allows it to disappear quickly from the site. The drying rate is controlled primarily by the heat transfer rate from the ball to the drying front.

The mechanisms that are held responsible for heat transfer within the cake are thermal conduction and a process of multiple distillation cycles. The first is the dominant mechanism in the saturated region and dry region. Thermal conductivities of wet fines vary widely with solids composition, microstructure and moisture content [Hillel, 1980]. They are highest when the cake is saturated, and drop significantly as the substance dries and the pores dehydrate, since liquid water is a much better conductor than vapour. The second heat transfer mechanism confines itself to the unsaturated region of the cake. It involves moisture evaporating from the hot side of the pores, raising the local vapour pressure. The vapour moves some distance against the temperature gradient through the pore canals. It then condenses and releases heat of evaporation. At the same time, capillary backflow of liquid takes place, allowing the process to reiterate as in a heat pipe [Ramaswamy, 1992].

As the contribution of the multiple distillation cycles to the heat transfer rate is known to be significant, the evaporation rate will be high as long as the wet region stretches out over the thickness of the cake layer. It may show an initial peak value if heat was stored in the cake during the preceding mixing operation. (The amount of stored heat depletes quickly though as the liquid flashes.) The evaporation rate can also increase after a moment if the cake was initially saturated, since the multiple distillation cycles are less prominent at high water contents [Moyne, 1992]. At some point in time however, the process is hampered by a poorly conductive dry area that expands from the hot surface. This is indicated by large temperature gradients in this area compared to those in saturated and unsaturated wet zones [Lindsay, 1992]. The dry region is believed to appear at an early stage in high intensity paper drying, and to control the evaporation rate from then on [Ramaswamy, 1992]. Whether this is the case in the Delta dryer is uncertain. It depends on the convective vapour flow rate in the unsaturated region in relation to the rate at which liquid moves in direction of the pressure gradient by capillary suction. Hence, the critical moisture content at which a dry zone develops is influenced by the internal pore structure and the driving forces.

A final stage with particularly low evaporation rates shows when most water has disappeared. There are three explanations for this. First of all, the heat of evaporation of bound water is larger than for free water, so that deep drying of hygroscopic material requires larger driving forces. Secondly, the thermal conductivity of the cake decreases steeply at low moisture contents due to poor thermal contact between solid particles [Ghildyal, 1987]. And thirdly, the heat resistance of the contact layer increases as part of the cake breaks loose from the heat transferring surface. The available drying time is limited, which is why the process eventually ends abruptly. If this were not the case, evaporation would go on until the equilibrium vapour pressure above the remaining fraction of bound water matches the vapour pressure of the surrounding atmosphere.

4.3.2 Determination of evaporation kinetics

It is difficult to predict the kinetics of evaporation based on heat and mass transfer mechanisms, because of the varying temperature gradients, pressure gradients, and moisture distributions that are affecting each other. An additional issue is that relevant properties of the cake, such as local permeability and thermal conductivity, are

unknown. They depend on the composition of the cake, are sensitive to the sizes, shapes and distributions of constituents, and vary with temperature [Kroes, 1999]. This makes them hard and labour-intensive to derive from simple tests.

An attractive alternative is to describe the resultant of all heat and mass transfer phenomena by a macroscopic model, in which an apparent property of the cake is defined incorporating all relevant parameters. Determination of this overall property requires a laboratory test rig that approaches the real situation, since the empirically derived data will be applicable within a certain bandwidth only. Fig.4.7 shows the set-up that was developed to simulate the conditions inside the Delta dryer.

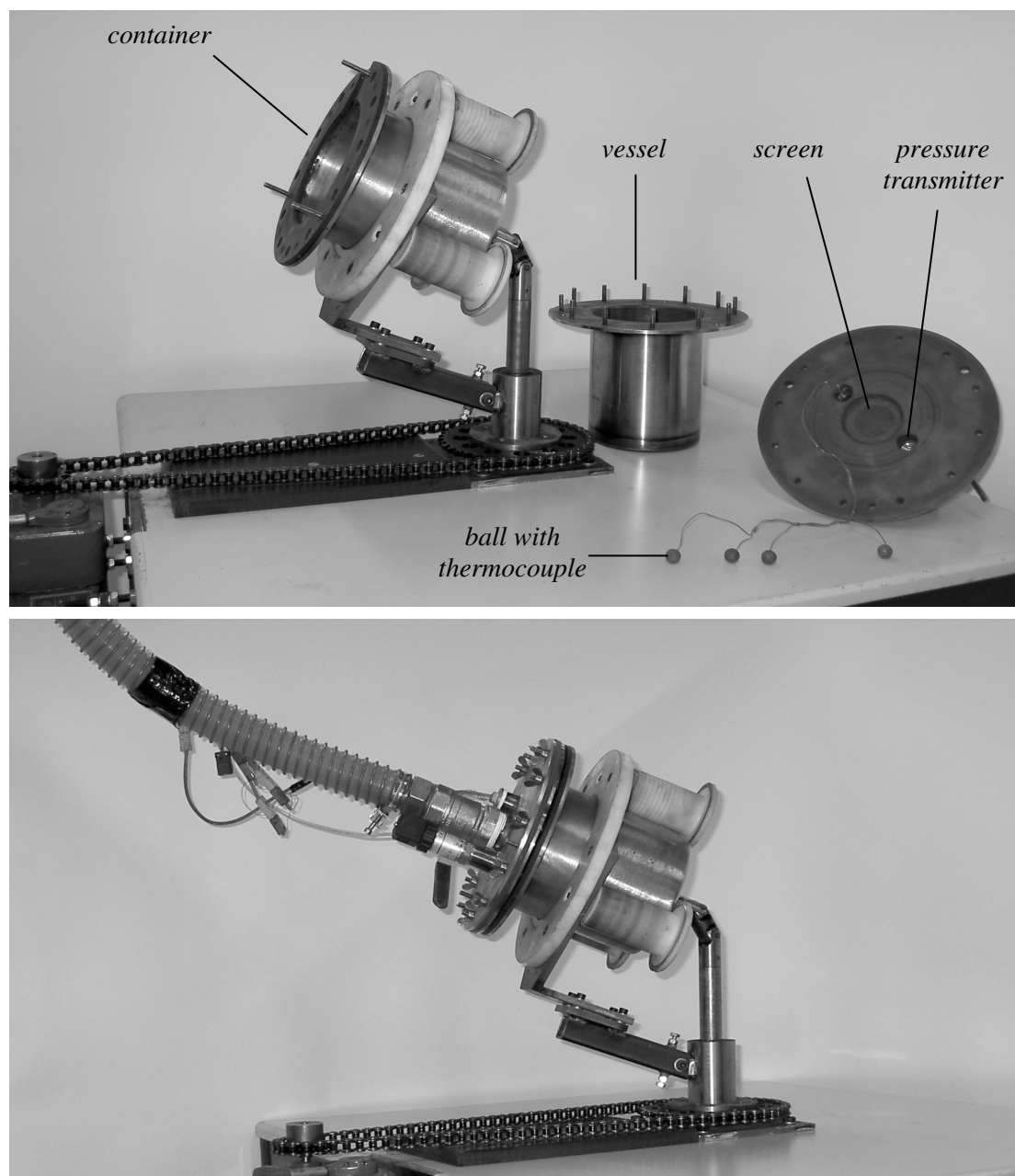


Figure 4.7: Laboratory test rig. The vessel is placed in a container that holds to the desk by means of an axis and universal joint. An electrically driven arm with rollers swings around, letting the vessel rotate in motion. An opening in the centre of the lid holds a small-meshed screen. It connects via a valve to a flexible vacuum hose.

The experimental procedure starts with the addition of weight quantities of cake and cold steel balls (diameter 12mm) to a small pressure vessel. The content is stirred in order to distribute the wet substance over the surface areas. The vessel is sealed and placed inside an oven where it heats up to uniform temperature. It is then positioned in an insulating container onto the test rig, connected to a vacuum pump by means of a flexible hose, and set in motion. The quasi-rotating motion of the vessel replicates the turbulence in a ball bed segment of the Delta dryer. The mechanism that imposes this action lets the balls run around as in a ball mill, while preventing the vacuum hose and data acquisition wires from twisting.

While the vacuum pump carries away the vapour, the hot mixture dries and cools off since there are no external supplies of heat. The steel vessel, which has not been insulated from the inside, takes active part in the process. Its wall thickness is $D_{ball}/6$, which eliminates most side effects by making the ratio between heat transferring surface area and heat capacity identical to that of a ball. The cooling rate, which is a measure for the evaporation rate, is registered using thin flexible thermocouples that stick into some of the balls. Together with a pressure transmitter, these thermocouples provide the primary output of the experiment.

4.3.3 Results from kinetic experiments

Fig.4.8 and Fig.4.9 present cooling profiles for dredging spoil and sewage cake. The samples were dried from moisture contents of 50% and 80% respectively. Initial mass ratios between steel and cake water were around 80:1. Heat and mass balances confirmed the temperature drop to correspond with the amount of evaporated water. Discrepancies between the curves of each graph are due to local fluctuations in the balls-cake mixture, making some balls cool off faster or further than others (see Fig.4.2). The average initial layer thickness was smaller for sewage cake than for dredging sludge because of the difference in initial moisture content. This distinction became more prominent during the process as the cake shrank. At the end of the experiment, more than 80% of the product was found to be liberated from the balls.

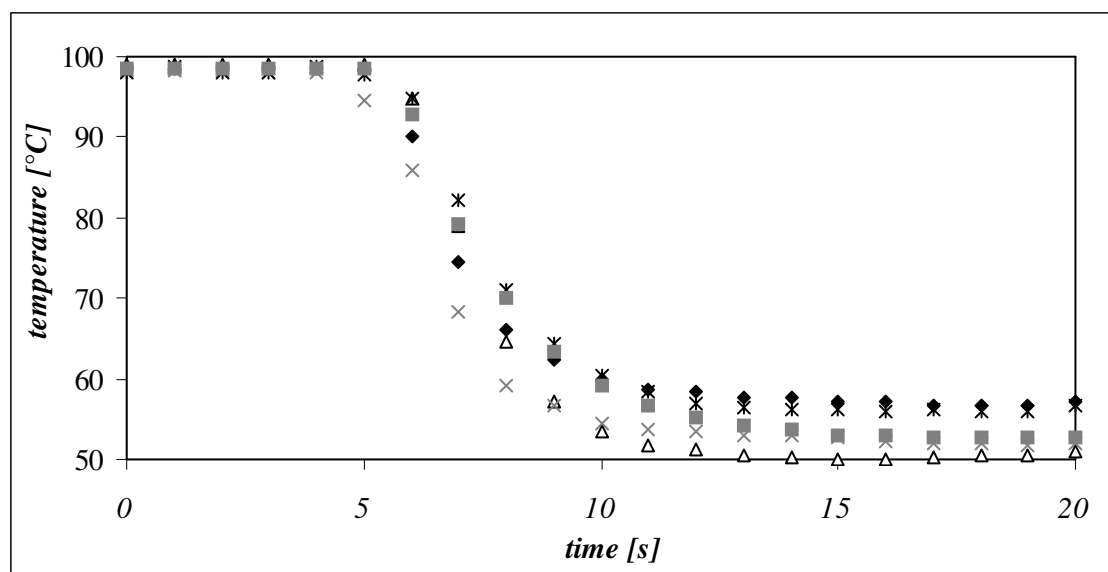


Figure 4.8: Cooling profiles for dredging cake. The average initial layer thickness was 0.3mm if the cake would have spread out over the total surface of the balls. In reality, it was ~0.5mm.

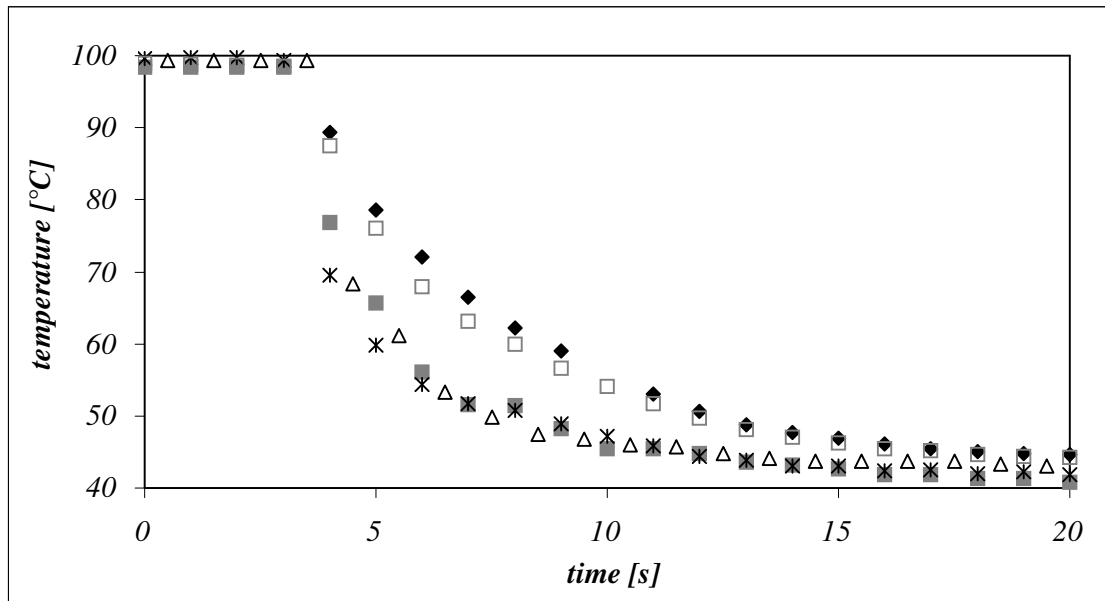


Figure 4.9: Cooling profiles for sewage cake. The average initial layer thickness was less than 0.25mm if the cake would have spread out over the entire surface of the balls. In reality, it was ~0.35mm.

The measured cooling profiles point towards high drying rates. These, however, are not fully representative for the Delta drying process. The driving forces for evaporation are smaller in a real installation than in the laboratory set-up. The available drying time of the Delta dryer is longer as well: 1 minute is believed acceptable.

The experimental results can be translated to realistic conditions by assuming the drying rate proportional to the difference between the temperature of the balls and the saturation temperature of the vapour atmosphere. This proportionality was reported for high intensity drying of paper, at least for ~0.25mm thick sheet at average evaporation rates in the range of 15-150kg/h·m² [Ahrens, 1986]. (For comparison: the average drying rate during the laboratory experiments was approximately 50kg/h·m².) Based on the description of prominent heat and mass transfer mechanisms in Section 4.3.1, the relationship between drying rate and temperature gradient is imagined as in Fig.4.10. The curves are for constant moisture contents, showing three regimes: A, B and C.

- A. The heat transfer rate in regime A relies on multiple distillation cycles in the unsaturated region and thermal conduction in the saturated and unsaturated regions of the cake (see Fig.4.6). The dry zone is negligibly small, as capillary backflow of liquid is fast enough to compensate convective vapour transports. Since the contribution of the multiple distillation cycles to the heat transfer rate is claimed to be proportional to the temperature gradient [De Vries, 1963], a similar linear relationship is expected between the drying rate and the temperature gradient.
- B. Higher temperature gradients result in higher convective vapour flow rates. However, they do not raise capillary flow rates. This implies that a threshold exists, above which a dry zone expands from the ball surface. The thermal conductivity in this dry zone is relatively poor, so that the overall heat transfer rate does not increase as much as before.

- C. A third regime may show at steep temperature gradients. This regime is characterised by the absence of an unsaturated zone. The saturated zone contains virtually all liquid. The width ratio between the dry zone and the saturated zone depends on the volume-average moisture content. Unlike in regime *B*, the widths are not significantly affected by the temperature gradient. Heat transfer is by thermal conduction through the dry zone and saturated zone, which explains the slope of this part of the curve in Fig.4.10.

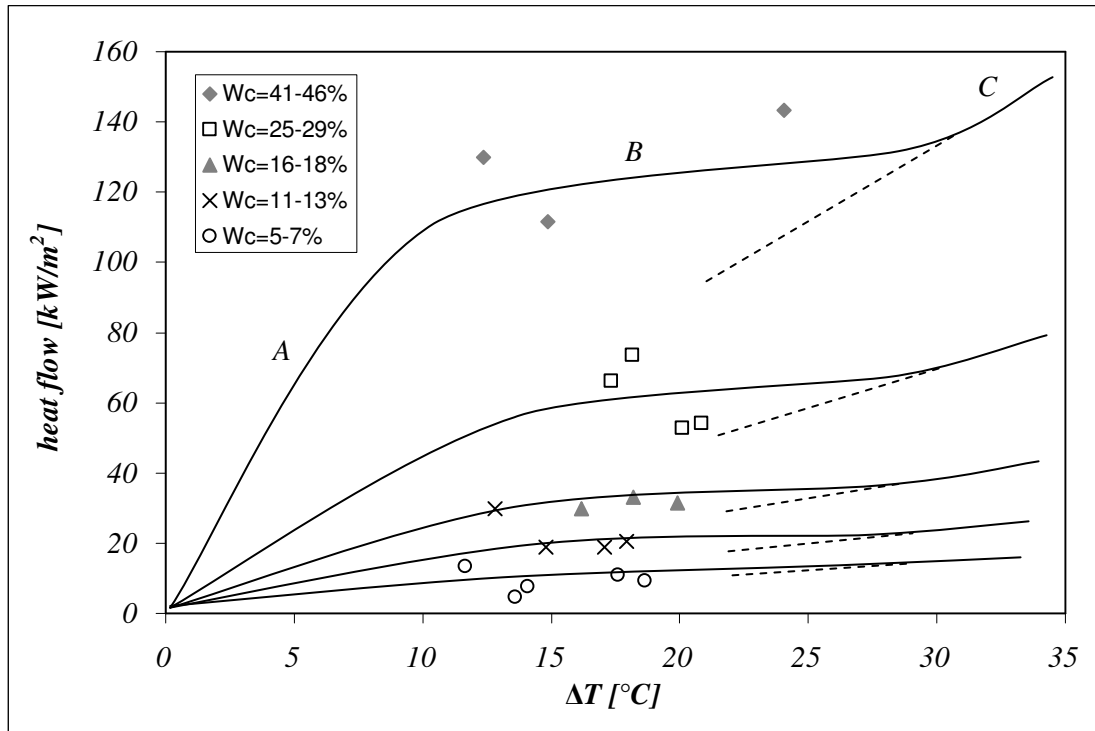


Figure 4.10: Expected relationship between the heat transfer rate and the temperature gradient in the cake layer.

The data points in Fig.4.10 were derived from the cooling profiles for dredging sludge (Fig.4.8) and the accompanying pressures in the vessel. They were added to the graph for the sake of explanation. Additional experiments are necessary to unveil the exact shape of the curves, and to verify whether the data points actually belong to regime *B*. An important conclusion to be drawn at this moment is that it is safe to assume the drying rate overall proportional to the temperature gradient. This assumption does not lead to overestimation of the kinetics of evaporation.

Fig.4.11 and Fig.4.12 show apparent heat transfer coefficients of the tested cake samples as function of moisture content. For dredging spoil at moisture contents above 20%, the values are significantly higher than for a conductive film of water (thermal conductivity $\sim 0.6 \text{ W/mK}$) with comparable thickness ($\sim 0.5 \text{ mm}$). Sewage cake shows heat transfer coefficients that are lower than those of dredging spoil by a factor of three, despite the fact that the cake was spread out in thinner layers.

The faster the cake dries, the larger the drying capacity is and the higher the process efficiency, since heat losses per unit of removed moisture will be less. Increased

temperature gradients obviously accelerate the process. However, they are not necessarily beneficial. They raise the temperature differences between the evaporation and condensation stages and therefore cost energy as well. With this in mind, the consequences of the presented data are further investigated in Paragraph 6.2. The total temperature gradient within the flash stages is examined, taking into account other contributions as well.

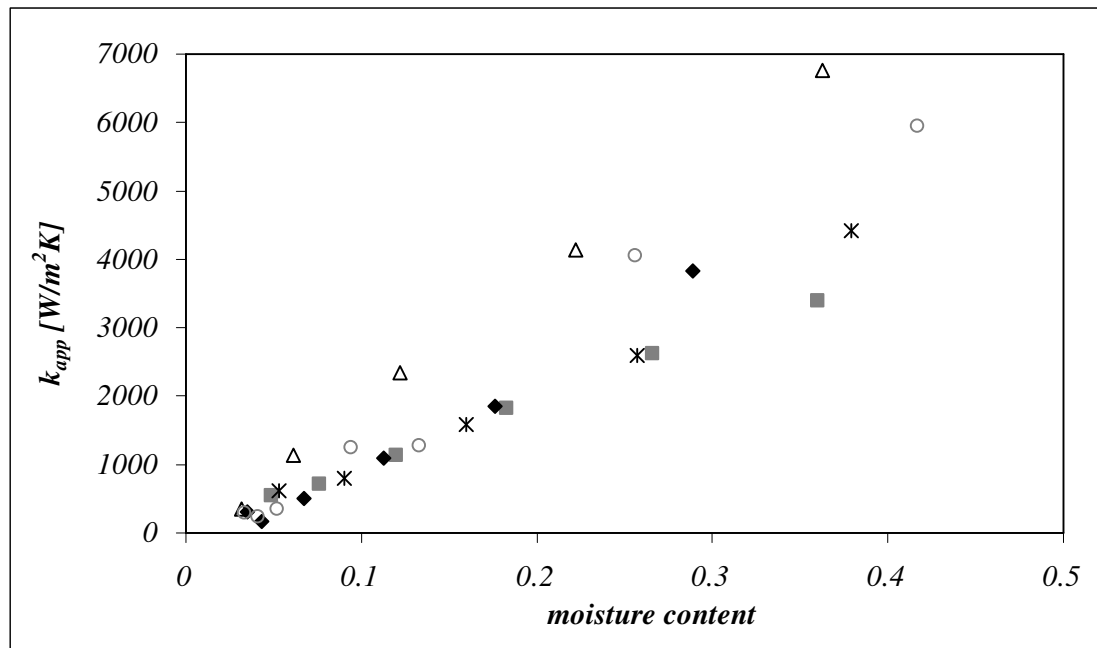


Figure 4.11: Apparent heat transfer coefficient for dredging spoil as function of moisture content (on a wet basis). The average initial layer thickness was $\sim 0.5mm$.

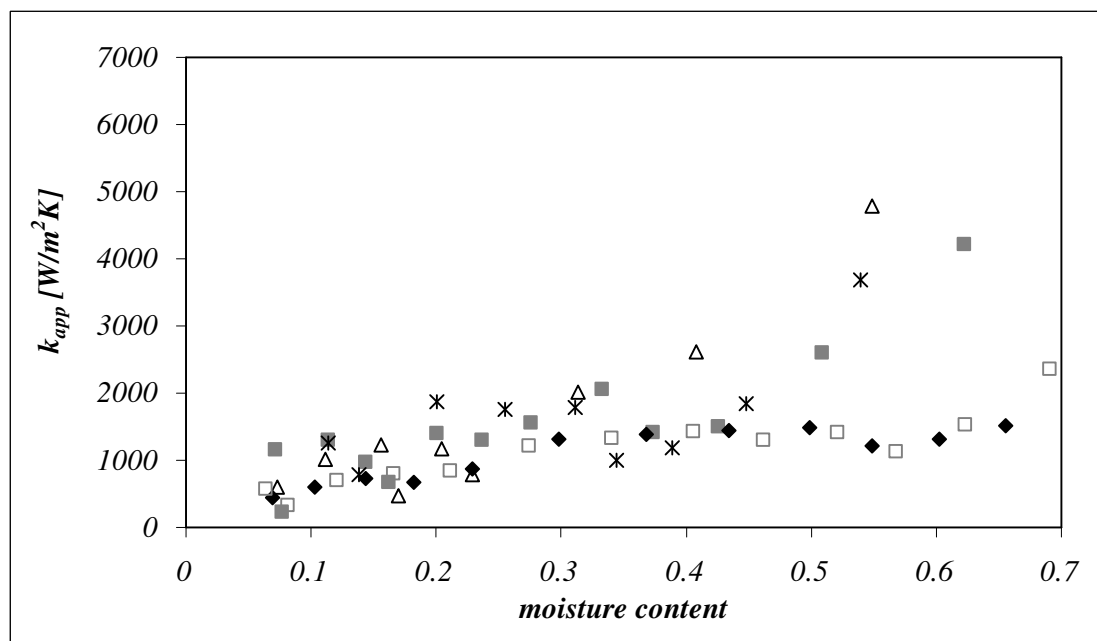


Figure 4.12: Apparent heat transfer coefficient for sewage cake as function of moisture content (on a wet basis). The average initial layer thickness was $\sim 0.35mm$.

4.4 Separation of balls and cake

4.4.1 Physical description of the screening step

The product should detach and separate from the balls stream after the end of the drying period. The separation step takes place inside a drum screen. The recovery depends on the rate at which the product liberates from the balls, and on the velocity at which the liberated particles move towards the screen. Two liberated fractions are distinguished: a 'loose' fraction A, which comes off almost instantly as the balls enter the screen; and a fraction $\alpha_{liberated}$ that detaches in the course of the milling operation. The latter fraction grows with time and intensity of the screening operation, as proposed by Equation 4.3.

$$\frac{\partial \alpha_{liberated}}{\partial t} = F \cdot \lambda \cdot (1 - \alpha_{liberated}) \quad \text{with:} \quad \lambda = \frac{1}{1 + e^{\frac{Wc - Wc_{50}}{\Delta Wc}}} \quad [4.3], [4.4]$$

The factor λ in Equation 4.3 is only for the purpose of numerical computations in simulations of the drying process. Its value is about unity for all practical cases, but drops rapidly when the moisture content rises to the sticky limit. Arbitrarily, the relationship between λ and the moisture content of the product is approximated by a sigmoid function (Equation 4.4), as graphically expressed in Fig.4.13. The parameters Wc_{50} and ΔWc are specific properties of the cake. Wc_{50} obviously relates to the sticky limit. Considering the data of Table 4.2 and keeping in mind that the desired moisture content of the product is typically 10%, the cake should have passed well beyond this point at the time it reaches the end of the drying treatment.

The factor F is the real parameter describing the kinetics of the liberation process. Technically, it is a function of both the milling action of the drum screen and the consistency of the dry cake. However, the operation of the drum screen will generally be close to the optimum milling conditions, and so, for practical purposes, F is rather a property of the dry cake, indicating the ease of removal from the balls.

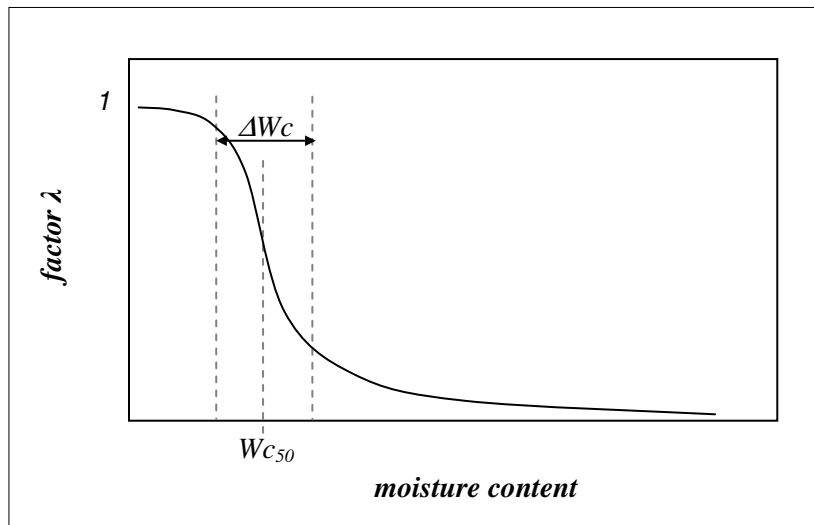


Figure 4.13: Graphical presentation of λ as function of the moisture content of the product.

Recently liberated particles are mostly locked up in the empty spaces between the balls. Motion of the bed is required to make them drop towards the screen. The turbulence created by the drum screen is sufficient from this perspective. It even homogenises the bed, which is noticeable after adding a handful of coloured balls: these distribute randomly within two to three revolutions of the screen.

The fraction of recovered product within an infinitesimal time step follows from Equation 4.5. It is assumed proportional to the mean vertical velocity of the product (v) and inversely proportional to the height of the ball bed (h). Equation 4.6 approximates the vertical velocity. t_{drop} is the period in which the product passes a single layer of balls. It is basically the sum of the falling time and the period between two opportunities. The influence of the geometry of the screen and additional shear created by the presence of stirring rods is incorporated into an empirical factor F_{mix} . Substitution of [4.3] in [4.5] and integration yields the total recovered fraction as a function of time (Equation 4.7).

$$\frac{\partial \alpha_{recovered}}{\partial t} = (\alpha_{liberated} - \alpha_{recovered}) \cdot \frac{v}{h} \quad [4.5]$$

$$v = \frac{D_{ball}}{t_{drop}} \quad \text{with:} \quad t_{drop} \approx \sqrt{\frac{2D_{ball}}{g}} + \frac{F_{mix} \cdot height}{\pi \cdot D_{screen} \cdot rvps} \quad [4.6]$$

$$\alpha_{recovered} = 1 + \frac{1}{\frac{v}{h} - F \cdot \lambda} \left((A-1) \cdot \frac{v}{h} \cdot e^{-F \cdot \lambda \cdot t} + \left(F \cdot \lambda - A \cdot \frac{v}{h} \right) e^{-\frac{v}{h} \cdot t} \right) \quad [4.7]$$

4.4.2 Screening experiments

Laboratory screening experiments were done to check the separation step. The investigated samples were coal fines, sewage cake and dredged material. These were selected for their differences in initial moisture content, particle sizes distribution, and particle shapes. The ratio at which the samples were mixed with balls was comparable to the samples of Fig.4.2, i.e. based on 100kg of steel per kg of cake moisture. After being dried in an oven, the mixtures moved to a drum screen with a width of 0.2mm, diameter of 0.55m, revolution speed of 0.5Hz, and 25% open area. The mass of the ball bed was 40kg and the height was around 0.1m. The ball diameter was 12mm.

Fig.4.14 and Fig.4.15 present results from the screening experiments. Unsurprisingly, the coal appeared easiest to recover, since the material abounded in the mixture, detached easily, and dropped rapidly. The mass percentage of solids in the ball bed was almost nine times less for sewage cake than for coal. However, most of the dried sewage cake peeled off as thin flakes and could be recovered at an early stage. Dredged material, on the other hand, was found more difficult to liberate. The fines also produced significant amounts of dust. (This does not show from the graph since the dust was collected during the experiment.) In a real drying installation, part of the dust will carry along with the vapour towards the condenser.

Mass balances proved that it is possible to remove virtually all solids from the balls, on the condition that the duration of the milling operation is large enough. Recovery of 90-95% of the dried material is more realistic for practical reasons. This percentage obviously drops if the product is not bone dry.

The residual solids that move together with the balls to the next process step are lost. Part of the solids may be washed off with sprinkler water and leave the installation as a watery waste stream. The rest collects condensate, forming a sticky or viscous film around the balls. This film will trickle off less easily than pure water, which will affect the kinetics of the condensation process and the final dewatering stage in the steam vessel. The solids may also foul secondary pieces of equipment that function better when clean. Poor separation can thus result in more than one way to higher energy consumptions per kilogram of recovered product. Deep removal of the solids from the balls is therefore essential.

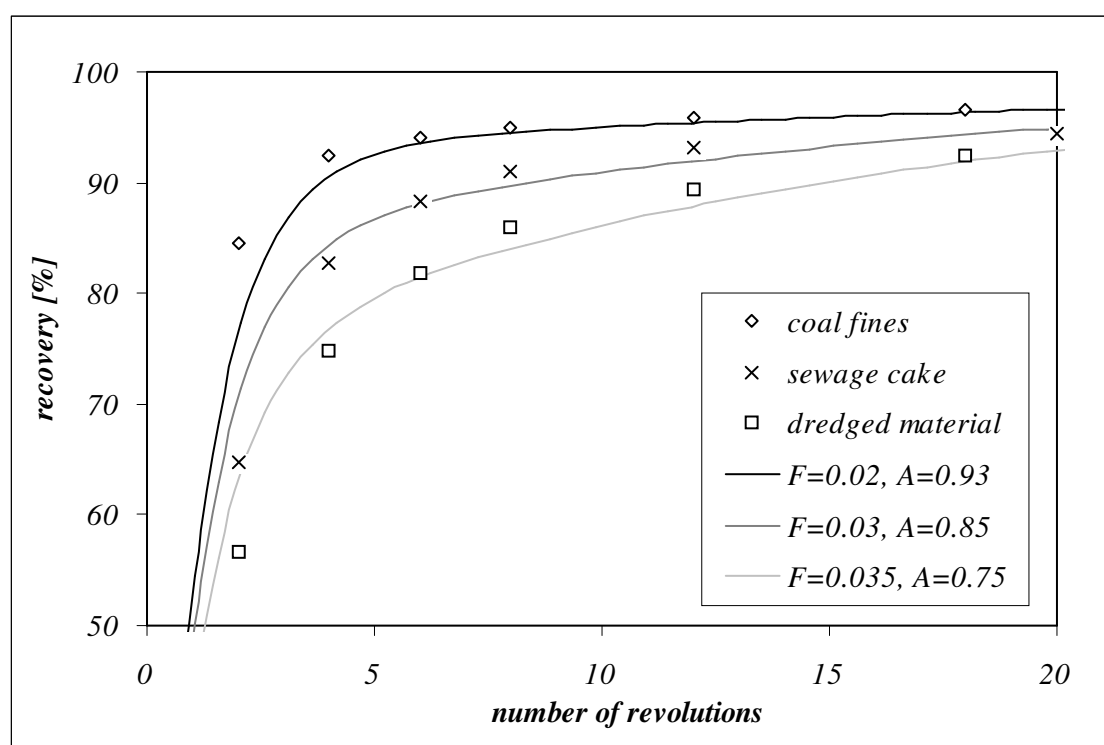


Figure 4.14: Recovery as function of the number of revolutions of the drum screen. The trend lines were fit using Equation 4.7 with $F_{\text{mix}}=2$ and different values for F and A .

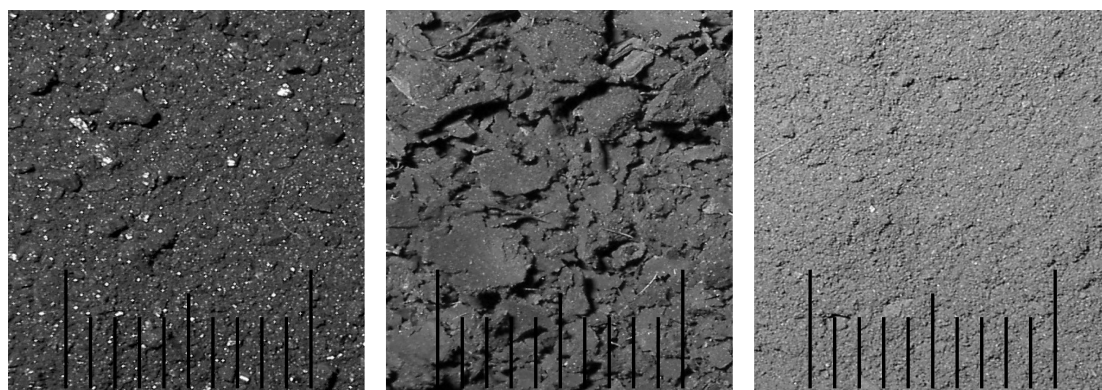


Figure 4.15: Photographs of recovered fractions of coal (left), sewage cake (centre) and dredging fines (right). Note the millimetre scale.

Chapter 5: The local equilibrium model and its results

Paragraph 3.3 explored the theoretically maximum potential of the Delta dryer concept. The ‘very simple model’, which was presented in that paragraph, did not take into account the presence of non-condensing gases, important heat leaks, and the hygroscopic properties of the cake, although these can exert large influences on the energy economy of the process. For this reason, the ‘local equilibrium model’ was developed. This dynamic physical model describes the process as it takes place in the core of the Delta dryer, i.e. in the two screw conveyors. It focuses on the primary physical phenomena in this part of the installation, without paying much attention to the performance of peripherals.

Like any model, the local equilibrium model offers a simplified view upon reality. Its main limitation implies the supposition that within each pitch of the screws containing balls, solids, water, and gas, the three phases are always in thermodynamic equilibrium. (In reality, there will be a temperature gradient from the centre of the balls outward to the gas phase.) Understanding the strengths and limitations of the model is essential for a correct interpretation of its results.

This chapter contains three paragraphs. The first paragraph describes the model. It starts with the introduction of a simplified dryer geometry and a short outline of the structure of the model. It subsequently goes more in depth while elaborating on the physical background of various sub-processes that cause fluxes of heat and mass. The essentials of these sub-processes are then combined to prescribe the overall process.

The second paragraph shows results of the model for standard input variables that were chosen as a reference. It presents temperatures, pressures, and moisture distributions inside the installation, as well as a mass and heat balance, and a relationship between energy consumption and the final moisture content of the product. The data were collected using simulation software. They all relate to stationary conditions, excluding transient stages such as the start-up phase or shortly after parameter changes.

The sensitivity analysis in the third paragraph evaluates influences of design issues and feed parameters. It displays effects of a sprinkler system, leaks in longitudinal direction of the conveyors, air leaks from outside, the number of vapour bridges, the hygroscopic nature of the cake, and the feed rate. To keep a structured overview and enable fair comparisons between results, all investigated simulations were derived from the reference by changing only one or two parameters at a time.

5.1 Description of the local equilibrium model

5.1.1 Simplified presentation of the core of the Delta dryer

The Delta drying process is the result of a number of sub-processes. The most important of these take place inside the two screw conveyors. Each screw conveyor comprises a hollow cylinder, a rotating axis with screw blades, a strip placed on top of the blades (for reasons explained later), and a brush bridging the gap between the screw blades and the hull. Fig.5.1 illustrates the components.

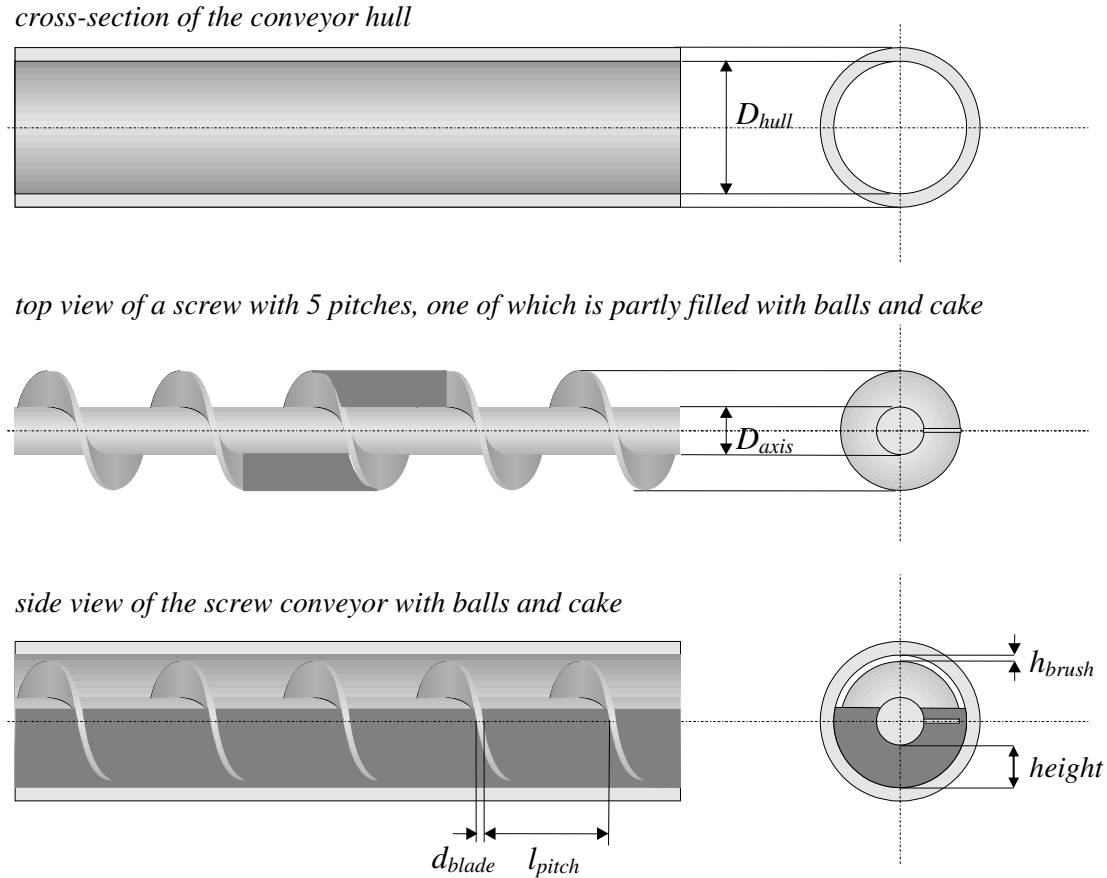


Figure 5.1: Views of a screw conveyor. The strip and brush on top of the screw blades have not been drawn.

The starting point for the local equilibrium model is an alternative geometrical presentation of the screw conveyors. This will make no difference fundamentally, but enhances insights. While the screw conveyors are in reality positioned anti-parallel to each other, they are for now imagined in line and level. The second screw continues where the first one ends, so that both conveyors combined form one long conveyor (Fig.5.2). Wet cake and hot balls enter from one side; dried material leaves via a screen positioned approximately half way down the conveyor; and the preheated balls return instantaneously from the end of the conveyor, to be steam heated, dewatered, and reintroduced into the drying loop.

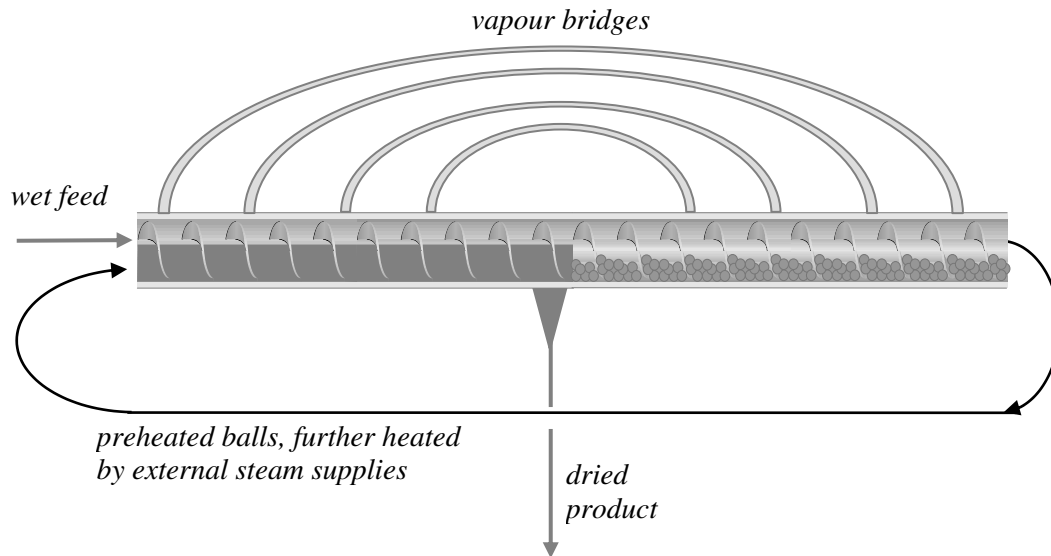


Figure 5.2: Schematic presentation of the two screw conveyors as a single long conveyor. Arrows indicate the feed and discharge operations and the closed circuit of balls running through the conveyor. A few vapour bridges are shown as well.

The free space between the axis and the hull can be visualised as a giant hollow telephone wire. The mixture of balls and cake fills the void at least up to the axis. If the imaginary telephone wire is stretched, a straight tube forms, containing alternating packages of balls and vapour. The ball packages act as barriers between the vapour compartments, providing a certain resistance to vapour flows along the axial direction of the tube. As a result, the tube is divided into a train of chambers or sections, the length of each section corresponding with the length of one pitch.

A section thus contains volumes of balls, solids, liquid water, vapour, air, and some steel representing part of the screw and the conveyor hull. It is considered the largest unit with a homogeneous composition, exhibiting average variables. The local equilibrium model allows neither pressure gradients nor temperature gradients between the product, balls, steel parts, or gas phase within a section. In approximation of reality, the content of every section is assumed to be always in thermodynamic equilibrium.

When in operation, the Delta dryer transports balls and cake through the conveyor, while the contents, pressures, and temperatures of the sections gradually change under the influence of various fluxes of mass and heat. These fluxes are the result of predefined connections (Fig.5.3). The connections represent:

- Leaks from outside
- Vapour bridges between sections
- Vapour bridges between sections and spaces with controlled conditions
- The permeability of the ball bed separating the sections
- Small gaps between the screw blades and the conveyor hull
- Thermally conductive parts of the screw blades and conveyor hull
- Feed and discharge openings for cake and liquid

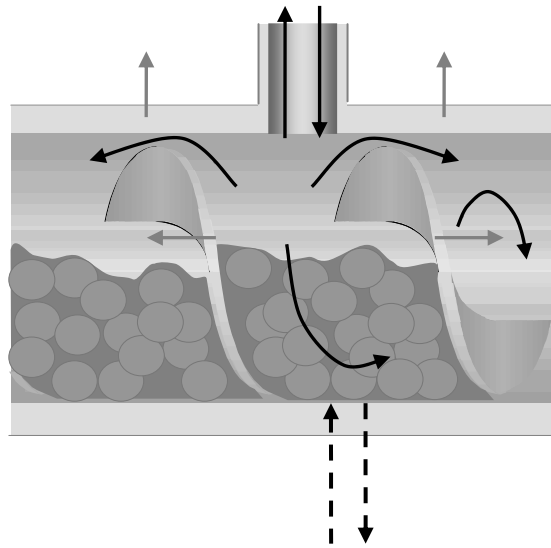


Figure 5.3: Possible fluxes of heat and mass: gas fluxes through the vapour bridges, the porous ball beds, and the gaps between the screw blades and the hull (black); exchanges of gas, liquid and solids with the exterior (dashed); and heat flows by conduction through the screw blades and the conveyor hull (grey).

The sections maintain their specific connections during a revolution period of the screw conveyor. The compositions of the sections change thanks to heat and mass flows. These changes are easy to calculate for the predefined fluxes, which are:

- Inputs of water and solids (as part of the cake) into the first section
- Discharges of dried product at the screening point
- Air leaks from outside
- Water supplies by the sprinkler installation
- Discharges of water at the sprinkler screen

Other fluxes are not known in advance. They depend on continuously changing gradients in temperature and pressure, as described in the next five sections. The sum of the incoming and outgoing fluxes is calculated for an infinitesimal time step, after which the pressure and temperature of each section is determined based on its new content and enthalpy. (This is further explained in Section 5.1.7.) The contents after a revolution of the screw equal the old contents plus the net fluxes integrated over the revolution period. At the end of that period, the content of each section shifts across one pitch length toward the next section. The content of the last pitch leaves the screw conveyor, heats up by external steam supplies, dewateres, and subsequently returns to the first section. Then, the procedure starts all over for the next time interval, during which new contacts are realised and new fluxes arise. It repeats for every following revolution of the screw conveyor.

Simulation software was developed to facilitate the integration steps. Data files containing design, control, and feed parameters serve as input for the simulations. Output files list principle variables for each section and every simulated revolution of the screw conveyors. External heat supplies, final moisture contents of the product, and magnitudes of various heat and mass flows are stored as well.

5.1.2 Gas fluxes through the vapour bridges

The vapour bridges between sections and to spaces with controlled conditions allow gas fluxes driven by pressure gradients. The gas mixture contains vapour and small amounts of air. Fines that may move along with the gas are left out of consideration, because no model is available for their estimation.

The Reynolds number (Re) and the friction factor (f) are defined after Equations 5.1 and 5.2. ΔP is the pressure drop in a tube section with length l_{bridge} , hydraulic diameter D_{bridge} and cross-sectional area A_{bridge} . The parameters ρ_{gas} , μ_{gas} and v_{mean} are the density, dynamic viscosity and the mean velocity of the fluid. W is the mass flow rate.

$$Re \equiv \frac{D_{bridge} \cdot \rho_{gas} \cdot v_{mean}}{\mu_{gas}} = \frac{W \cdot D_{bridge}}{A_{bridge} \cdot \mu_{gas}} \quad [5.1]$$

$$f \equiv \frac{\Delta P \cdot D_{bridge}}{2 \rho_{gas} \cdot v_{mean}^2 \cdot l_{bridge}} = \frac{\Delta P \cdot \rho_{gas} \cdot D_{bridge} \cdot A_{bridge}^2}{l_{bridge} \cdot W^2} \quad [5.2]$$

It is well-known that f is inversely proportional to Re for low Reynolds numbers (laminar flow), and constant at high Reynolds numbers (turbulent flow). For the range of possible flow velocities, the friction factor is therefore approximated as the sum of a turbulent term and a laminar term. At high Re , the constant turbulent term dominates the laminar term, and vice versa at low Re . The pressure drop over a tube segment is then also a sum of both terms.

$$f = C1 + \frac{C2}{Re} \quad [5.3]$$

$$\Delta P = \frac{a}{\rho_{gas}} \cdot W^2 + \frac{b}{\rho_{gas}} \cdot \mu_{gas} \cdot W \quad [5.4]$$

a and b are geometric parameters of the tube. They can be derived if the constants $C1$ and $C2$ are known (see Fig.5.4). For a vapour bridge comprising multiple interlinked components in series, the total pressure drop is the sum of the pressure drops over all individual components. If μ_{gas} and ρ_{gas} are constant, the geometric parameters a and b of the components need to be summed (Equation 5.5).

$$\Delta P_{total} = \sum_{all \ comp.} \Delta P = W^2 \left(\frac{1}{\rho_{gas}} \cdot \sum_{all \ comp.} a \right) + W \left(\frac{\mu_{gas}}{\rho_{gas}} \cdot \sum_{all \ comp.} b \right) \quad [5.5]$$

The mass flow rate can be solved from Equation 5.5. W_i is the gas flux ($\Phi_{gas,i}$) through the tube from a section with higher pressure to a connecting section i with lower pressure. The fluxes of air (Φ_{air}), vapour (Φ_{vapour}) and heat (Φ_{heat}) into section i follow from the air fraction in the gas (F_{air}) and the specific enthalpy of the gas (h_{gas}) (Equations 5.7-5.9).

$$W_i = \frac{2\rho_{gas} \cdot (P_j - P_i)}{\mu_{gas} \sum b + \sqrt{(\mu_{gas} \cdot \sum b)^2 + 4\rho_{gas} \cdot (P_j - P_i) \cdot \sum a}} \quad [5.6]$$

$$\Phi_{air,i} = \Phi_{gas,i} \cdot F_{air,i} \quad [5.7]$$

$$\Phi_{vapour,i} = \Phi_{gas,i} \cdot (1 - F_{air,i}) \quad [5.8]$$

$$\Phi_{heat,i} = \Phi_{gas,i} \cdot h_{gas,i} \quad [5.9]$$

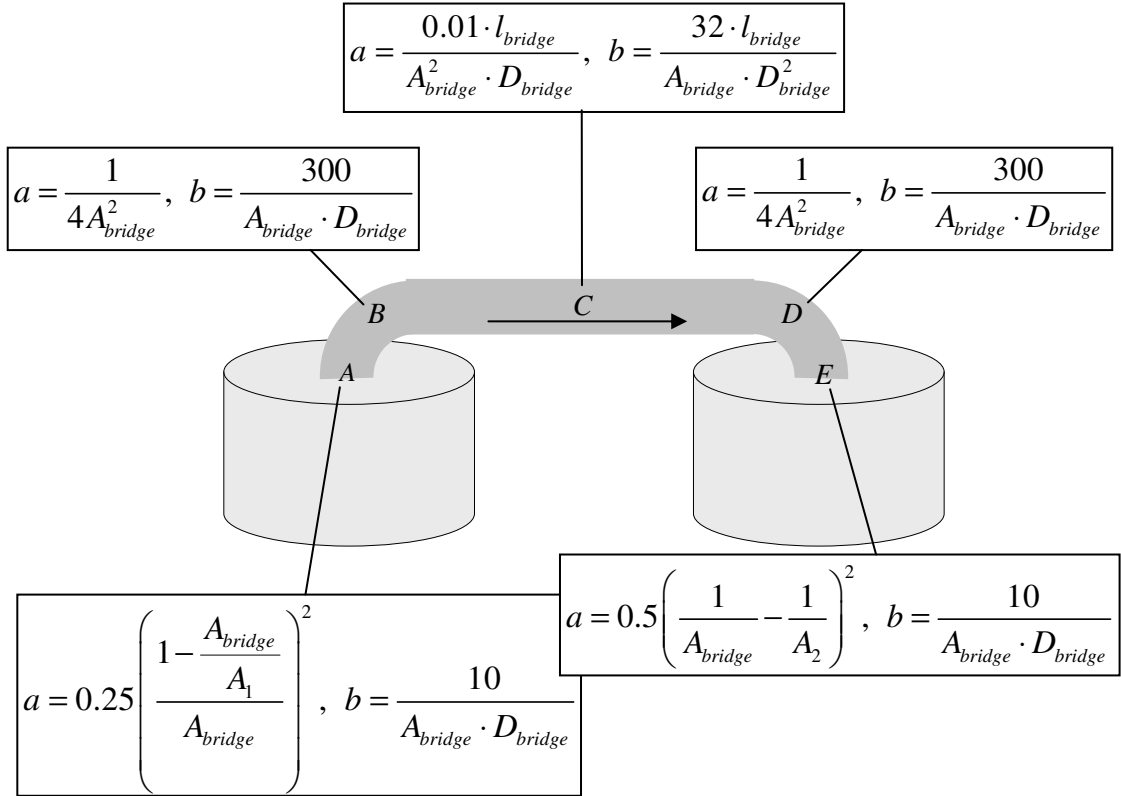


Figure 5.4: Two compartments with cross-sectional areas A_1 and A_2 , interconnected via a vapour bridge with length l_{bridge} , hydraulic diameter D_{bridge} , and cross-sectional area A_{bridge} . The bridge comprises a sudden contraction (A), two 90-degree knees (B, D), a straight tube section (C), and a sudden expansion (E). Presented values for a and b are from [Perry, 1997].

5.1.3 Gas fluxes through the ball bed

The resistance of the ball bed slows down unfavourable gas flows along the axial direction of the conveyors, but is not large enough to prevent them. Ergun formulated a relationship between the pressure drop in a packed bed with length l_{bed} , the superficial flow velocity of the gas (v_s), and the dynamic viscosity and density of the gas (Equation 5.10) [Ergun, 1952]. The parameter ε_{void} represents the void volume fraction in the bed. Its value for loosely packed beds is approximately 0.37, from which, in this particular case, the volume fractions of solids and water must be subtracted.

$$\frac{\Delta P}{l_{bed}} = \frac{1.75(1-\varepsilon_{void}) \cdot \rho_{gas} \cdot v_s^2}{\varepsilon_{void}^3 \cdot D_{ball}} + \frac{150(1-\varepsilon_{void})^2 \cdot \mu_{gas} \cdot v_s}{\varepsilon_{void}^3 \cdot D_{ball}^2} \quad [5.10]$$

The gas velocity in Equation 5.10 is substituted by the mass flow rate, yielding Equation 5.11. A_{bed} equals $l_{pitch}(D_{hull}-D_{axis})/2$. The length of the bed approximates V_{bed}/A_{bed} , but its effective value may well be lower depending on the shape of the bed and the fill level in the conveyor.

$$\begin{aligned} \Delta P &= \frac{1.75(1-\varepsilon_{void}) \cdot l_{bed}}{\varepsilon_{void}^3 \cdot D_{ball} \cdot A_{bed}^2 \cdot \rho_{gas}} \cdot W^2 + \frac{150(1-\varepsilon_{void})^2 \cdot \mu_{gas} \cdot l_{bed}}{\varepsilon_{void}^3 \cdot D_{ball}^2 \cdot A_{bed} \cdot \rho_{gas}} \cdot W \\ &= \frac{a}{\rho_{gas}} \cdot W^2 + \frac{b}{\rho_{gas}} \cdot \mu_{gas} \cdot W \end{aligned} \quad [5.11]$$

The mass flow rate is solved analogously to the mass flow rate in Equation 5.5. The fluxes of air, vapour and heat from section $i-1$ to section i are calculated using Equations 5.7-5.9.

$$W_i = \frac{2\rho_{gas} \cdot (P_{i-1} - P_i)}{\mu_{gas} \cdot b + \sqrt{\mu_{gas}^2 \cdot b^2 + 4\rho_{gas} \cdot (P_{i-1} - P_i) \cdot a}}, \text{ with:} \quad [5.12]$$

$$\begin{aligned} a &= \frac{1.75(1-\varepsilon_{void}) \cdot l_{bed}}{\varepsilon_{void}^3 \cdot D_{ball} \cdot A_{bed}^2} \\ b &= \frac{150(1-\varepsilon_{void})^2 \cdot l_{bed}}{\varepsilon_{void}^3 \cdot D_{ball}^2 \cdot A_{bed}} \end{aligned}$$

5.1.4 Gas fluxes through the gap between screw blades and hull

The tolerances between the screw blades and the conveyor hull will be narrow, but anything near a press fit is obviously not practicable. Few millimetre wide gaps between the blades and the hull allow gas leaks between neighbouring sections. The sudden contraction, expansion, and the length of the path provide a certain resistance to these gas flows. The resistance increases substantially when a brush is positioned around the outer contour of the blades, as depicted in Fig.5.5. Additional pieces of sheet are placed on top of the blades, preventing short-circuiting between sections when the blades pass the vapour bridge openings.

The mass flow rates through the gaps can be calculated using Equation 5.13. Σa and Σb were derived from the equations referring to a packed bed. D_{hair} is the hair diameter of the brush, ε_{brush} is the volume fraction of hair in the brush, and the other terms represent the resistances due to sudden contraction and expansion. A_{gap} is the shared gap area between a section and its neighbour. As each section has a length of one pitch, A_{gap} equals the product of h_{gap} and the gap length per pitch. Although part of the area is actually located below the surface of the ball bed, the whole gap is described as if it connects two empty gas compartments.

The fluxes of gas, air, vapour, and heat follow once more from Equations 5.7-5.9.

$$W_i = \frac{2\rho_{gas} \cdot (P_j - P_i)}{\mu_{gas} \sum b + \sqrt{\mu_{gas}^2 \sum^2 b + 4\rho_{gas} \cdot (P_j - P_i) \sum a}} \quad , \quad \text{with:} \quad [5.13]$$

$$\sum a = \frac{1.75 \cdot \epsilon_{brush} \cdot B_{brush}}{(1 - \epsilon_{brush})^3 \cdot D_{hair} \cdot A_{gap}^2} + \frac{0.75}{A_{gap}^2}$$

$$\sum b = \frac{150 \cdot \epsilon_{brush}^2 \cdot B_{brush}}{(1 - \epsilon_{brush})^3 \cdot D_{hair}^2 \cdot A_{gap}} + \frac{10}{A_{gap} \cdot h_{gap}}$$

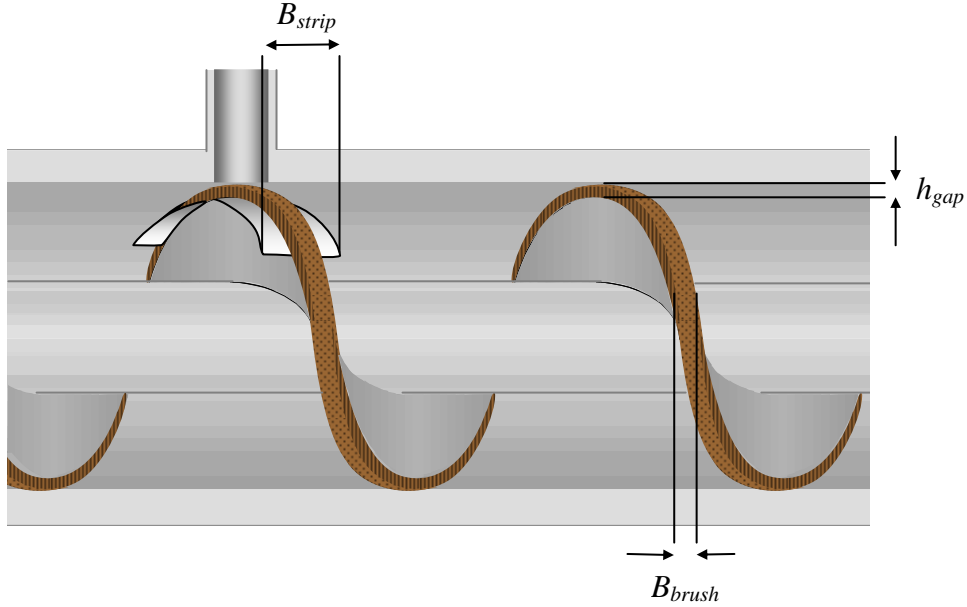


Figure 5.5: Schematic cross-section of a conveyor hull with vapour bridge and screw axis. A piece of metal sheet and a brush are positioned on top of the screw blades.

5.1.5 Heat fluxes through the conveyor hull and screw blades

A Delta dryer operates at elevated temperatures and therefore loses heat to its surroundings. The model takes into account the losses by conduction through the conveyor hull. It uses Fourier's law with three contributions to the heat resistance: the steel hull itself, the insulation material covering the surface, and a thin condensed water film that may be present on the inside. The temperatures of the considered section and the outer surface are T_i and T_{out} respectively. Possible losses from other parts of the installation are ignored.

$$\Phi_{heat,i} = \frac{\pi \cdot D_{hull} \cdot l_{pitch} \cdot (T_{out} - T_i)}{\frac{d_{waterfilm}}{\lambda_{liquid}} + \frac{d_{hull}}{\lambda_{hull}} + \frac{d_{insulation}}{\lambda_{insulation}}} \quad [5.14]$$

Heat transfer between neighbouring sections also takes place by conduction through the screw blades. The blades are covered with a thin liquid film, surrounded by a laminar vapour layer. The thickness of this laminar layer is approximately 0.1mm at atmospheric pressures [Wesselingh, 1990]. The equation describing the heat flux from section j to section i is analogous to Equation 5.14. $A_{blade/section}$ is the area of the screw blade in between two sections.

$$\Phi_{heat,i} = \frac{A_{blade/section} \cdot (T_j - T_i)}{\frac{d_{waterlayer}}{\lambda_{liquid}} + \frac{d_{blade}}{\lambda_{screw}} + \frac{d_{vapour}}{\lambda_{vapour}}} \quad , \text{ with:} \quad [5.15]$$

$$A_{blade/section} \approx \frac{D_{hull} - D_{axis}}{2} \cdot \frac{\sqrt{l_{pitch}^2 + \pi^2 D_{hull}^2} + \sqrt{l_{pitch}^2 + \pi^2 D_{axis}^2}}{2} \quad [5.16]$$

5.1.6 Heat fluxes due to the stationary position of the screws

The total enthalpy of the sections within the screw conveyor includes the enthalpy of the screw blades and the conveyor hull. Unlike the balls and the cake, the screw does not pass through the drying stages. During every revolution ($1/rvps$ seconds), the total enthalpy of a section gradually increases with the enthalpy of a successive pitch of the screw and decreases with the enthalpy of its own part of the screw. The corresponding heat flux to section i from its successor is:

$$\Phi_{h,i} = Cp_{pitch} \cdot (T_{i+1} - T_i) \cdot rvps \quad [5.17]$$

Cp_{pitch} is the heat capacity of one pitch of screw conveyor. It is the sum of the heat capacities of a piece of hull, axis, screw blade, and sheet on top of the blade. The heat capacity of the brush is negligible.

$$\begin{aligned} Cp_{pitch} = & Cp_{steel} \cdot \rho_{steel} \left(\pi D_{hull} \cdot l_{pitch} \cdot d_{hull} \right) \\ & + Cp_{steel} \cdot \rho_{steel} \left(\pi D_{axis} \cdot l_{pitch} \cdot d_{axis} \right) \\ & + Cp_{steel} \cdot \rho_{steel} \cdot \frac{1}{4} \cdot (D_{hull} - D_{axis}) \cdot \left(\sqrt{l_{pitch}^2 + \pi^2 D_{hull}^2} + \sqrt{l_{pitch}^2 + \pi^2 D_{axis}^2} \right) \cdot d_{blade} \\ & + Cp_{steel} \cdot \rho_{steel} \left(B_{strip} \cdot d_{strip} \sqrt{l_{pitch}^2 + \pi^2 D_{hull}^2} \right) \end{aligned} \quad [5.18]$$

5.1.7 Determination of temperatures, pressures, and compositions

The compositions of the sections change because of the fluxes of heat and mass that are entering and leaving. The contents at the end of a revolution of the screw equal the contents prior to that plus the net incoming fluxes integrated over the time step. In differential notation:

$$\frac{\partial \text{content}_i}{\partial t} = \sum_{\substack{\text{all} \\ \text{fluxes}, i}} \Phi_i \quad [5.19]$$

The actual integration is performed numerically with the help of the implicit ODE solver DASSL [NAG, 2002]. The integration procedure is not straightforward, since the pressures and temperatures (and together with those some secondary parameters) are not constant. They depend on the enthalpy of a section, the amounts of water, vapour, air and solids present, and on each other.

A section i contains masses of air ($m_{\text{air},i}$), solids ($m_{\text{solid},i}$), liquid water ($m_{\text{liquid},i}$), water vapour ($m_{\text{vapour},i}$), and steel ($m_{\text{steel},i}$). It has a total enthalpy ($H_{\text{tot},i}$). If the content is in thermodynamic equilibrium, the following relationship exists with temperature.

$$H_{\text{tot},i} = m_{\text{air},i} \cdot C_{p,\text{air}} \cdot T_i + m_{\text{solid},i} \cdot C_{p,\text{solid}} \cdot T_i + m_{\text{steel}} \cdot C_{p,\text{steel}} \cdot T_i \\ + m_{\text{vapour},i} \left(h_{\text{evap}}^{\text{0}^\circ \text{C}} + C_{p,\text{vapour}} \cdot T_i \right) + m_{\text{liquid},i} \cdot C_{p,\text{liquid}} \cdot T_i \quad [5.20]$$

The temperature could be solved if the vapour mass were known. The ideal gas law prescribes a simple correlation between $m_{\text{vapour},i}$, the vapour volume (V_{vapour}), molar mass of water (M_{water}), vapour pressure and absolute temperature. The vapour volume approximates the total volume of the section minus the volume of the balls. The internal volume of the vapour bridges as well as the relatively small volumes of liquid and solids are negligible.

$$m_{\text{vapour},i} = \frac{P_{v,i} \cdot V_{\text{vapour}}}{R \cdot (T_i + 273^\circ \text{C})} \cdot M_{\text{water}} \quad , \text{ with:} \quad [5.21]$$

$$V_{\text{vapour}} \approx \frac{\pi (D_{\text{hull}}^2 - D_{\text{axis}}^2) \cdot l_{\text{pitch}}}{4} - d_{\text{blade}} \cdot A_{\text{blade/pitch}} - \frac{m_{\text{balls}}}{\rho_{\text{steel}}} \quad [5.22]$$

The vapour pressure in Equation 5.21 is a function of the moisture content of the cake and the property Hygr representing the hygroscopic nature of the solids fraction (see Paragraph 4.2). Since it relates to the free vapour pressure, it depends strongly on temperature as well.

$$\frac{m_{\text{liquid},i}}{m_{\text{solid},i} + m_{\text{liquid},i}} > \text{Hygr} \rightarrow P_{v,i} = P_{v,\text{free}} \\ \frac{m_{\text{liquid},i}}{m_{\text{solid},i} + m_{\text{liquid},i}} < \text{Hygr} \rightarrow P_{v,i} = \frac{m_{\text{liquid},i}}{m_{\text{solid},i} + m_{\text{liquid},i}} \cdot \frac{1}{\text{Hygr}} \cdot P_{v,\text{free}} \quad [5.23]$$

Together, equations 5.20, 5.21 and 5.23 contain three unknown variables: $P_{v,i}$, T_i and $m_{vapour,i}$. These are thus fundamentally solvable. A numerical approach is necessary though, since the nature of the interrelations makes substitution of equations too complicated. The iterative process that finally yields the correct temperature of a section is the well-known ZEROIN algorithm [Dekker, 1969].

If T_i is known, P_v and $m_{vapour,i}$ can be calculated, and all missing secondary variables can be determined using Equations 5.24-5.26.

$$m_{water,i} = m_{liquid,i} + m_{vapour,i} \quad [5.24]$$

$$P_i = P_{v,i} + \frac{m_{air,i} \cdot R \cdot (T_i + 273^\circ C)}{V_{vapour} \cdot M_{air}} \quad [5.25]$$

$$h_{gas,i} = \frac{m_{air,i} \cdot Cp_{air} \cdot T_i + m_{vapour,i} \left(h_{evap}^{0^\circ C} + Cp_{vapour} \cdot T_i \right)}{m_{air,i} + m_{vapour,i}} \quad [5.26]$$

5.1.8 Definition of new contents for the first pitch

At the end of the integration procedure, the content of each section shifts across one pitch length towards the next section. The content of the last section is brought to the desired starting temperature before being reintroduced to the first pitch of the screw conveyor. The power consumption equals the revolution speed of the screw (*rvps*) times the enthalpy required to bring the content of the last pitch up to a predefined temperature T_{heat} .

$$W_{heat} = rvps \cdot (T_{heat} - T_i) \cdot (m_{kogels,i} \cdot Cp_{steel} + m_{liquid,i} \cdot Cp_{liquid} + m_{solid,i} \cdot Cp_{solid}) \quad [5.27]$$

The power consumption should obviously not exceed the maximum steam power supply. Otherwise, the end temperature will not reach T_{heat} , but a lower value (Equation 5.28).

$$T_{heat} = \frac{\frac{W_{max}}{rvps} + (m_{kogels,i} \cdot Cp_{staal} + m_{liquid,i} \cdot Cp_{liquid} + m_{solid,i} \cdot Cp_{solid}) T_i}{(m_{kogels,i} \cdot Cp_{staal} + m_{liquid,i} \cdot Cp_{liquid} + m_{solid,i} \cdot Cp_{solid})} \quad [5.28]$$

The thickness of the liquid film around the balls grows as the steam condenses. Depending on the subsequent dewatering step, a certain amount of liquid will move along with the balls to the first section. This amount of water is neglected in the local equilibrium model.

5.2 The reference simulation

The purpose of the local equilibrium model was threefold: deciding whether the MSFE-process can be arranged inside two screw conveyors; investigation of the robustness of the process; and prediction of influences of design choices, operational parameters, and feed properties on the thermal performance. The model was not meant to unveil optimum control settings and design specifications in terms of drying efficiency. It is rather a tool demonstrating trends and supporting the development of a prototype installation. This paragraph discusses results for the reference simulation.

5.2.1 Standard input and output variables

The most essential input data for the reference simulation are listed in the table below. These data are realistic. The scenario they describe entails balls with a temperature of 105°C to be mixed with wet feed at water-steel mass ratios around 1:100. This yields a mixture with a temperature close to 100°C (thus around atmospheric pressure), which is subsequently exposed to reduced pressures for a period of 50s. The specifications of the screw conveyors are considered technically feasible, their fill level, dimensions, and revolution speed allowing evaporation capacities up to a ton of water per hour. The one-sided tolerance between the screw blades and the hull is 2mm. It is bridged by a brush. One parameter has been overestimated though, which is the hygroscopic nature of the feed. The reference value for the parameter *Hygr* is 0.5. It is significantly higher than the empirically derived values in Paragraph 4.2.

Output data for the reference simulation are presented in Fig.5.6. The graphs display average temperature and pressure profiles, the amounts of solids and liquid present, and the moisture content of the cake as a function of the position inside the conveyors.

Table 5.1: Selection of input data for the reference simulation.

design parameters
inner diameter of hull 0.5; outer diameter of axis 0.273; gap between blades and hull 2mm
pitch length 0.2m; total number of pitches 70 (35 pitches per screw conveyor)
thickness wall of hull 8mm; wall of axis 8mm; thickness of blades 6mm
brush height 8mm; width 5mm; steel volume fraction 0.5; hair diameter 0.2mm
strip width 50mm; thickness of strip 6mm
linear ball speed at dewatering screen 2m/s; residence time on screen 0.3s
ball diameter 12mm; bed volume fraction 0.5; ball fraction in the bed 0.63 (66kg/section)
vapour bridge geometry: diameter 66mm, length 1m, 2knees
24 sections interconnected via vapour bridges: 1-70, 3-68, 5-66, ..., 23-48
13 sections connected to screen: sections 25 to 35, and 36, 37, 38
10 sections connected to condenser: sections 25 to 35
operational parameters
revolution time of screw conveyor 2s
steam temperature 105 °C; condenser temperature 40 °C
air leaks into 1 st section 0.0001kg/s, T=20 °C
water input into 1 st section 0.3kg/s, T=20 °C; water input into 36 th section 1kg/s, T=40 °C
solids input into 1 st section 0.3kg/s, T=20 °C, $\rho=2600\text{kg/m}^3$, $C_p=1\text{kJ/kgK}$, $Hygr=0.5$
recovery of dried product at screen 96%; removal of remaining solids at sprinkler 100%
sprinkler water output at sprinkler screen 80%

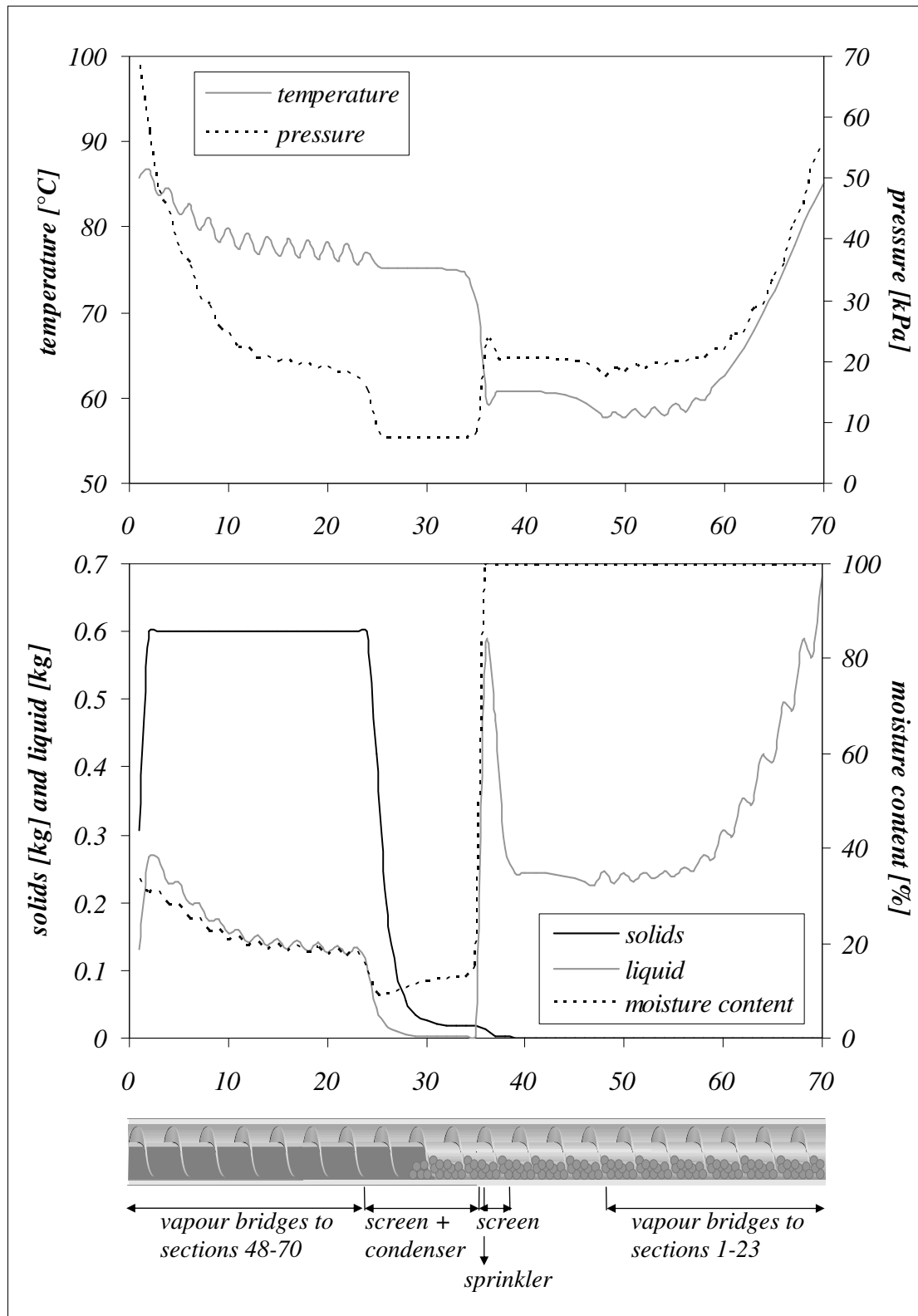


Figure 5.6: Average temperatures, pressures, solids contents, liquid contents, and moisture contents of the cake as a function of section number. The drawing below the graphs shows the positions of vapour bridges, screens, connections to the condenser, and the sprinkler system in the simulated screw conveyor.

Some distinctive trends are recognizable in Fig.5.6. The content cools off and dries in the first 23 sections. The ten following sections, which connect to the condenser, exhibit lowest pressures and moisture contents. The temperature drops steeply and the moisture content rises suddenly in the region next to the discharge area, which obviously has to do with the presence of the sprinkler installation. Finally, the contents of the last 23 sections heat up and collect condensate, furnishing evidence of the fact that part of the heat of evaporation is actually recovered here.

5.2.2 Final moisture content vs. power consumption

It is difficult to study effects of parameter changes as long as relationships between variables cannot be isolated and dealt with separately. This is especially the case for the primary output of the simulations, i.e. the energy consumptions and the final moisture contents of the product. They strongly depend on each other.

The next paragraph frequently concentrates on energy consumptions per kg of removed water. It is important to realise that this quotient is not an unambiguous performance indicator, since evaporation of water generally becomes more and more difficult at lower moisture contents. The performance of a simulated installation is much better captured in a drying curve showing the final moisture content of the product as a function of the energy consumption for specific input variables. Effects of parameters changes can be examined easily by comparing such curves.

Fig.5.7 presents the drying curve for the reference simulation. Apparently, a finished product with less than 10% water requires an external steam power supply of at least 0.52MW. For the given feed rate, initial moisture content of the cake, and final moisture content of the product, this power consumption corresponds with an energy consumption of 1.9MJ per kg of removed water.

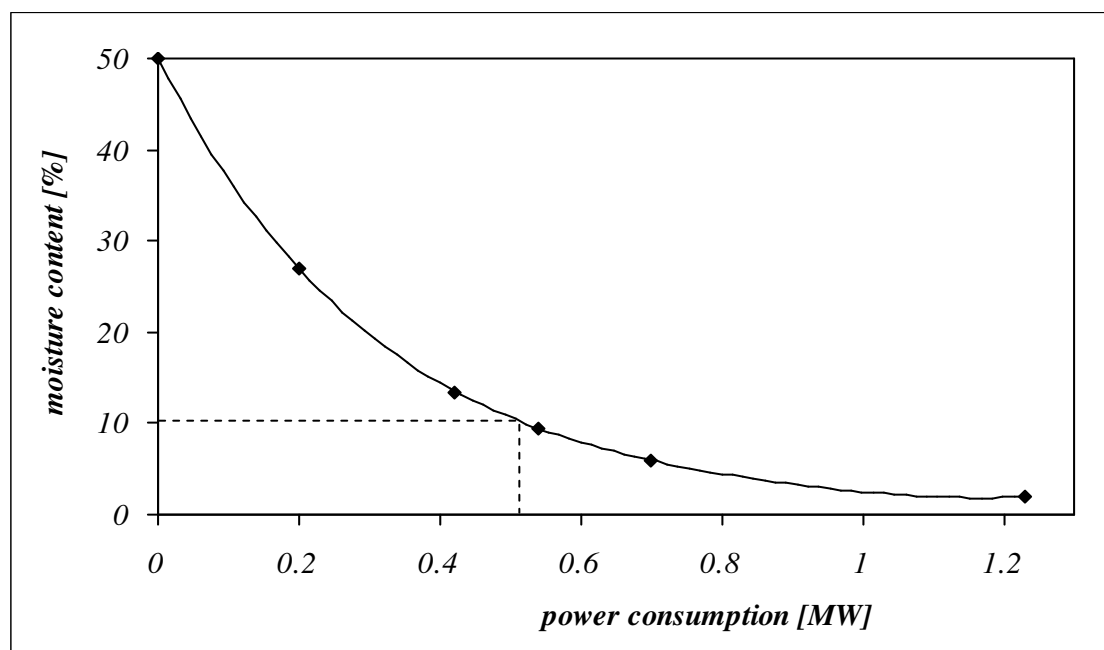


Figure 5.7: Final moisture content of the cake vs. power consumption. The data relate to the reference simulation.

5.2.3 Heat and mass balance

Various amounts of water, steam, and solids are entering and leaving the installation during a simulated revolution period. The temperatures and states of the input streams differ from the output streams. Fig.5.8 gives an overview of the mass flows and accompanying temperatures. The data are based on predefined fluxes and results from the reference simulation.

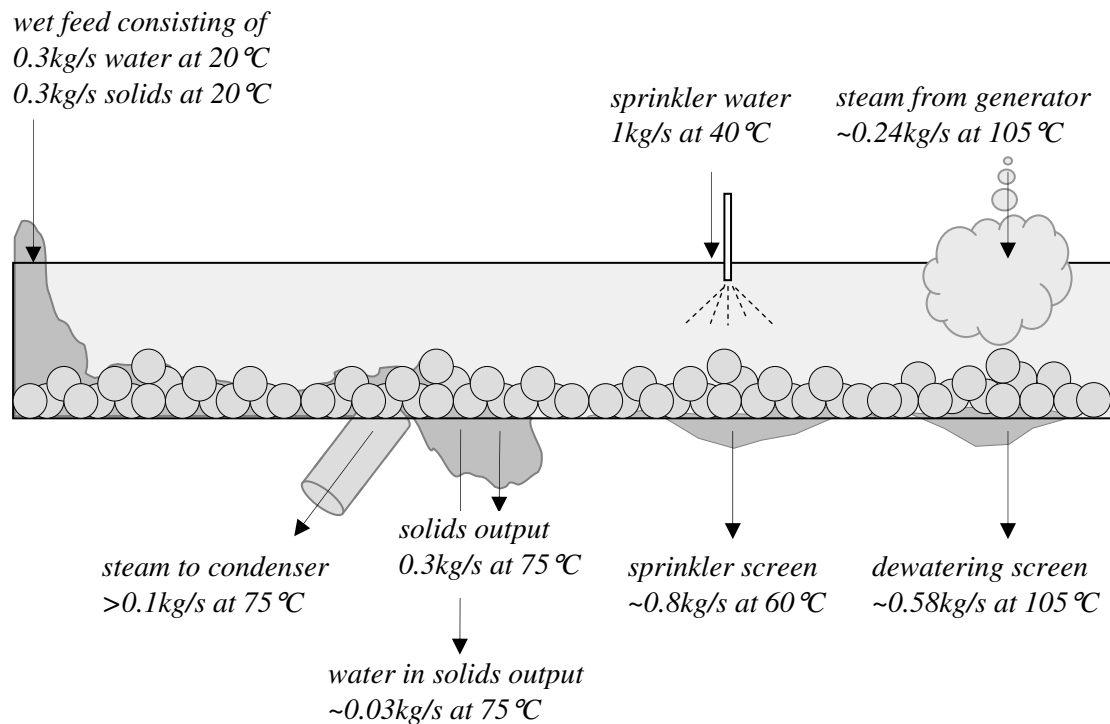


Figure 5.8: Visualisation of all mass flows entering and leaving the screw conveyors of the reference simulation.

The sums of all mass and heat flows are nil as soon as the conditions inside the installation are stationary. The fluxes to and from the installation are listed in Table 5.2. The heat losses by conduction through the conveyor hull were determined by comparing the energy consumption of the reference simulation with the energy consumption of an unrealistically well-insulated installation.

According to the table, the heat leaks through the dewatering screen, through the sprinkler screen and to the condenser are largest. Not all heat is lost though. Part of the water from the condenser serves as input for the sprinkler installation (hence the sprinkler water temperature of 40°C), whereas the output of the dewatering screen is either directly recycled to the steam generator, or indirectly benefited by means of a heat exchanger. If no heat from the dewatering screen were recovered, the power consumption for the reference simulation would increase to 0.63MW , which corresponds with the steam power supply in the table.

Despite the anticipated energy savings mentioned, there is still heat left to be recovered. Excess water from the dewatering screen could for instance be applied to further raise the temperature of the sprinkler water, or to preheat the wet cake by connecting the hot water outlet to the feed extruder. Next to these options, there are several possibilities to reduce heat losses.

It is difficult, however, to foresee secondary effects of adaptations on the heat flows and conditions inside the installation. Interesting modifications to the standard case are therefore judged on the basis of simulation results. They are discussed in the next paragraph.

Table 5.2: Enthalpy gains and losses from different flows of water, steam and solids, and by thermal conduction through the conveyor hull. The reference temperature is 20°C.

<i>input sources</i>	<i>parameters</i>	<i>heat gain [MW]</i>
<i>water from feed</i>	+0.30kg/s, 20°C, $C_p=4.18\text{kJ/kgK}$	+0.000
<i>solids from feed</i>	+0.30kg/s, 20°C, $C_p=1.00\text{kJ/kgK}$	+0.000
<i>sprinkler water input</i>	+1.00kg/s, 40°C, $C_p=4.18\text{kJ/kgK}$	+0.084
<i>steam from generator</i>	+0.24kg/s, 105°C, $C_p=1.90\text{kJ/kgK}$, $H_{\text{evap}20^\circ\text{C}}=2.45\text{MJ/kg}$	+0.626
<i>total</i>	+1.54kg/s water and +0.3kg/s solids	+0.71

<i>output sources</i>	<i>parameters</i>	<i>heat loss [MW]</i>
<i>water from end product</i>	-0.03kg/s, 75°C, $C_p=4.18\text{kJ/kgK}$	-0.007
<i>solids from end product</i>	-0.30kg/s, 75°C, $C_p=1.00\text{kJ/kgK}$	-0.017
<i>dewatering screen</i>	-0.58kg/s, 105°C, $C_p=4.18\text{kJ/kgK}$	-0.206
<i>sprinkler screen</i>	-0.80kg/s, 60°C, $C_p=4.18\text{kJ/kgK}$	-0.134
<i>heat through conveyor hull</i>	-<0.001MW	-0.001
<i>steam to condenser</i>	-0.13kg/s, 75°C, $C_p=1.90\text{kJ/kgK}$, $H_{\text{evap}20^\circ\text{C}}=2.45\text{MJ/kg}$	-0.345
<i>total</i>	-1.54kg/s water and -0.3kg/s solids	-0.71

5.3 Sensitivity analysis

5.3.1 Heat loss caused by the sprinkler installation

Fig.5.9 shows a hypothetically ideal temperature profile inside the large conveyor, together with results from a simulation. The simulated temperature decreases in the first part of the conveyor and rises in the second part, which is according to plan. A large temperature drop, however, is visible at the end of the first conveyor. It demands an explanation.

The temperature drop can be ascribed partly to the low temperature of the sprinkler water compared to the temperature of the section. The calculation below shows that the standard sprinkler water supply (2kg per section) contributes to five degrees in this way.

$$\Delta T_{balls} = - \frac{m_{sprinkler\ water} \cdot C_{p\ water} \cdot \Delta T_{water}}{m_{balls} \cdot C_{p\ steel}} = - \frac{2\text{ kg/s} \cdot 4180\text{ J/kgK} \cdot (60^\circ\text{C} - 40^\circ\text{C})}{66\text{ kg} \cdot 500\text{ J/kgK}} = -5^\circ\text{C}$$

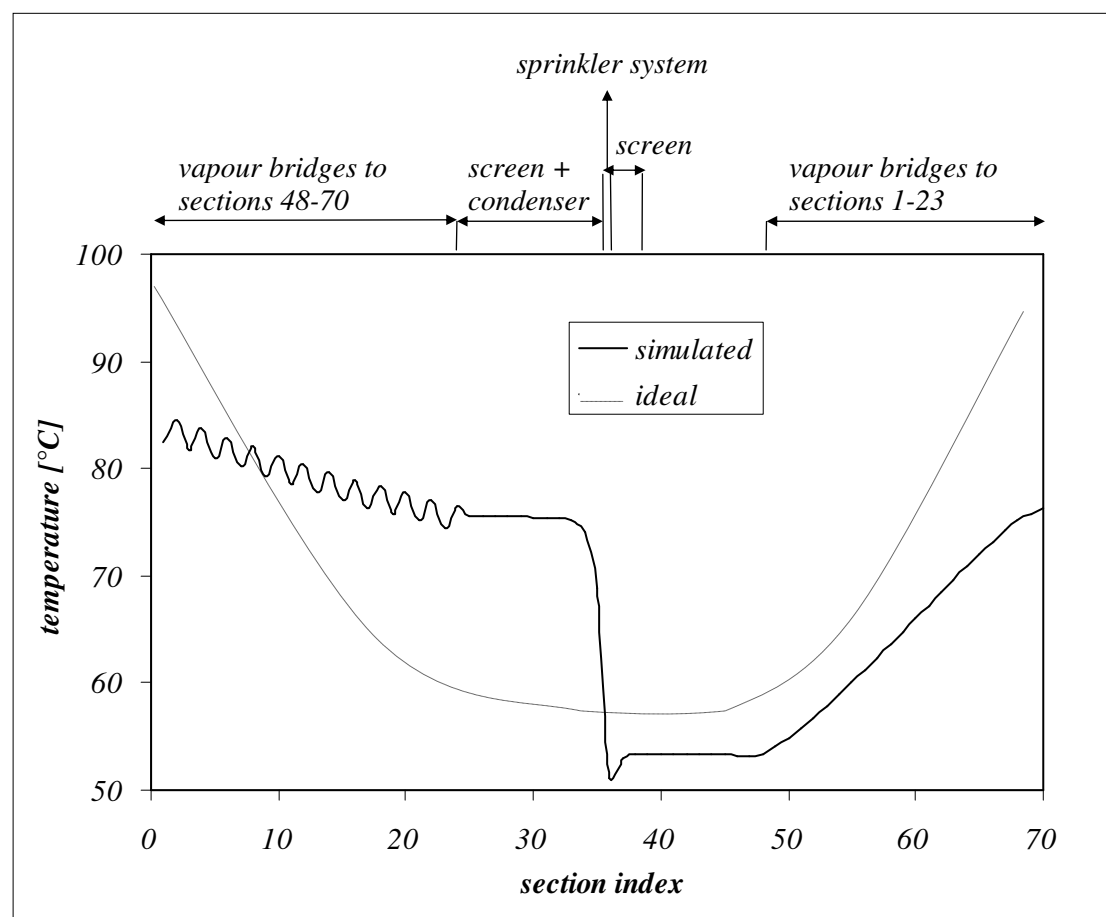


Figure 5.9: Simulated temperature profile and optimum temperature profile in the screw conveyor. The simulated data refer to conditions at the end of a revolution of the screw. Note the positions of the vapour bridges, screens, condenser, and sprinkler installation.

The temperature drop is thus not just the result of the heat capacity and low temperature of the sprinkler water. In fact, it is merely owing to the combined presence of liquid in section 36 and connections from sections 27-35 to the condenser. This is explained as follows.

At the time the balls pass the centre of the simulated screw conveyor, virtually all moisture has disappeared via the vapour bridges, to the condenser, and together with the discharged product. In the absence of water, the contents of the sections do not cool off any further. The local vapour pressure decreases to the free vapour pressure belonging to the temperature of the condenser, which is considerably lower than the temperature of the balls. As the contents shift to section 36 and pass underneath the water shower, part of the liquid flashes and flows via adjacent sections to the condenser (Fig.5.10). The evaporating sprinkler water takes away heat from the balls, making them cool off significantly.

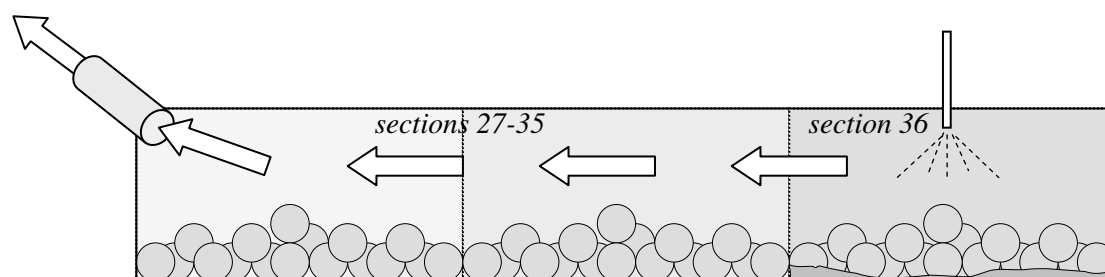


Figure 5.10: Explanation of the main cause of the temperature drop in section 36. Part of the sprinkler water evaporates from the balls and escapes to the condenser.

Considering the cause of the temperature drop, there are several ways thinkable of raising the energy economy of the installation. One of the things that come to mind is to reduce the capacity of the condenser. This, however, may not be a good idea as it would result in lower temperature gradients around the last evaporation stages (the stages that exhibit small driving forces already), making it more difficult to keep the moisture content of the dried product at a constant low level.

Another option is to trim down the sprinkler water supply. Fig.5.11 shows that a reduction from 1kg/s to 0.5kg/s does not bring substantial improvements. Only small changes to the temperature profile are observed, which is understandable as an evaporation rate of 0.1kg/s already corresponds with a temperature drop of 15 degrees. The amount of water present still suffices for continuous evaporation.

A different situation develops if the sprinkler water supply is fully closed. The temperature drop vanishes together with the vapour leaks from the sprinkler system to the condenser, and the temperature profile becomes much more favourable than before. It would thus be advantageous to remove the sprinkler system from the drawing board. It is uncertain however, if the sprinkler can be left out in practise, since dirt travelling along with the balls could cause fouling problems in equipment further down the line. It must be verified whether the recovery of dried material from the balls is large enough to make the sprinkler installation superfluous, and if any alternatives are worth considering. One of these alternatives is to remove the sprinkler installation to a place further away from the condenser, which is discussed in Section 5.3.2.

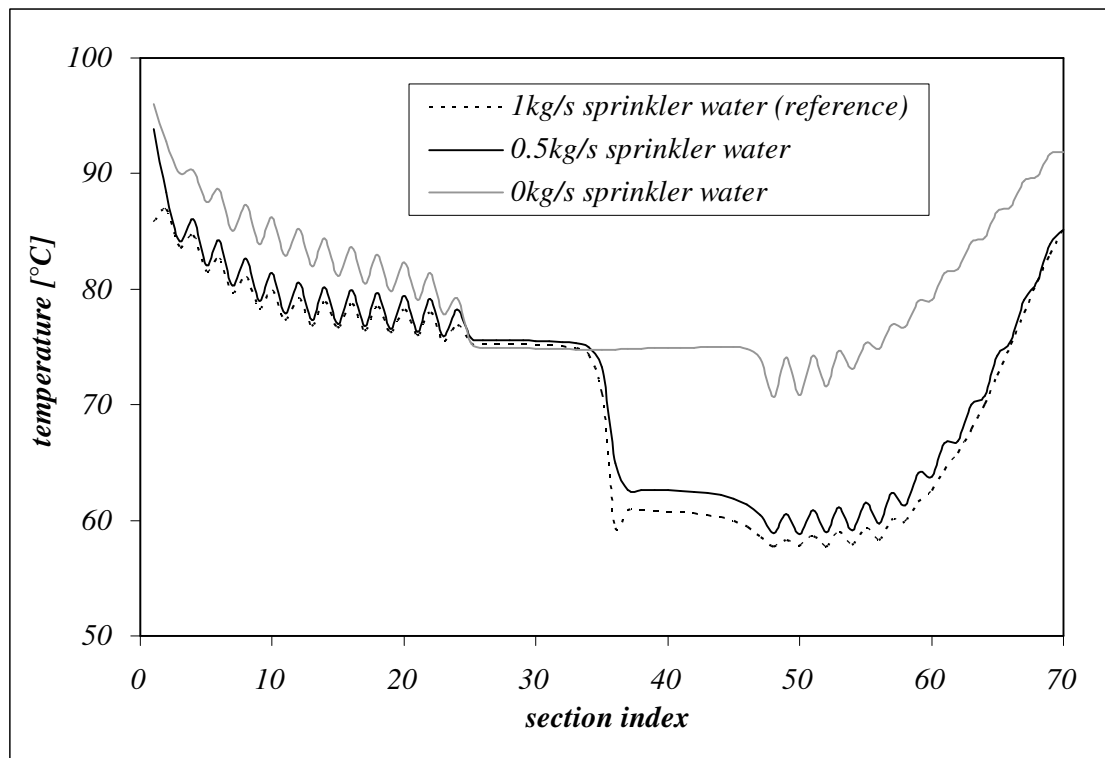


Figure 5.11: Temperature as a function of section number for three simulations. The input parameters for the simulations were identical, except for the sprinkler water supply.

5.3.2 Vapour leaks along axial direction of the screws

The permeability of the ball bed and the gap between the screw blades and the conveyor hull allow vapour leaks to the condenser. A brush between the blades and the hull increases the resistance to vapour flows, thus improving drying efficiencies. The same holds for an increased number of ball bed segments between sections connected to the condenser and sections containing sprinkler water. According to the simulation results in Fig.5.12, both a brush and a number of added pitches result in reduced heat losses. They yield higher ball temperatures after the last condensation stage.

The data in Table 5.3 show effects of various combinations of brushes, sprinkler water supplies, and increased distances between the sprinkler installation and the condenser. Whereas a normal brush reduces energy consumptions from 2.75MJ per kg of evaporated moisture to 2.0MJ/kg, a very dense brush in combination with four added sections appears to bring this value down to 1.7MJ/kg.

A brush is beneficial at all times, also in case no sprinkler water is used. This is because the vapour flow that dissipates to the condenser partly stems from released product vapour. Even if the losses to the condenser were for some reason unaffected, a brush would raise thermal efficiencies just by reducing vapour exchanges between the stages of the MSFE-process (see Paragraph 3.2).

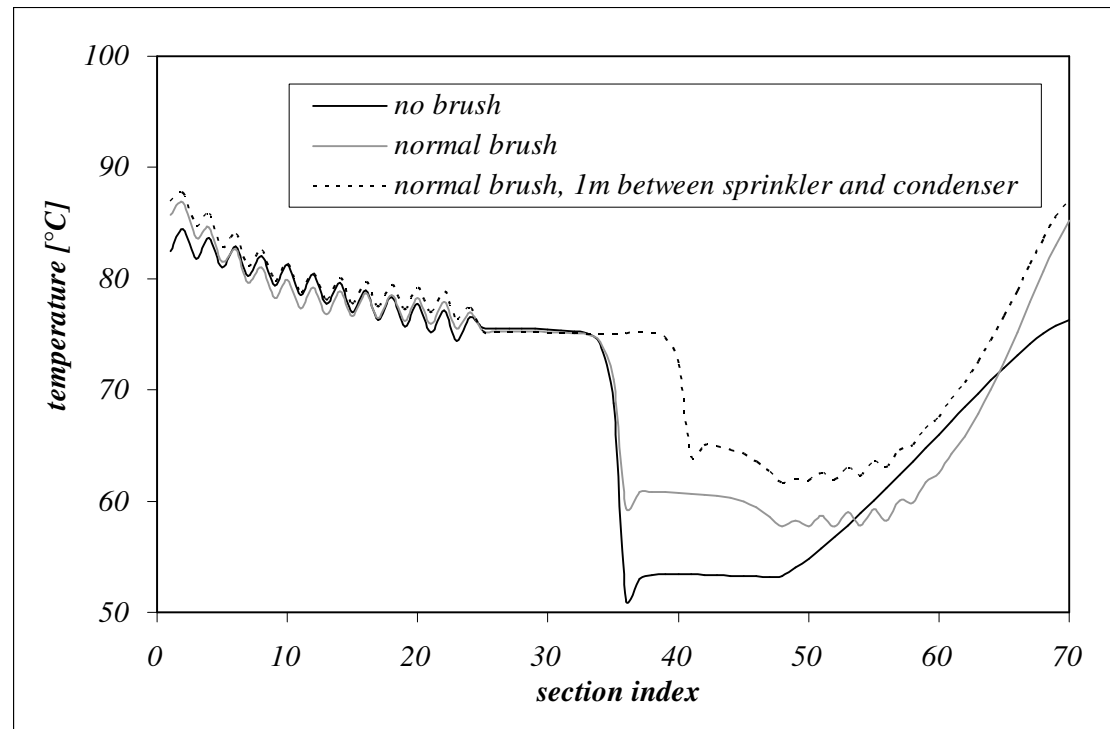


Figure 5.12: Temperature profiles for three simulations, showing the influence of a brush between the screw blades and the hull, as well as the effect of an increased distance between the condenser and the sprinkler installation.

Table 5.3: Specific energy consumptions and final moisture contents for simulations with different sprinkler water supplies, brush types, and distances between the condenser and sprinkler installation. The moisture content of the dried product is on a wet basis.

sprinkler water supply [kg/s]	sections between condenser and sprinkler	brush type	final moisture content [%]	energy per kg of removed water [MJ/kg]
1	1	none	9.5	2.75
1	1 (reference)	normal	9.7	2.01
1	1	dense	9.4	1.90
1	5	none	9.8	2.58
1	5	normal	9.5	1.82
1	5	dense	9.5	1.70
0.5	1	none	9.6	2.68
0.5	1	normal	9.8	1.94
0	1	none	10.4	2.26
0	1	normal	10.8	1.36
0	1	dense	10.8	1.15

5.3.3 Air leaks from outside

As the drying process takes place within a limited amount of time, it relies on high vapour flow rates. The necessary pressure gradients correspond with vapour pressure gradients as long as no other gasses are present. However, they may be levelled off if air has penetrated the installation. The air will flow along with the vapour, will not condense onto the metal balls, and will build up at the points of lowest pressure.

A vacuum pump positioned near the condenser is to remove all noncondensables that show up inside the installation. Besides, the installation should be well sealed in order to restrain air leaks. This is not too difficult for the largest part of the apparatus, since in essence, it is a closed transport system for metal balls and is at some places even slightly above atmospheric pressure. Two areas require special attention though: the feed entrance and the discharge end. Volatiles that may form during the treatment of organic cake should be taken care of as well.

The effect of noncondensables was investigated on the basis of simulation results. Some of those results are presented in Table 5.4 and in the graph of Fig.5.13. They reveal that the thermal efficiency of the drying process does not depend strongly on air leaks around the discharge area. Leak streams as large as 5g/s do not leave a trace as the air is removed almost straightaway via the condenser and vacuum pump. The discharge area is thus of no particular concern, since it does not take more than a standard sluice system (e.g. comprising two pneumatically operated slide valves) to reduce the leak streams below the simulated values.

Air leaks near the feed entrance are more problematic than at the discharge area. Gas that enters here builds up in sections 48 to 60, which has to do with the extended route to the condenser and the vacuum pump (see Fig.5.14). The gas flows from the first sections via vapour bridges to the second conveyor. It moves ahead to adjacent sections of lower pressure, and accumulates at places where pressure gradients are small. For the investigated simulations, the build-up in the second conveyor raises the local pressure in excess of the pressure in facing sections of the first conveyor. Hence, the gas makes a detour via the last vapour bridges before reaching the vacuum pump, which explains the peak values near section 20 in Fig.5.13.

Table 5.4: Simulation results, showing final moisture contents and energy consumptions at different air leaks. The data of the first row relate to the reference case.

<i>air leaks from outside</i>	<i>final moisture content [%]</i>	<i>energy per kg of removed water [MJ/kg]</i>
<i>0,1g/s at feed entrance (first section)</i>	9.7	2.01
<i>0,1g/s at feed entrance + 5g/s at output (2,5g/s at both sections 25 and 35)</i>	9.5	2.01
<i>no air leaks</i>	9.7	1.99
<i>0,5g/s at feed entrance (first section)</i>	9.7	2.04

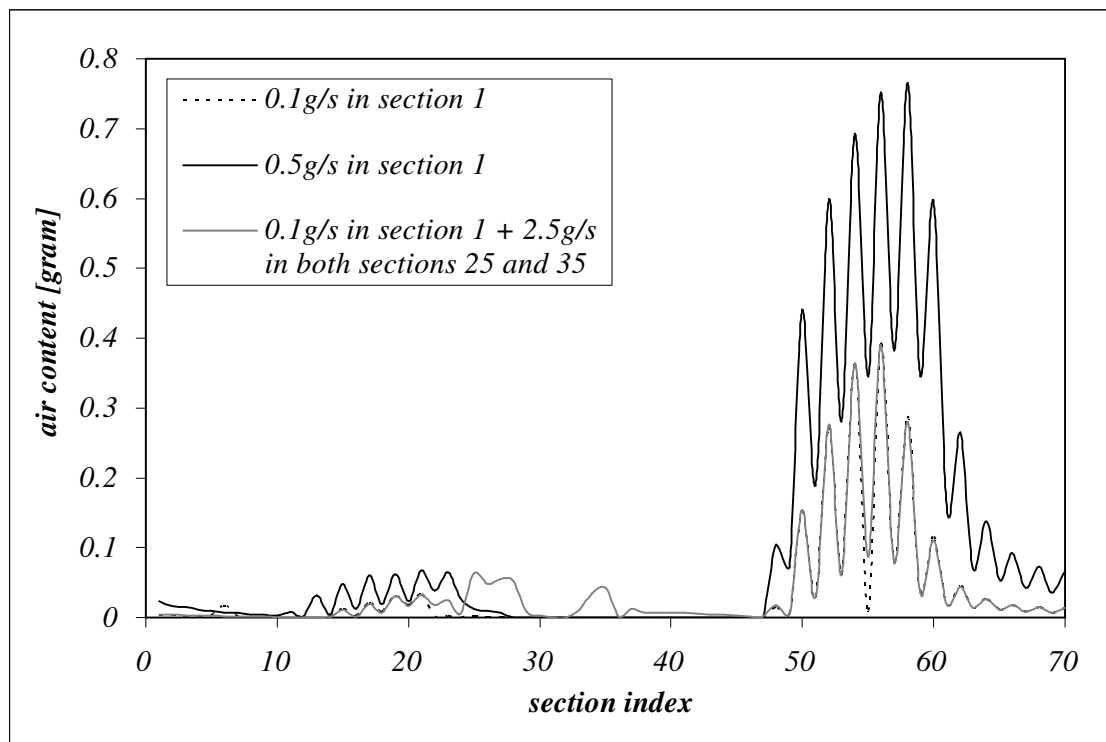


Figure 5.13: Air contents of the sections for simulations with different air leaks.

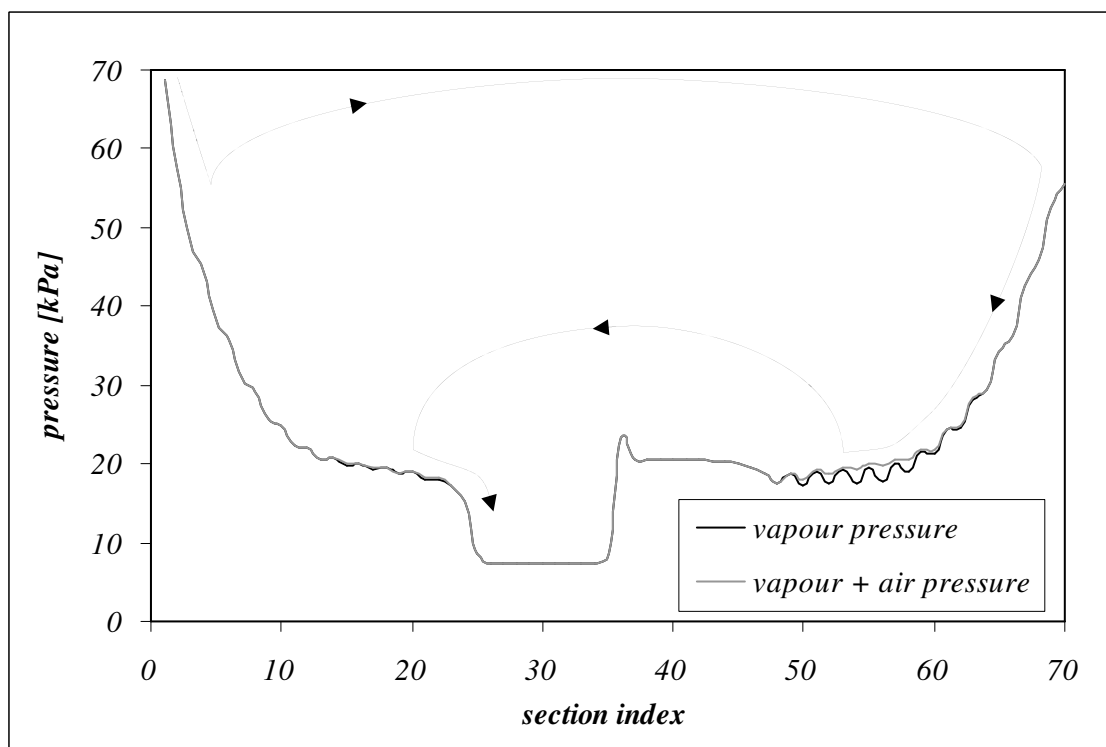


Figure 5.14: Average vapour pressure and total pressure inside the screw conveyors of the reference simulation. The arrows indicate the flow path from the feed area to the vacuum pump.

5.3.4 Thermal efficiency vs. the number of vapour bridges

A relationship between the number of flash stages and the thermal performance of an MSFE-process was presented in Chapter 3. According to the simple model, the efficiency of the process would rise further and further with every additional stage. The minimum energy consumption for 12 ideally operating evaporation steps proved to be 0.2MJ per kg of evaporated water, although it increased rapidly to 0.6MJ/kg when taking into account heat losses due to input and output streams. This latter value, however, is still significantly lower than the energy consumptions predicted by the local equilibrium model.

Simulations were performed with different numbers of vapour bridges, enabling further comparisons between the very simple model and the local equilibrium model. Results are presented in Table 5.5 and Fig.5. Clearly, the process efficiency of the simulated dryer does not improve much in case a larger number of flash stages are used. A point of saturation is reached around 10 vapour bridges, making the contributions of additional bridges even negligible.

Table 5.5: Results for various numbers of vapour bridges and conveyor lengths. The first row presents data for the reference simulation.

<i>number of vapour bridges</i>	<i>number of sections</i>	<i>final moisture content [%]</i>	<i>energy consumption [MW]</i>	<i>energy per kg removed water [MJ/kg]</i>
12	70	9.7	0.540	2.01
8	54	9.8	0.548	2.04
4	38	10.0	0.601	2.25
2	30	10.6	0.686	2.58
1	26	11.2	0.787	2.99

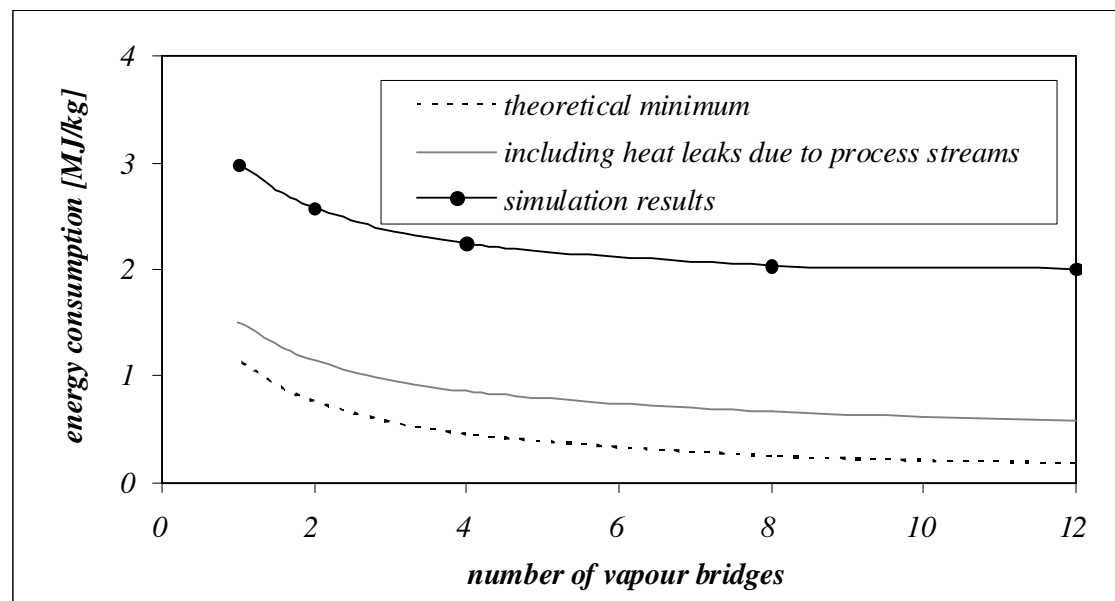


Figure 5.15: Three relationships between specific energy consumptions and the number of flash stages.

According to Fig.5.15, the length of the simulated conveyors could be reduced without compromising the overall thermal performance of the installation. This conclusion is supported by the relatively low pressure gradients at the end of the first conveyor and in the first part of the second conveyor (Fig.5.6). Whether it also holds in reality depends on the kinetics of the evaporation and condensation steps. The local equilibrium model is based on infinite kinetics. Compared to its results, the actual drying process will be slower (see Paragraph 6.2). It will extend over more sections, and may require somewhat longer screw conveyors.

5.3.5 Hygroscopic properties of the cake

The smaller contributions of successive vapour bridges are partly due to the fact that the free vapour pressure progressively rises with temperature. Equal temperature gradients at lower absolute temperature result in lower pressure gradients, and thus in smaller driving forces for vapour transports. The hygroscopic nature of the cake solids adds to this effect by reducing the equilibrium vapour pressure once the material has been dried beyond critical moisture contents. Fig.5.16 clarifies this. It shows the pressure and temperature differences between sections that interconnect via vapour bridges. Although the temperature gradients over the last bridges are actually highest, the pressure gradients are particularly small.

Apparently, the combination of low temperatures and low moisture contents is detrimental to the final stages of the drying process; especially as these stages will also have to deal most with the presence of air, which is an aspect that has not even been considered here.

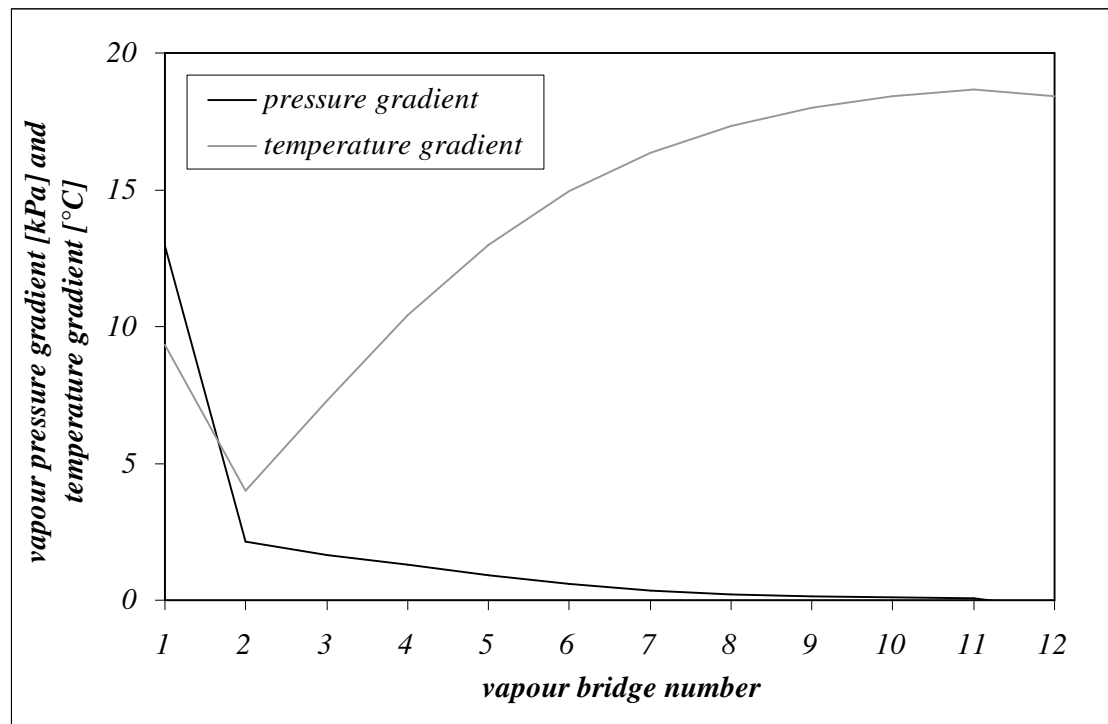


Figure 5.16: Pressure and temperature gradients over the vapour bridges. The data are for the settings of the reference simulation, yet without air leaks.

The local equilibrium model uses the parameter $Hygr$, as defined in Paragraph 4.2, to describe the hygroscopic nature of the cake. Fig.5.17 presents moisture contents of the cake and temperature profiles inside the screw conveyor for different values of $Hygr$.

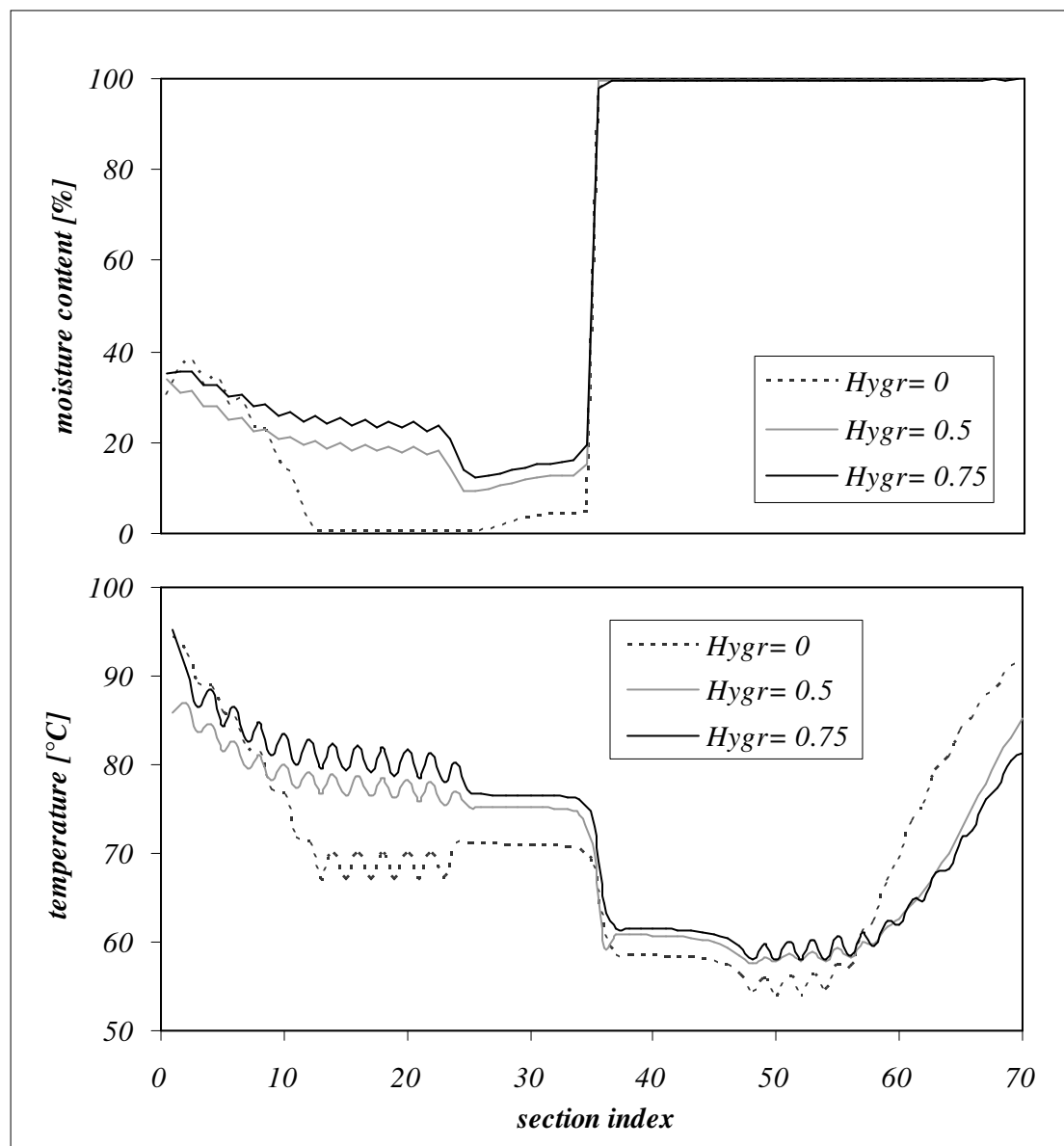


Figure 5.17: Temperatures and moisture contents of the cake as function of section number, for different values of $Hygr$.

For $Hygr=0.5$ and $Hygr=0.75$, the drying process appears to be confined to the first 12 sections of the conveyor, i.e. to the first six evaporation stages. The initial drying rate is high. As the process continues, it decreases more and more, resulting in moisture contents of 20-25% around the last stage. Fortunately, the condenser removes part of the remaining water, so that the final moisture content of the product is acceptable. The vapour to the condenser, however, is lost. The latent heat involved cannot be

recovered in the MSFE-process and thus contributes directly to the total energy consumption of the dryer.

According to Fig.5.17, feed types with $Hygr=0$ fully dry up within the first six evaporation stages. The contributions of following stages are negligible here as well. Half the number of vapour bridges is left unused, which is unfavourable for the efficiency of the MSFE-process. Strangely enough, the situation establishes automatically and does not improve after a long time.

Fig.5.18 presents final moisture contents of the product as a function of energy consumptions, for different values of $Hygr$. The graph includes drying curves for the reference simulation ($Hygr=0.5$) and for $Hygr=0.1$. The hygroscopic nature of the cake appears to be a major factor determining the efficiency of the drying process. Poorly hygroscopic feeds dry up with less energy and to higher solids contents. It must therefore be stressed that the hygroscopic properties used as input for the reference simulation were roughly overestimated. Laboratory experiments (Paragraph 4.2) showed that realistic values for $Hygr$ are around 0.1 instead of 0.5. Energy consumptions may therefore be lower than predicted so far: according to the simulation results, values as low as 0.9MJ per kg of evaporated water could come within reach, shifting the upper line in Fig.5.15 downward.

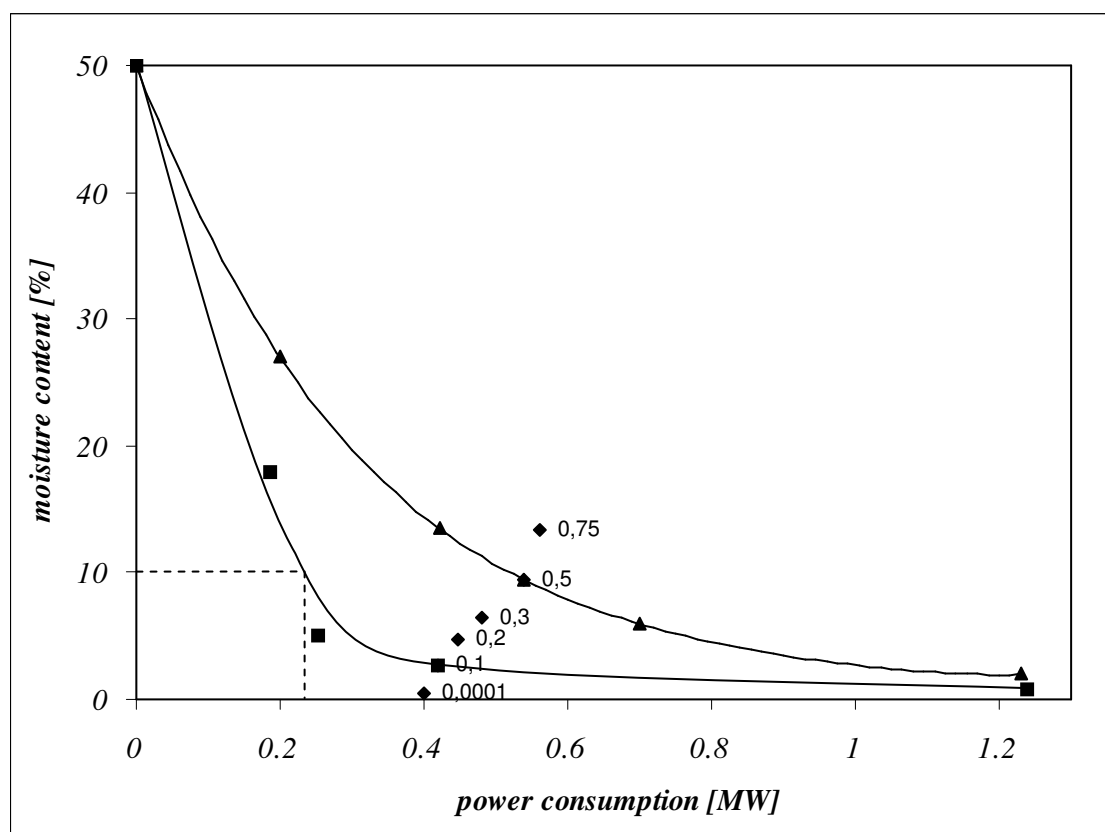


Figure 5.18: Moisture contents vs. power consumptions for feed types with different hygroscopic properties. The drying curves for the reference simulation and for $Hygr=0.1$ were included.

5.3.6 Input of solids and water as part of the feed

An important variable is the initial moisture content of the cake. The drying treatment obviously costs more energy if the feed contains more water. However, it requires less energy per kg of evaporated water if the final moisture content reaches values below the hygroscopic threshold, since the liquid is on average more weakly bound. Fig.5.19 shows drying curves for three moisture contents of the feed. Such curves enable the operator of a drying plant to react adequately to fluctuations in the feed.

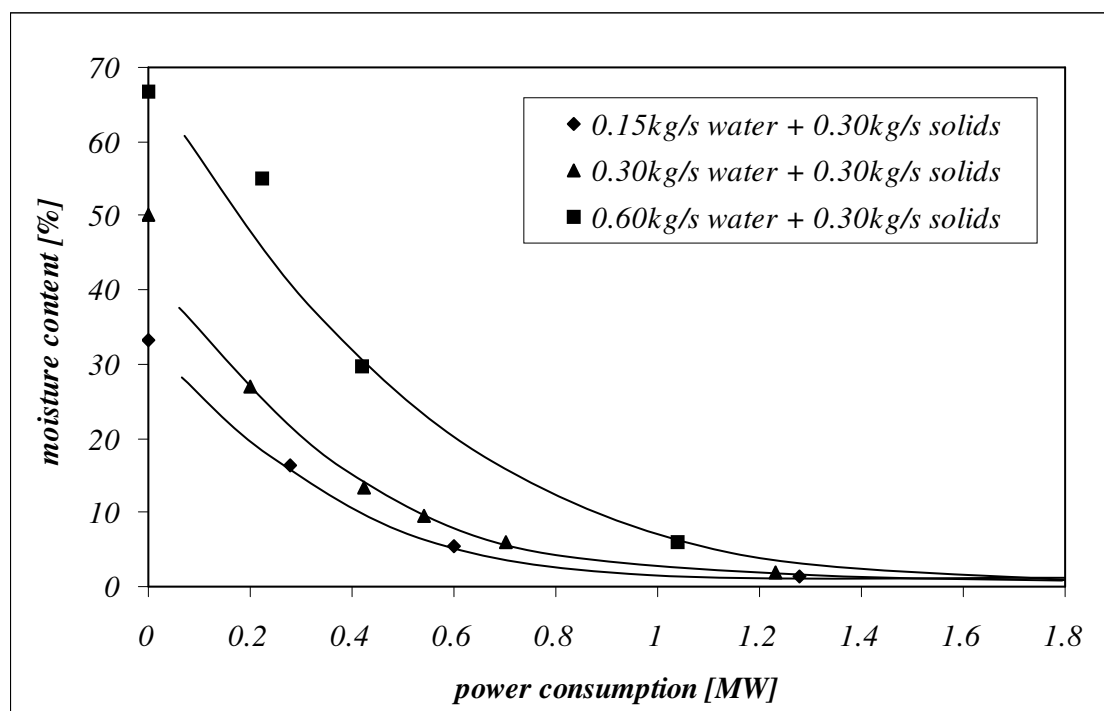


Figure 5.19: Moisture content vs. power consumption for different moisture contents of the feed.

The steam supply, the revolution speed of the screw conveyors, and the amount of wet cake entering the installation per unit of time are the principle control parameters for the drying process. They can be adjusted to optimise the relationship between energy consumption, final moisture content, and drying capacity. The volume of the ball bed is not freely variable. It should always be as large as possible.

Fig.5.20 shows effects of different feed rates on energy consumptions. The horizontal axis indicates energy consumptions per kg of feed, to enable fair comparisons between the drying curves. The drying curves for higher feed rates are clearly more favourable since the leak streams through the ball bed and the gaps between the screw blades and the hull are less dominant. Increased cake-balls ratios also allow the balls to transfer more heat and cool off further in the first part of the screw conveyor, which helps reducing unnecessary losses to the condenser (Fig.5.21). Consequently, more vapour condenses onto the balls in the second part of the conveyor, so that the final ball temperature after the last condensation step is higher.

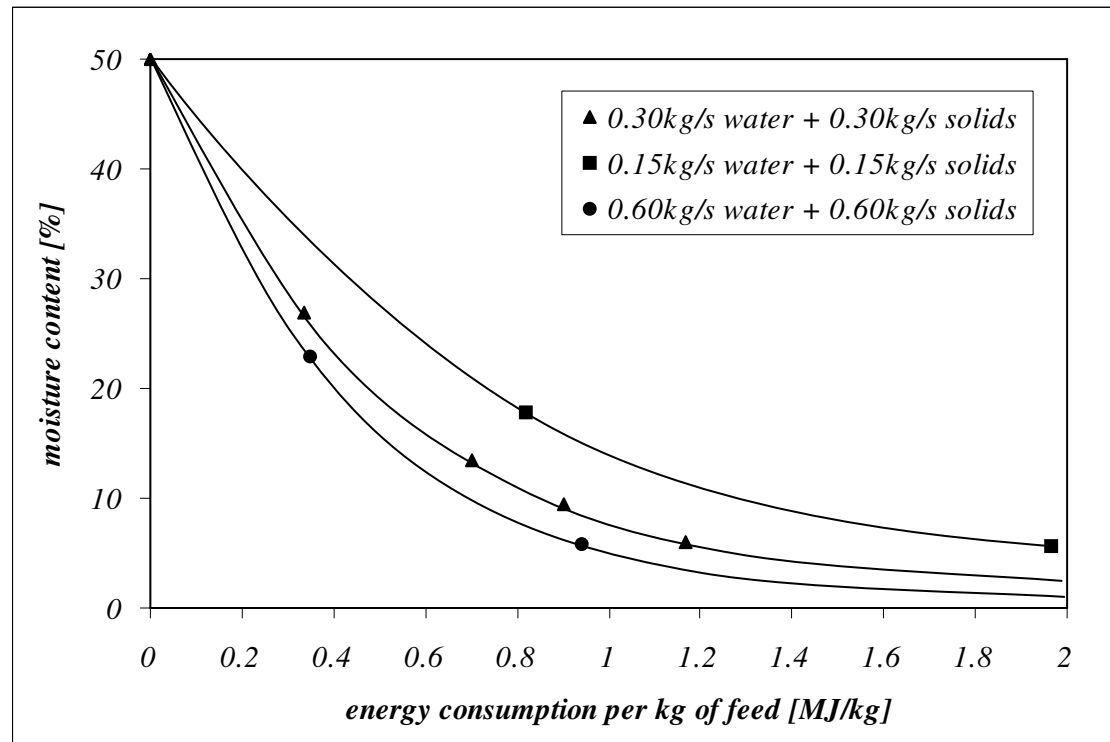


Figure 5.20: Moisture content vs. energy consumption per kg of feed, for different feed rates.

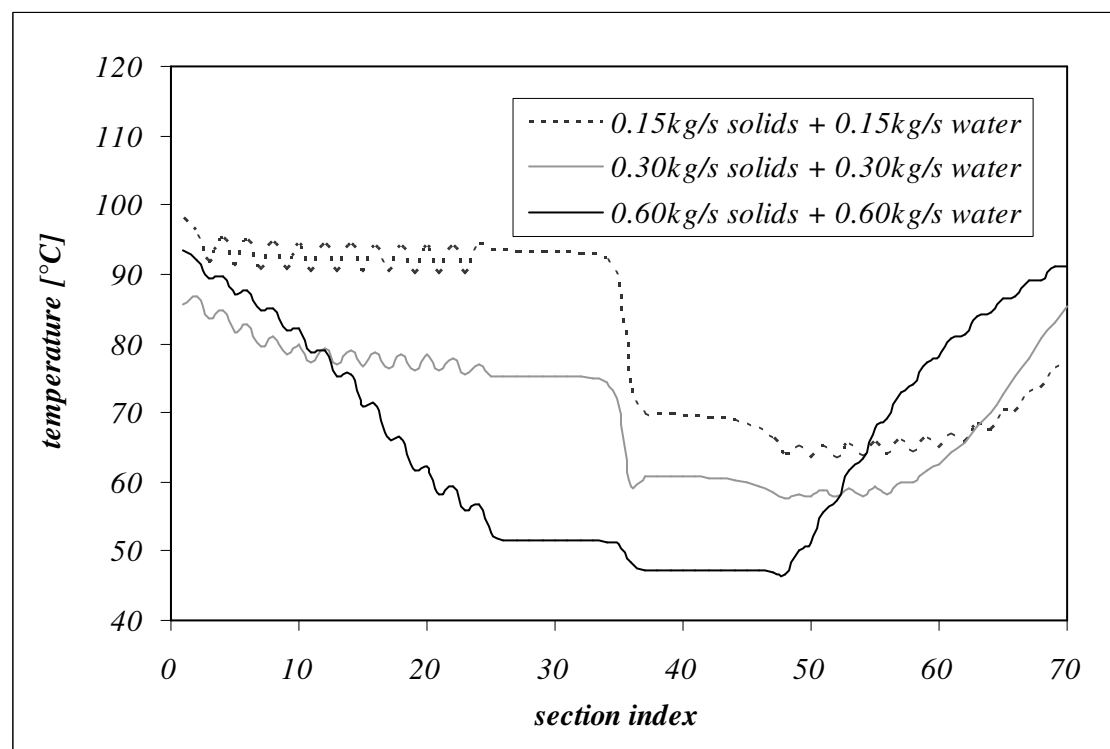


Figure 5.21: Temperature as function of the position in the screw conveyors, for different feed rates. Feed rates of 0.15 kg/s, 0.3 kg/s and 0.6 kg/s yield finished products with moisture contents of 6%, 10% and 23% respectively.

It is thus more attractive to increase feed rates than to trim down external steam supplies in case the cake tends to dry easily below the required moisture content of the finished product. The equilibrium model is unfortunately not detailed enough to explore maximum drying capacities, since it does not take into account important rate-determining factors, i.e. the kinetics of the evaporation and condensation processes. However, it is clear that a certain amount of heat is doomed to be lost at the end of the first conveyor in order to generate a pressure difference between sections of the first and second conveyor. The drying process would otherwise become too slow or not even start at all in the first place.

Chapter 6: Process kinetics within the steam vessel and flash stages

The local equilibrium model and accompanying simulation software enable the user to forecast drying conditions for various feed properties, design specifications, and operational parameters. Although the model makes allowance for many phenomena that take place in the core of the process, it does not properly address every important aspect. The accuracy and relevance of the simulation results, including those presented in Chapter 5, are therefore limited.

Two shortcomings in particular may reduce the reliability of quantitative performance predictions.

- The balls are supposed to heat up instantly to the steam temperature as they enter the steam vessel. No attention is paid to the kinetics of the condensation process during the short residence in the vessel. It is neither questioned how and to what extent the balls get rid of the condensate before they are mixed again with fresh feed.
- The local equilibrium model divides the screw conveyors into relatively large sections and focuses on heat and mass flows entering and leaving them. It does not take notice of transport mechanisms within a section. The kinetics of these processes, however, is finite. It will affect the overall performance of the dryer.

The two paragraphs of this chapter address these shortcomings. They reveal the errors that were introduced by ignoring them. The structure of both paragraphs is the same. They outline the area of investigation, concentrate on separate aspects, and then combine the insights in order to predict the total effect on the efficiency of the drying process.

6.1 Heating and dewatering inside the steam vessel

6.1.1 Description of the heating process

The steam vessel is a separate unit of the Delta dryer, shielded from the rest of the installation by means of sluices (Fig.6.1). It brings the balls in contact with external steam supplies, heating them up to their ultimate temperature. An inclined dewatering screen removes most of the condensate and carries it away to a float controlled valve. For proper functioning, the balls should run as a monolayer over the screen as the steam expands from a slit and forms a jet crossing the balls stream. The pressure of the jet may be able to remove most of the condensate as well as residual solids. If not, the rotational velocity of the balls will create a centrifugal force that should be sufficient to reduce the water film to an acceptable thickness. It is important to take into account the amount of water moving along with the balls to the lower rotary sluice, since it moistens the feed in the successive mixing operation: it represents a heat leak.

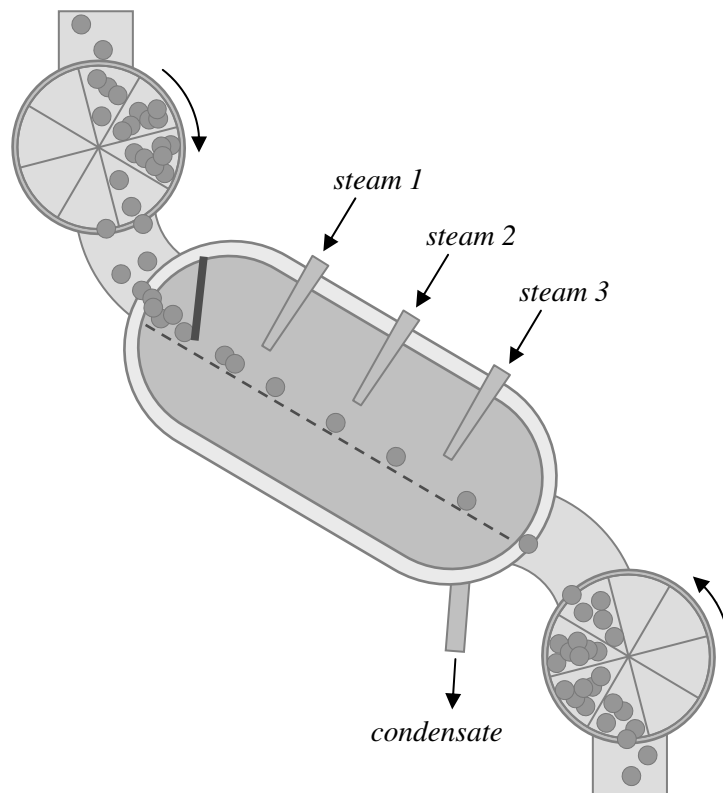


Figure 6.1: Schematic cross-section of the steam vessel and rotary sluices. The balls enter the vessel via a rotary sluice. They tumble down, collide with a rubber sheet, and start rolling down a dewatering screen. Meanwhile, steam expands from one of three nozzles. The lower rotary sluice lets the balls exit the scene.

The heating process starts as the balls leave the upper sluice and enter the steam atmosphere. The pressure inside the vessel is above atmospheric pressure and highest in the sequence of operations. Hence, there is no air present. The steam temperature

(T_{steam}) is uniform. The film of condensate and the steel mass of the balls form the heat resistances between the steam atmosphere and the heat capacity of the balls. The heat flux through the water film into a ball is given by the left part of Equation 6.1, in which convection inside the water layer has been neglected. T_{balls} is the volume average temperature of the balls, and T_{film} is the surface temperature of the liquid film in degrees Celsius. The effective thickness of the ball is $D_{ball}/10$ [Wesselingh, 1990]. The amount of heat conducted into the ball equals the enthalpy gain corresponding with the temperature rise of the ball. It also closely approximates the enthalpy loss of the condensing steam, since the heat capacity of the thin water layer is small. The condensate formed increases the average thickness of the water film ($\bar{d}_{film}$).

$$\frac{\pi D_{ball}^2 (T_{steam} - T_{balls})}{\frac{D_{ball}}{10\lambda_{steel}} + \frac{d_{film}}{\lambda_{liquid}}} \approx \frac{1}{6} \pi \cdot D_{ball}^3 \cdot \rho_{steel} \cdot Cp_{steel} \cdot \frac{\partial T_{balls}}{\partial t} \quad [6.1]$$

$$\approx \left(H_{evap, 0^\circ C} + Cp_{vapour} \cdot T_{steam} - Cp_{water} \cdot T_{film} \right) \cdot \pi D_{ball}^2 \cdot \rho_{liquid} \left(\frac{\partial \bar{d}_{film}}{\partial t} \right)_{condensation}$$

The thermal resistance of a 20µm thick water film is larger than of a steel ball with a diameter of 12mm. The thickness of the water film thus strongly acts upon the heating process. The next paragraphs elaborate on the centrifugal dewatering effect and the dewatering effect by the steam jet, assuming film-type condensation. The models are combined with Equation 6.1 to predict the ball temperature and thickness of the water film as function of the position down the screen.

6.1.2 Removal of condensate by centrifugal forces

While the balls are running down the dewatering screen, the forces acting on the water film are gravitational and centrifugal. The centrifugal forces depend on the angular velocity of the balls (ω), as formulated in Equation 6.2. The angular velocity increases linearly with time, provided that the balls roll down the screen without slipping and bouncing, and provided that the aerodynamic drag and roll friction acting on them are negligible. When comparing the centrifugal and gravitational forces for a representative screen inclination angle and ball diameter, gravity appears to be insignificant after the first tenth of a second. The residence time of the balls in the steam vessel is much longer, which is why the gravitational dewatering effect is left out of consideration in the further discussion.

$$a_{centrifugal} = \frac{\omega^2 \cdot D_{ball}}{2}, \text{ with: } \frac{\partial \omega}{\partial t} = \frac{10}{7} \frac{g \cdot \sin \alpha_{screen}}{D_{ball}} \quad [6.2]$$

Fig.6.2 shows a schematic front view and side view of a wet ball revolving around its axis. The centrifugal force moves the liquid towards the equator, where it splashes off. The local flow velocity in the water film depends on the distance from the ball surface, the film thickness, and the component of the centrifugal force parallel to the spherical surface of the ball. A uniform velocity profile exists within an infinitely thin, ring-shaped film segment with constant distance to the rotation axis. The angle φ , as indicated in the drawing of Fig.6.2, locates this segment.

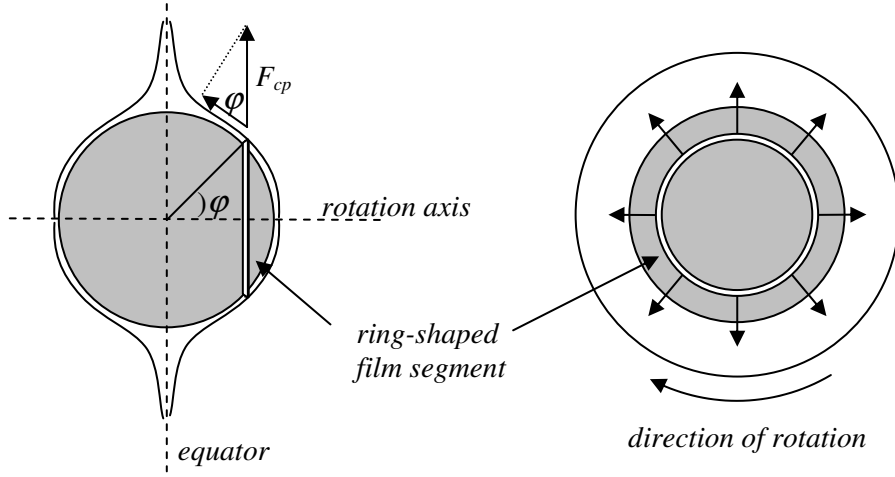


Figure 6.2: Front view (left) and side view (right) of a rotating wet ball. The centrifugal force is uniform within a ring-shaped segment of the liquid film.

The velocity profile in the ring-shaped film segment is comparable to the parabolic velocity profile for fully developed laminar flow past a vertical plate of infinite dimensions. The mean velocity of the liquid can therefore be expressed analogously to this well-known case, with the acceleration due to gravity (g) being replaced with the component of the centrifugal acceleration parallel to the ball surface ($\omega^2 \cdot D_{ball} \cdot \sin \varphi \cdot \cos \varphi / 2$). Equation 6.3 gives this relationship. η_{liquid} is the dynamic viscosity of the liquid, which decreases with temperature.

$$\bar{v}_{\varphi} = \frac{d_{film,\varphi}^2}{3 \cdot \eta_{liquid}} \cdot \frac{\rho_{liquid} \cdot \omega^2 \cdot D_{ball} \cdot \sin \varphi \cdot \cos \varphi}{2} \quad [6.3]$$

If the centrifugal dewatering effect is substantially large, the liquid film will be thin and its mass will be much smaller than the total steam mass required to heat the ball. This means that the net volume change of the water film is small compared to the amount of condensate formed, or, in other words, that the dewatering rate approximates the condensation rate.

A thin liquid film also implies that its heat resistance is relatively small. If, by approximation, the ball heats up uniformly, the condensate is evenly supplied to the whole surface. The volume of water leaving the film segment then equals the volume entering it plus the volume of condensate that forms onto the surface of the film segment (Equation 6.4). C is the volume of condensate produced per unit of time and surface area. It approximates $\left| \partial \bar{d}_{film} / \partial t \right|_c$ caused by centrifugal forces, since the condensation rate and dewatering rate are almost equally large. Integration and substitution of Equation 6.3 yields Equation 6.5.

$$\partial (\bar{v}_{\varphi} \cdot \pi \cdot D_{ball} \cdot \sin \varphi \cdot d_{film,\varphi}) = \frac{C \cdot \pi \cdot D_{ball}^2 \cdot \sin \varphi \cdot \partial \varphi}{2} \quad [6.4]$$

$$d_{film,\varphi} = \sqrt[3]{\frac{3\eta_{liquid} \cdot C}{\rho_{liquid} \cdot \omega^2} \cdot \frac{1 - \cos \varphi}{\sin^2 \varphi \cdot \cos \varphi}} = \sqrt[3]{\frac{3\eta_{liquid}}{\rho_{liquid} \cdot \omega^2} \cdot \left| \frac{\partial \bar{d}_{film}}{\partial t} \right|_c \cdot \frac{1}{(1 + \cos \varphi) \cos \varphi}} \quad [6.5]$$

The summed volume of all infinitely thin water film segments equals the surface area of the ball times the average film thickness $\bar{d}_{film}$.

$$2 \int_{\varphi=0}^{\varphi=\pi/2} \frac{\pi \cdot D_{ball}^2 \cdot \sin \varphi}{2} \cdot \sqrt[3]{\frac{3\eta_{liquid}}{\rho_{liquid} \cdot \omega^2} \cdot \left| \frac{\partial \bar{d}_{film}}{\partial t} \right| \cdot \frac{1}{(1 + \cos \varphi) \cos \varphi}} \cdot \partial \varphi = \pi \cdot D_{ball}^2 \cdot \bar{d}_{film} \quad [6.6]$$

It gives the following correlation between $\partial \bar{d}_{film} / \partial t$, ω and $\bar{d}_{film}$ (Equation 6.7). Note that part of the water that is sprayed from the equator of the balls may hit the surface of other balls. The factor f has therefore been introduced as the fraction of removed water that reaches the float controlled valve. The rest, $(1-f)$, is initially removed, but subsequently taken up by other balls.

$$\frac{\partial \bar{d}_{film}}{\partial t} \approx \left(\frac{\partial \bar{d}_{film}}{\partial t} \right)_{condensation} - f \cdot \frac{2}{15} \cdot \frac{\omega^2 \cdot \bar{d}_{film}^3}{v_{liquid}} \quad [6.7]$$

6.1.3 Removal of condensate by the steam jet

At a certain distance down the dewatering screen, the rolling balls contact the steam jet. This steam jet expands from a slit covering the full width of the screen. Its dimensions allow the balls to make a single rotation while its momentum splashes off part of the condensate. Because of the brief contact, a negligible amount of condensate forms during the action.

The steam flow past a monolayer of balls is roughly comparable to the flow of an incompressible fluid through a perforated plate. Equation 6.8 gives the accompanying pressure drop for the latter [Perry, 1997]. γ is the fractional free area of the perforated plate, which follows from Equation 6.9. It is a function of the width of the screen (b_{screen}), the velocity of the balls, and the balls throughput (W_{balls}). The superficial gas velocity at close distance to the nozzle (v_{steam}) can be derived from the balls throughput (W_{balls}), the ball temperature at the end of the steam vessel ($T_{balls,final}$), and the dimensions of the steam jet (Equation 6.10).

$$\Delta P \approx 0.7 \cdot \frac{\rho_{gas} \cdot v_{steam}^2}{2} \cdot \left(\frac{1 - \gamma^2}{\gamma^2} \right) \quad [6.8]$$

$$with: \gamma = 1 - \frac{3}{2} \cdot \frac{W_{balls}}{D_{ball} \cdot \rho_{steel}} \cdot \frac{1}{b_{screen} \left(v_{ball,t=0} + \frac{5}{8} g \cdot \sin \alpha_{screen} \cdot t \right)} \quad [6.9]$$

$$and: v_{steam} = \frac{W_{balls} \cdot Cp_{steel} \cdot (T_{balls,final} - T_{balls,t=0})}{\left(H_{evap,0^\circ C} + T_{steam} \cdot Cp_{steam} - T_{balls,final} \cdot Cp_{water} \right) \cdot b_{screen} \cdot d_{nozzle} \cdot \rho_{steam}} \quad [6.10]$$

Fig.6.3 shows a side view and top view of a ball subjected to the steam jet. The force acting on the ball is the pressure drop of the jet multiplied by the projected area per ball. A uniform shear stress τ_θ exists within a small ring-shaped water film segment with thickness $d_{film,\theta}$. The ring has a 90-degree angle to the jet. The shear stress can be calculated from Equation 6.11.

$$\tau_\theta \approx 0.7 \cdot \frac{\rho_{gas} \cdot v_{steam}^2}{2} \cdot \left(\frac{1-\gamma^2}{\gamma^2} \right) \cdot \cos \theta \cdot \frac{1/4 \pi D_{ball}^2}{(1-\gamma)} \cdot \frac{1}{\pi D_{ball}^2} \quad [6.11]$$

The local shear stress causes a linear velocity profile in the liquid film. The mean velocity of the water film segment is:

$$v_\theta = \frac{\tau_\theta \cdot d_{film,\theta}}{2\eta_{liquid}} \quad [6.12]$$

The thickness of the film as function of the global coordinates of the ball is not exactly known. A ring-shaped film segment at a more or less arbitrarily chosen position is therefore taken as a reference. The local shear stress in that film segment is assumed to approach the average shear stress; the local thickness is assumed to come close to the average thickness of the film; and the liquid volume leaving the segment is supposed to be representative for the volume change of the liquid film. Combination of Equations 6.11 and 6.12 yields Equation 6.13 for the ring depicted in Fig.6.3.

$$\left(\frac{\partial d_{film}}{\partial t} \right)_{jet} \approx - \frac{d_{film,\theta} \cdot v_{\theta=45^\circ} \cdot \cos \theta}{D_{ball}} \quad \cos \theta = 0.7 \quad \approx - \frac{\rho_{gas} \cdot v_{steam}^2 \cdot \bar{d}_{film}^2 \cdot (1+\gamma)}{32 \cdot D_{ball} \cdot \eta_{liquid} \cdot \gamma^2} \quad [6.13]$$

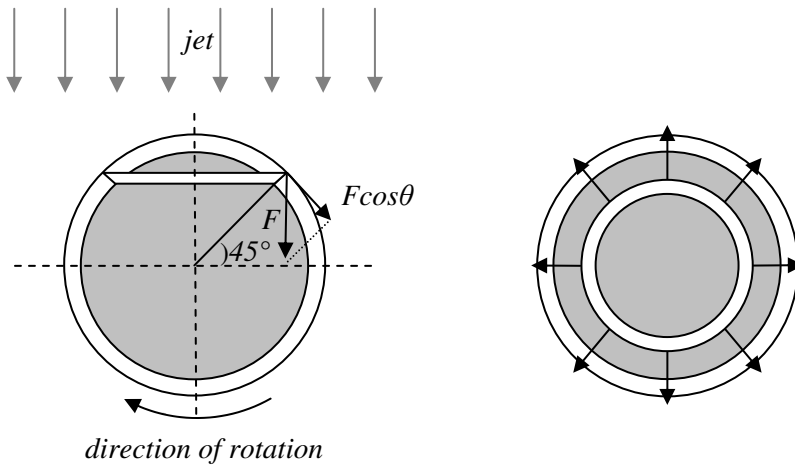


Figure 6.3: Schematic side view (left) and top view (right) of a wet ball that passes the steam jet. The ring-shaped film segment in the drawing is assumed to have an average thickness and exhibit average dewatering rates.

6.1.4 Combined heating, wetting, and dewatering

The unit of Fig.6.1 is reconsidered now, comprising an upper rotary sluice, feed entrance, dewatering screen, outlet, and lower rotary sluice. As the balls leave the upper sluice and tumble down towards the screen, steam starts to condense onto the metal surface, forming a liquid layer with growing thickness. Part of the condensate may trickle down. In the main, however, dewatering is ineffective, since the moisture simply redistributes between neighbouring balls and the wall of the feed pipe.

A rubber sheet at the entrance of the dewatering screen decelerates the balls and distributes them over the width of the screen. As the balls start to roll, the centrifugal forces acting on the liquid film increase, removing part of the condensate. Somewhere down the line, the steam jet contributes to that.

After passing the end of the screen, the balls cover some distance to the lower rotary sluice. Since their temperature is still lower than the steam temperature at this point, heating continues until the sluice compartment closes. The liquid film present at the end of the dewatering screen plus the condensate formed down the outlet and in the lower sluice move along with the balls to the next process step.

Combined heating and dewatering effects were calculated using a numerical approach based on Equations 6.1, 6.7 and 6.13. Amongst the input data for the calculations were: $T_{steam}=120^{\circ}\text{C}$, $T_{balls,t=0}=50^{\circ}\text{C}$, $b_{screen}=0.6\text{m}$, $\alpha_{screen}=45^{\circ}$, $d_{nozzle}=0.03\text{m}$, $W_{balls}=30\text{kg/s}$, $f=0.7$, and $D_{ball}=12\text{mm}$. The graph of Fig.6.4 presents the resulting ball temperature and thickness of the water film as function of the distance travelled. The film thickness is shown for three positions of the steam jet: at 1.5m, 1.9m and 2.3m.

The first 1.2m in the graph represents the distance from the upper sluice to the dewatering screen. The balls accelerate, their temperature rises, and the layer of condensate grows. Just before the dewatering screen, which extends from 1.2m to 2.5m, the balls come to a momentary halt and start rolling. The thickness of the water layer decreases now. As a secondary effect, the heating process becomes more efficient.

The graph of Fig.6.4 indicates that the dewatering effect is particularly large as the balls pass the steam nozzle. After the short intense contact with the steam jet, however, the heating process continues and the layer of condensate grows back rapidly. It takes just a short moment before equilibrium re-establishes between the wetting rate and centrifugal dewatering rate. Hence, the position of the jet does not exert large influence on the amount of water present at the end of the screen: the film thickness is approximately $10\mu\text{m}$ for each of three simulated positions. This value matches results from experiments using a similar set-up and a feed of pre-wetted balls at atmospheric conditions. Film thicknesses up to $25\mu\text{m}$ were found as well though for large balls throughputs and excessive quantities of added water. These are explained by small values of f in Equation 6.7.

The last meter in the graph represents the distance from the end of the screen to the lower sluice. The ball temperature is close to the steam temperature now, but it still rises. The condensate is not carried off anymore. The sluice compartment fully closes in the next 0.25s, allowing the water film to grow to a thickness of $25\mu\text{m}$. This final film thickness implies that every kg of balls takes along 1.6ml of water. It corresponds with a heat leak of almost 0.4MJ per kg of evaporated cake moisture for the standard ratio between balls throughput and drying capacity. Part of the heat can be recovered in the MSFE-process though.

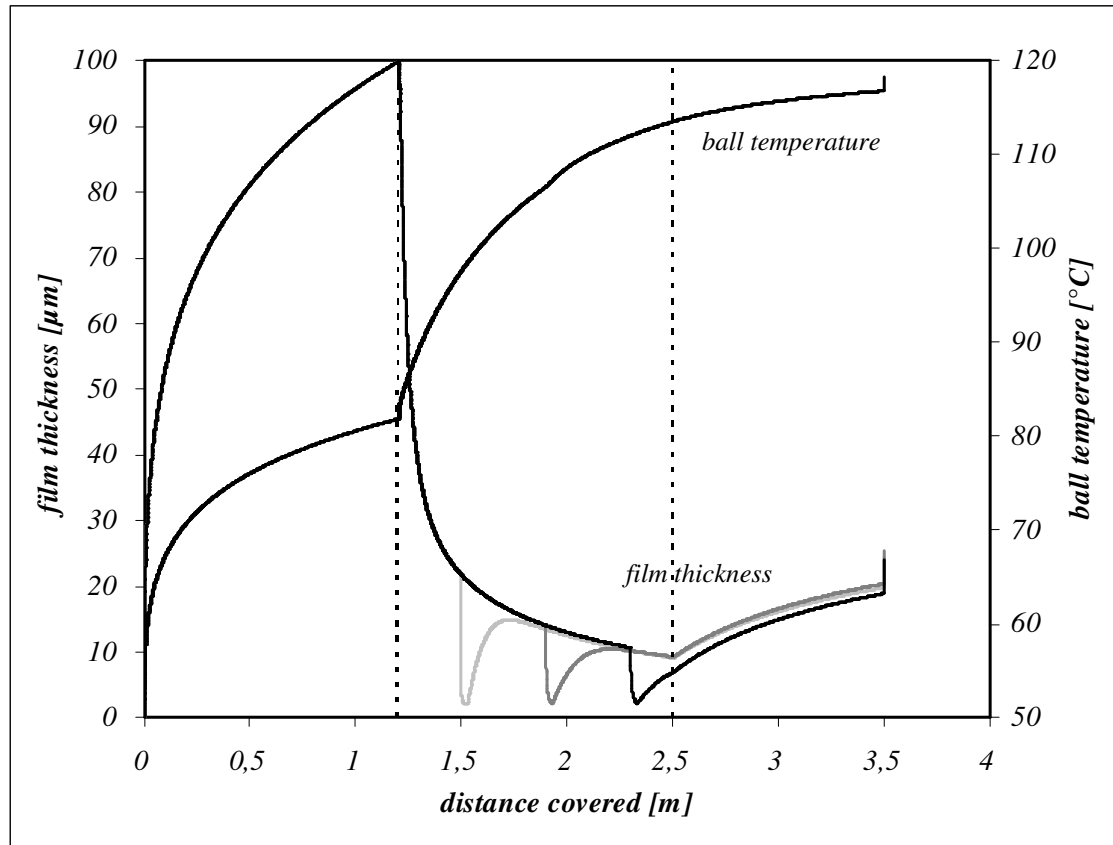


Figure 6.4: Simulation results showing the ball temperature and thickness of the liquid film as function of the distance covered. Three positions for the steam jet were investigated.

Considering the heat loss involved, it pays to reduce the amount of condensate that moves along with the balls. One approach is to improve dewatering efficiencies, e.g. by adapting the dimensions of the dewatering screen. A second tactic is to increase the residence time of the balls inside the steam vessel, so that their temperature more closely approximates the steam temperature at the time the balls reach the end of the dewatering screen. This leads to lower heat transfer rates and thus to the formation of less condensate in the period after that.

From this perspective, it can be concluded that the velocity of the balls in Fig.6.4 is too high. A smaller screen inclination angle (e.g. 30°) would yield better results on the condition that the balls run faster than is necessary for a monolayer. The rubber sheet at the entrance of the screen is another adaptation that increases residence times.

The graphs of Fig.6.5 display film thicknesses and ball temperatures for three situations: in case the balls do not meet any obstruction down the vessel; in case a weak rubber sheet just manages to stop the balls at the entrance of the screen; and in case a more rigid sheet brings the balls to a halt for the period of one second. The differences between the curves are significant. The obstructions allow the balls to reach higher temperatures and collect less condensate on their way to the lower rotary sluice. As a result, the final thickness of the water film drops from 40 μm to 25 μm and 15 μm. The specific heat losses associated with these values are 0.6 MJ/kg, 0.4 MJ/kg, and 0.25 MJ/kg respectively, based on the standard (100:1) mass ratio between balls and evaporated cake moisture.

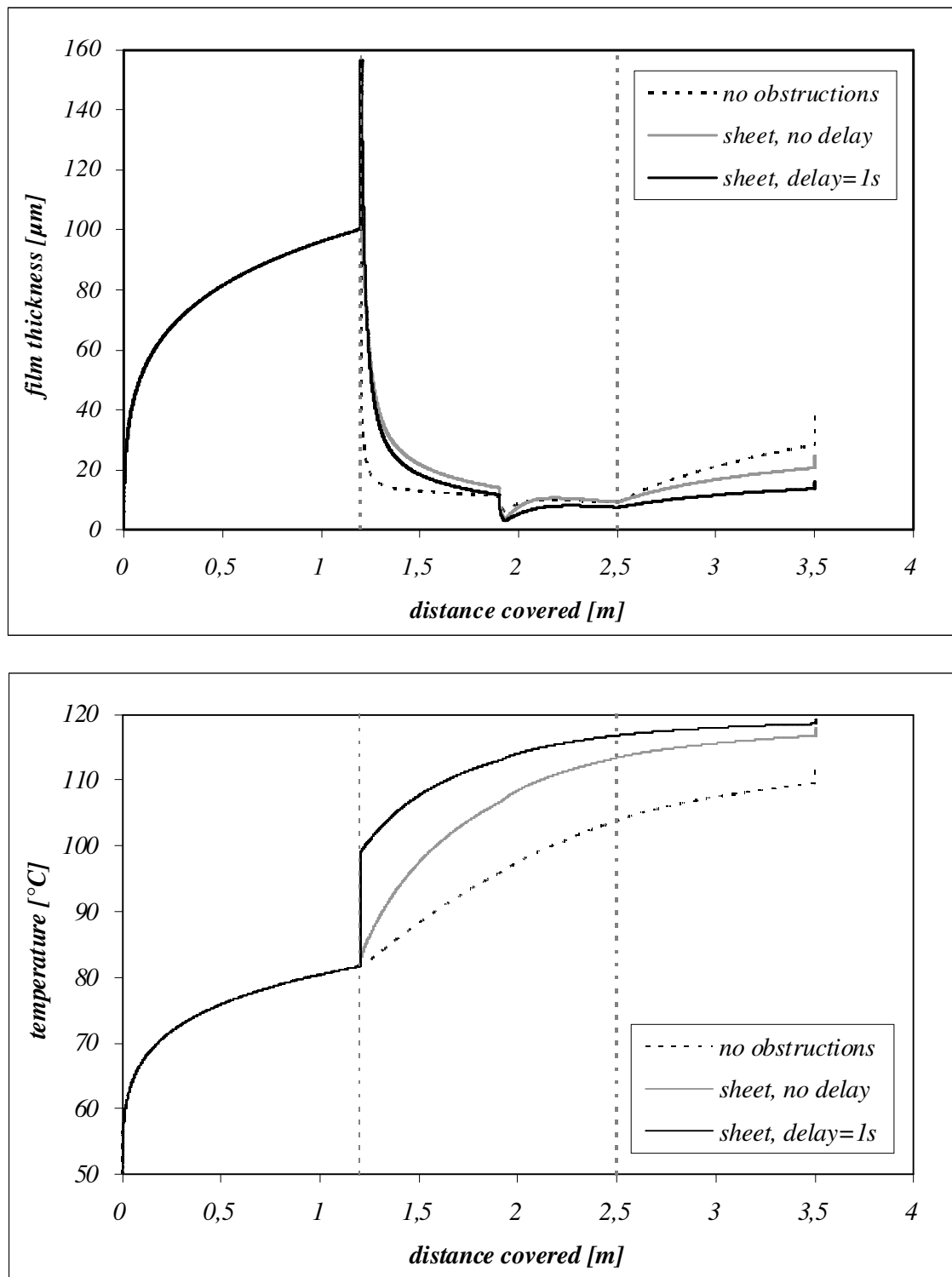


Figure 6.5: Ball temperatures and film thicknesses as function of the distance from the upper sluice. The graphs distinguish three cases: without any obstruction down the route; with a rubber sheet positioned at the entrance of the screen (bringing the balls just to a halt); and with a time delay of one second at the entrance of the screen.

6.2 Heat and mass transfer within the sections of a flash stage

6.2.1 Description of a flash stage

The local equilibrium model of Chapter 5 describes various fluxes of heat and mass between sections of the Delta dryer. Each section corresponds with one pitch of the screw conveyors, containing certain masses of steel, water, solids, vapour, and air. The model does not pay attention to fluxes that are present within a section. It assumes the content to be in thermodynamic equilibrium. This is of course a simplification, as equilibrium is not attained within the limited drying time. Various resistances to heat and vapour flows are present, and the process therefore automatically generates temperature and pressure gradients.

Fig.6.6 presents a transverse cross-section of the screw conveyors, showing two sections connected by a vapour bridge. The steel balls, the wet cake, and the contact layers between balls and cake contribute to the overall heat resistance at the left side where evaporation takes place. The ball beds and the vapour bridge form obstacles to vapour transports. The condensation process at the right side is hampered by a thin gas layer with high partial air pressure, and the heat resistances of the condensate and the balls.

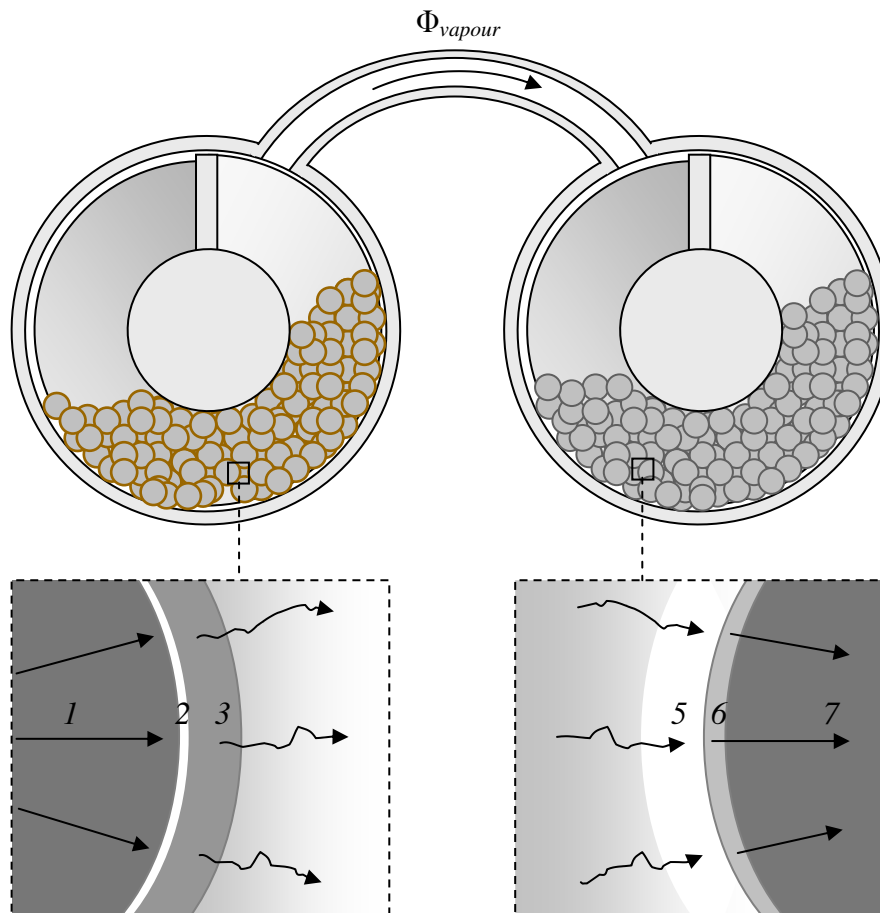


Figure 6.6: Cross-section of the screw conveyors with magnifications of ball-vapour interfaces. Heat is transferred from a hot ball (1) through a contact layer (2) and cake layer (3). Released vapour flows through the ball bed, passes a vapour bridge, and continues through the ball bed at the other side. It migrates through a laminar gas film (5) to condense onto a liquid film (6). The ball (7) eventually heats up by conduction.

The kinetics of the evaporation process was investigated in Paragraph 4.3. The heat flux from the ball through the contact layer and cake layer was described by Fourier's law, using an apparent heat transfer coefficient. Values for this coefficient, which were derived empirically, are relatively large initially. They decrease as the moisture content of the cake drops and the contact between the balls and the cake deteriorates. The apparent heat resistance does not include the resistances of the ball beds, the gaseous boundary layer, the condensate, and the balls at the condensation side. The contributions of the latter two can be added relatively easy though. The layer of condensate grows as the balls move through the screw conveyor. Its thickness depends on the amount of heat recovered in the flash stages and the amount of sprinkler water that is left. The moisture distribution depends on the presence of dirt at the surface of the balls, which influences the fraction that trickles down to the bottom of the bed.

6.2.2 Resistance of the ball bed

Released vapour from the cake must travel some distance through the ball bed, pass a vapour bridge, and proceed through the ball bed of the second conveyor. The pressure gradient that allows the transport results from the temperature differential between the bottom and the surface of the ball bed, and between facing sections of a flash stage. Because the content of the first screw conveyor cools during its transport and the counter-currently moving content of the second conveyor heats up, a natural vapour pressure drop exists along axial direction of the screw conveyors. The local equilibrium model describes leak streams between neighbouring sections based on this pressure drop. Within a flash stage, however, only the resistance of the vapour bridge is taken into account. The natural pressure gradient is assumed sufficiently large to carry all vapour through the bed. Whether this assumption is reasonable depends on the permeability of the bed and on the drying rate. Temperature is an important factor as well. Low temperatures yield low vapour densities and thus high volumetric flow rates. They necessitate relatively large pressure gradients, induced by temperature gradients that are even more significant. The subject of investigation here is the relationship between temperature and drying rate for which the natural leak streams are just large enough to allow complete vapour exchanges between facing sections of the flash stages.

The pressure drop over an infinitesimally short bed section is a function of the superficial gas flow rate (v_s) and the free volume fraction of the ball bed (ϵ_{void}), as formulated in Equation 6.14 [Ergun, 1952]. The upper limit for the required pressure drop is for a uniform mass flow rate of gas in the bed. This situation is plausible if the vapour that passes the vapour bridge (Φ_{vapour}) flows through the entire length of the bed, i.e. in case the number of sections is very large compared to the number of vapour bridges. The lower limit for the required pressure drop is for a single section per vapour bridge, in which the drying rate or condensation rate is uniform. The gas flow rate at the furthest end of the bed ($l=l_{bed}$) is negligible, whereas closer to the surface, it is the sum of vapour flow contributions from deeper layers of balls (Equation 6.15).

$$\frac{\partial P}{\partial l} = \frac{1.75(1 - \epsilon_{void}) \cdot \rho_{gas} \cdot v_s^2}{\epsilon_{void}^3 \cdot D_{ball}} + \frac{150(1 - \epsilon_{void})^2 \cdot \eta_{gas} \cdot v_s}{\epsilon_{void}^3 \cdot D_{ball}^2} \quad [6.14]$$

$$\frac{l_{bed} - l}{l_{bed}} \cdot \frac{\Phi_{vapour}}{A_{bed} \cdot \rho_{gas}} < v_s < \frac{\Phi_{vapour}}{A_{bed} \cdot \rho_{gas}} \quad [6.15]$$

The natural pressure drop in a section can be derived from Equation 6.16 and the relationship between temperature and free vapour pressure. m_{balls} is the ball mass per section and t_{rev} is the revolution time of the screw conveyors. The natural pressure drop is set equal to the required pressure drop in the bed following from Equations 6.14 and 6.15. The graph of Fig.6.7 shows combinations of temperatures and evaporation rates that satisfy this condition.

$$\Phi_{vapour} \cdot \left\{ H_{evap, 0^\circ C} + T_{bed} (Cp_v - Cp_l) \right\} = m_{balls} \cdot Cp_{steel} \cdot \frac{\Delta T_{ballbed}}{t_{rev}} \quad [6.16]$$

The upper curve and lower curve of Fig.6.7 are for the upper limit and lower limit of the required pressure drop. They mark optimum working conditions. In the region below the curves (region A), the drying capacity necessitates an extra pressure gradient in the bed or an increased number of vapour bridges. In the region above the curves (region B), the natural temperature gradient carries all released vapour through the bed, and, on top of that, causes leak streams to the condenser.

The number of sections in relation to the number of vapour bridges was relatively small for the simulated Delta dryer in the previous chapter. Hence, the lower curve of the graph fits best. The mass flow rate through the vapour bridges was typically 0.02kg/s, corresponding with an optimum working temperature of 55°C. Peak values around 0.08kg/s (found for the first vapour bridge) require temperatures near 70°C. The conditions in the simulated Delta dryer were thus not far from ideal.

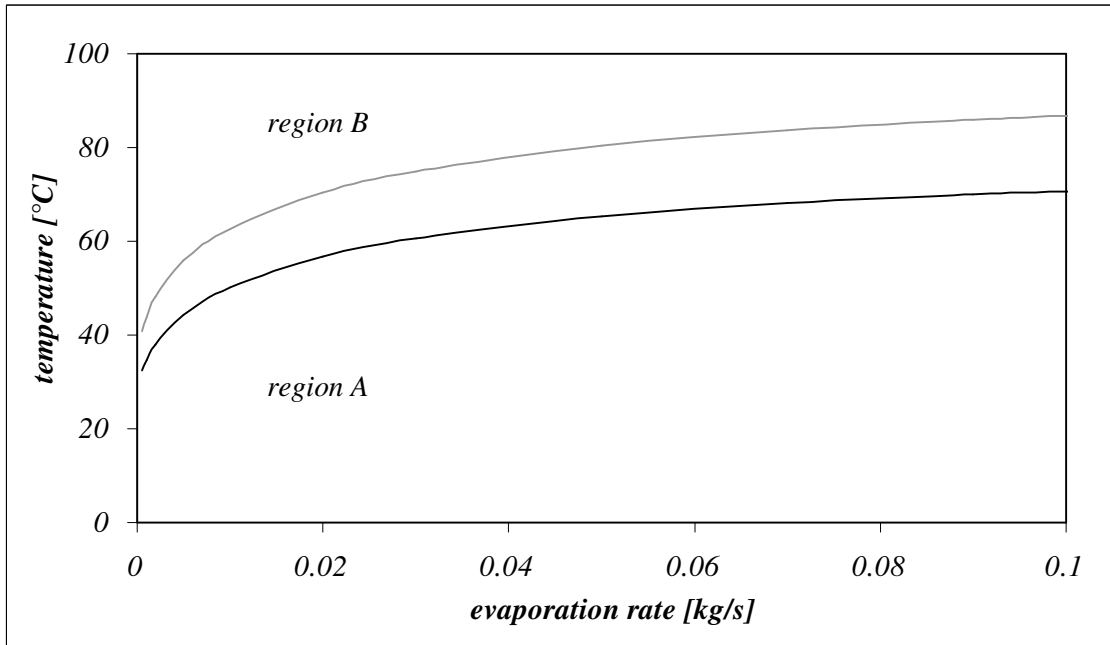


Figure 6.7: Temperature vs. evaporation rate, for which the natural pressure drop is just large enough to carry released vapour through the bed. The curves are based on calculations with $m_{balls}=66\text{kg}$, $D_{ball}=12\text{mm}$, $A_{bed}=0.023\text{m}^2$, $t_{rev}=2\text{s}$, and $\varepsilon_{void}=0.37$.

6.2.3 Resistance of a gaseous boundary layer surrounding a ball

While released vapour flows to the colder balls, small amounts of air are carried along as well. Unlike the vapour, the air does not condense onto the wet surface. It builds up, forming a thin boundary layer in which flow is laminar and only little intermixing occurs (Fig.6.8). The air levels off pressure gradients inside the boundary layer, reducing the local vapour flow velocity and thus the heat transfer rate.

The presence of such boundary layers is a well-known problem in heat exchangers. The negative impact on the performance of heat exchangers is usually described by empirical formulas. Unfortunately, literature does not mention anything comparable to the case examined in this paragraph. The heat and mass transfer rates through a laminar boundary layer to a solid particle of a packed bed are therefore derived analytically.

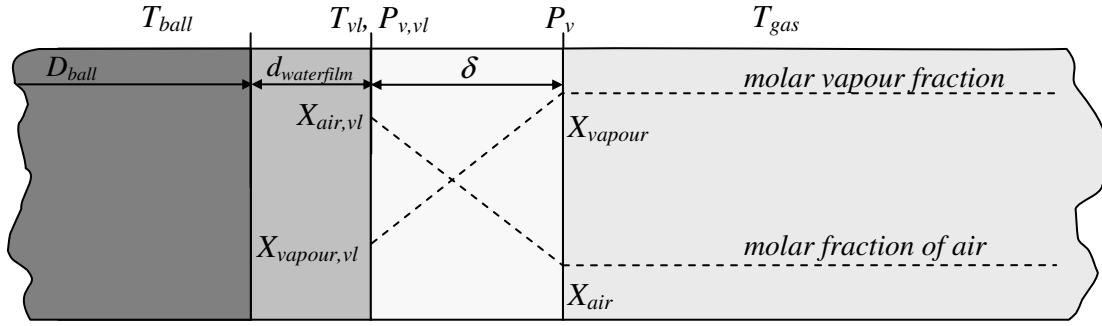


Figure 6.8: Magnified presentation of a ball fragment in an atmosphere of hot vapour and air. Condensate covers the surface of the ball. A gaseous laminar boundary layer with thickness δ surrounds the water film.

The thickness of the laminar boundary layer would continue to grow if there would not be a leak stream along axial direction of the screw conveyor. The momentum transfer by the gas stream reduces the layer thickness and keeps the condensation process going.

Colburn's empirical analogy between momentum transfer and heat transfer [Colburn, 1933] introduces a factor j_h as a function of the velocity of the gas stream (v_{gas}), the Colburn heat transfer coefficient of the layer ($h_{colburn}$), as well as the specific heat (Cp_{gas}), density (ρ_{gas}), thermal conductivity (λ_{gas}), and dynamic viscosity (η_{gas}) of the gas. The left side of Equation 6.17 defines this factor. The boundary layer thickness (δ) is one of the variables in the equation as part of the Colburn heat transfer coefficient (Equation 6.18). In analogous cases, the j_h factor for heat transfer is set equal a factor j_m for mass transfer. For fixed and fluidized beds, the latter is related to the Reynolds number and the void fraction in the bed, according to the right side of Equation 6.17 [Dwivedi, 1977].

$$j_h = \frac{h_{colburn}}{Cp_{gas}^{1/3} \cdot \rho_{gas} \cdot v_{gas}} \left(\frac{\eta_{gas}}{\lambda_{gas}} \right)^{2/3} = j_m = \frac{0.765}{\epsilon_{void} \cdot N_{Re}^{0.82}} + \frac{0.365}{\epsilon_{void} \cdot N_{Re}^{0.386}} \quad [6.17]$$

$$h_{colburn} = \frac{\bar{\lambda}_{boundary\ layer}}{\delta} \quad , \quad N_{Re} = \frac{D_{ball} \cdot \rho_{gas} \cdot v_{gas}}{\eta_{gas}} \quad [6.18, 6.19]$$

Heat transfer by conduction through the layer is negligible here in comparison with latent heat transfer by migration of vapour through the boundary layer. Since there are no net pressure gradients within the boundary layer, the average velocity of the vapour is diffusion-controlled. It complies with the Maxwell-Stefan equation. The driving force for diffusion is a concentration gradient $\delta x_i / \delta z$. Friction with dissimilar molecules forms an opposing force, which depends on the molar fraction of the foreign gas (X_j) and the mean velocity difference between the different types of molecules ($v_i - v_j$) (Equation 6.20). The diffusion coefficient D_{ij} follows from Equation 6.21 [Wesselingh, 1990].

$$\frac{1}{X_i} \cdot \frac{\partial X_i}{\partial z} = \sum_j X_j \cdot \frac{v_j - v_i}{D_{ij}} \quad , \quad \text{with: } D_{ij} = \frac{3.16 \cdot 10^{-8} \cdot \bar{T}^{1.75}}{\bar{P} (V_i^{1/3} + V_j^{1/3})^2} \cdot \sqrt{\frac{1}{M_i} + \frac{1}{M_j}} \quad [6.20, 6.21]$$

One component of the ‘binary’ gas mixture does not condense in this particular case. Its mean velocity in the boundary layer is thus zero. For simplicity reasons, the molar fractions of air and vapour are assumed to vary linearly across the boundary layer. This allows Equation 6.20 to be rephrased as Equation 6.22, in which molar fractions have been substituted by partial pressures: P is the total pressure; P_v is the vapour pressure at the periphery of the boundary layer; and $P_{v,vl}$ is the vapour pressure at the vapour-liquid interface, as indicated in Fig.6.8.

$$v_{vapour} = \frac{4D_{12}}{\delta} \cdot \frac{P_v - P_{v,vl}}{P_v + P_{v,vl}} \cdot \frac{P}{2P - P_{v,vl} - P_v} \quad [6.22]$$

Combination of Equations 6.17, 6.18, 6.19, 6.21 and 6.22 yields an implicit relationship between δ , $P_{v,vl}$, P_v/P , v_{vapour} , v_{gas} , D_{ball} , ϵ_{void} , a few constants, and some gas properties depending on P_v/P and T_{gas} . The thickness of the boundary layer and the pressure at the surface of the liquid film can be solved numerically. The temperature drop in the laminar boundary layer, which is of main interest here, is the difference between the gas temperature in the bulk and the saturation temperature belonging to the vapour pressure at the surface of the liquid film. (Conduction through the layer is negligible.)

Fig.6.9 shows the thickness of the laminar boundary layer and the temperature drop as function of the gas flow through the ball bed. This bed has the same dimensions as the ball bed sections of the simulated screw conveyors discussed in Paragraph 5.2. The presented data were derived for a gas stream with a bulk temperature of 87°C and partial air pressure of 0.001. The condensation rate was 0.15kg/s.

Clearly visible in the graph is the sudden increase in ΔT for leak streams below a certain threshold, whereas the thickness of the boundary layer grows more gradually with decreasing bed leaks. The grey line indicates the theoretical point at which the heat loss associated with the temperature drop equals the heat loss by the vapour leaks through the ball bed. Process conditions that belong at the left side of this line would benefit from increased bed leaks. This area of the graph, however, is irrelevant for the

Delta dryer since bed leaks are normally larger and condensation rates are much smaller than in the presented example. The temperature drop across the laminar layer will not even reach 0.3°C when using ‘worst-case’ simulation results from Chapter 5 as input for new calculations.

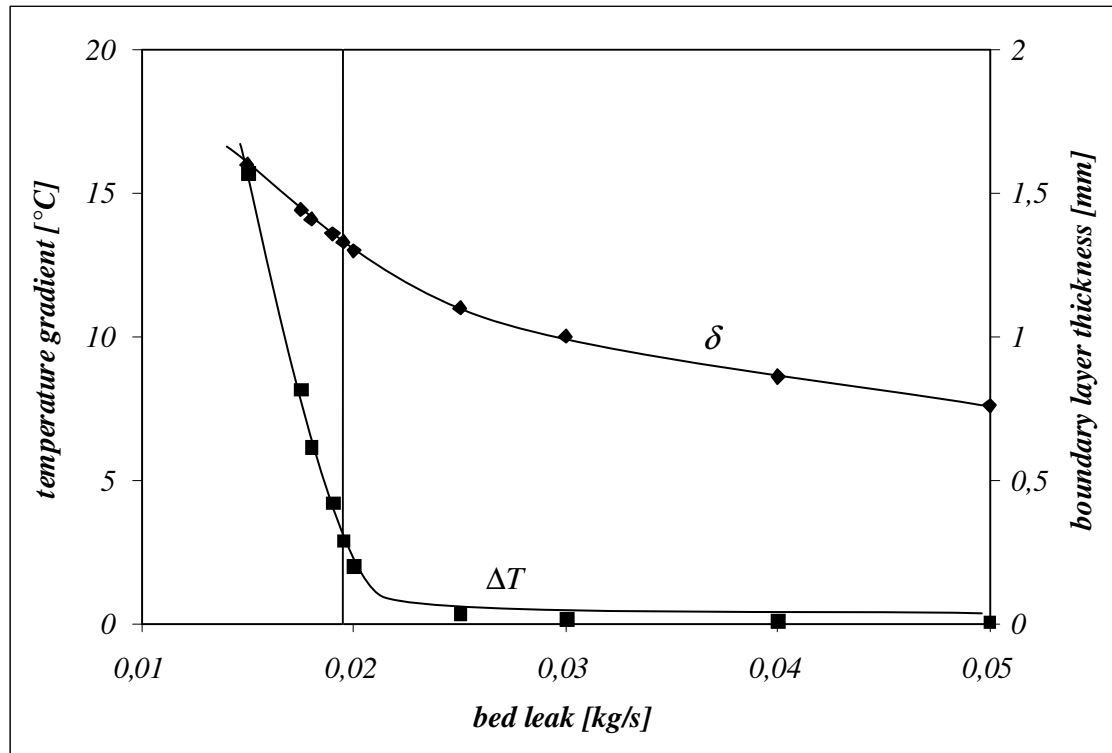


Figure 6.9: Temperature gradient (ΔT) and boundary layer thickness (δ) as function of the gas flow through the ball bed. The data are based on a condensation rate of 0.15kg/s and a ball bed with $m_{\text{balls}}=66\text{kg}$, $D_{\text{ball}}=12\text{mm}$, $\varepsilon_{\text{void}}=0.37$, and $A_{\text{bed}}=0.045\text{m}^2$.

6.2.4 Combined heat and vapour resistances within the sections

For economic reasons, the Delta dryer should combine high drying rates with low energy consumptions. It is impossible, however, to reach ultimate values for both variables at the same time. Higher drying capacities automatically lead to larger temperature gradients within the sections, whereas lower drying capacities increase heat losses per unit of dried material. As an added complication, temperature is a factor as well. As explained in Section 6.2.2, low drying temperatures may require additional pressure gradients within the ball bed, whilst high temperatures can result in significant vapour losses towards the condenser. Optimisation of process conditions therefore requires knowledge of the relations between drying capacity, temperature, balls-sludge ratio, bed leaks, and temperature gradients.

Section 6.2.2 showed the relationship between minimum process temperature and drying rate, for which the resistance of the ball bed does not hamper the process. For the standard configuration and operational parameters of the Delta dryer, the minimum temperatures will not be reached. Hence, no additional pressure gradients

will be present in the ball bed. The simulation results presented in the previous chapter were thus not too optimistic from this perspective. (In fact, the local equilibrium model is even pessimistic: the vapour flows involved in the flash stages were neglected when calculating the leak streams to neighbouring sections. Since they actually need to pass the ball bed as well, the leak streams should be smaller than predicted.) The laminar boundary layer appeared negligibly thin under normal operating conditions, and is thus neither an issue. This leaves the heat resistances of the balls, contact layers, cake layers, and liquid films as the primary rate-determining factors for the drying process.

The heat resistances cause extra temperature gradients between facing sections of the flash stages. These gradients can be calculated using Equation 6.23. The variables m_{balls} , m_{solids} and m_{liquid} are the masses of balls, cake solids and cake moisture present at the evaporation sides of the flash stages. k_{app} is the apparent heat transfer coefficient in the particular sections, which was introduced in Paragraph 4.3.

$$\frac{6 \cdot m_{balls} \cdot \Delta T}{\rho_{steel} \cdot D_{ball}} \cdot \frac{1}{\frac{1}{k_{app}} + \frac{d_{film}}{\lambda_{water}} + \frac{D_{ball}}{10 \cdot \lambda_{steel}}} = -\frac{\partial m_{liquid}}{\partial t} \cdot \{H_{evap, 0^\circ C} + T_{vapour} (Cp_v - Cp_l)\} \quad [6.23]$$

$$= -\frac{\partial T}{\partial t} \cdot (m_{balls} \cdot Cp_{steel} + m_{liquid} \cdot Cp_l + m_{solids} \cdot Cp_s)$$

The graphs of Fig.6.10 show moisture contents of dredging spoil and sewage cake as a function of drying time. The curves were reconstructed using Equation 6.23 with different values for ΔT . All evaporated cake moisture was assumed to condense onto the return balls, forming a uniform liquid layer with thickness d_{film} and thermal conductivity λ_{water} . Values for k_{app} , which were borrowed from Paragraph 4.3, dropped linearly with the moisture content of the cake.

The time range in the graphs is comparable with the residence time of the cake inside the Delta dryer. A temperature differential of 2.5-5°C is thus sufficient, allowing the cake to dry from 50% to 90% dry solids. As this extra temperature differential between the balls of the first and second conveyor will remain after the last condensation stage, the final temperature of the return balls will be 2.5-5°C lower than predicted by the local equilibrium model. Hence, energy consumptions will increase by 0.15-0.3MJ per kg of evaporated water.

This contribution to the total energy consumption of the Delta dryer is not the single consequence of the presence of heat resistances. When comparing the graphs of Fig.6.10 with simulation results from Chapter 5, the limited kinetics clearly causes lower initial drying rates. This will have some bearing on the macroscopic heat and mass fluxes. To investigate the total effect on the drying efficiency, the kinetic aspects discussed in this chapter should be combined with the thermodynamics of the local equilibrium model and worked into new simulation software.

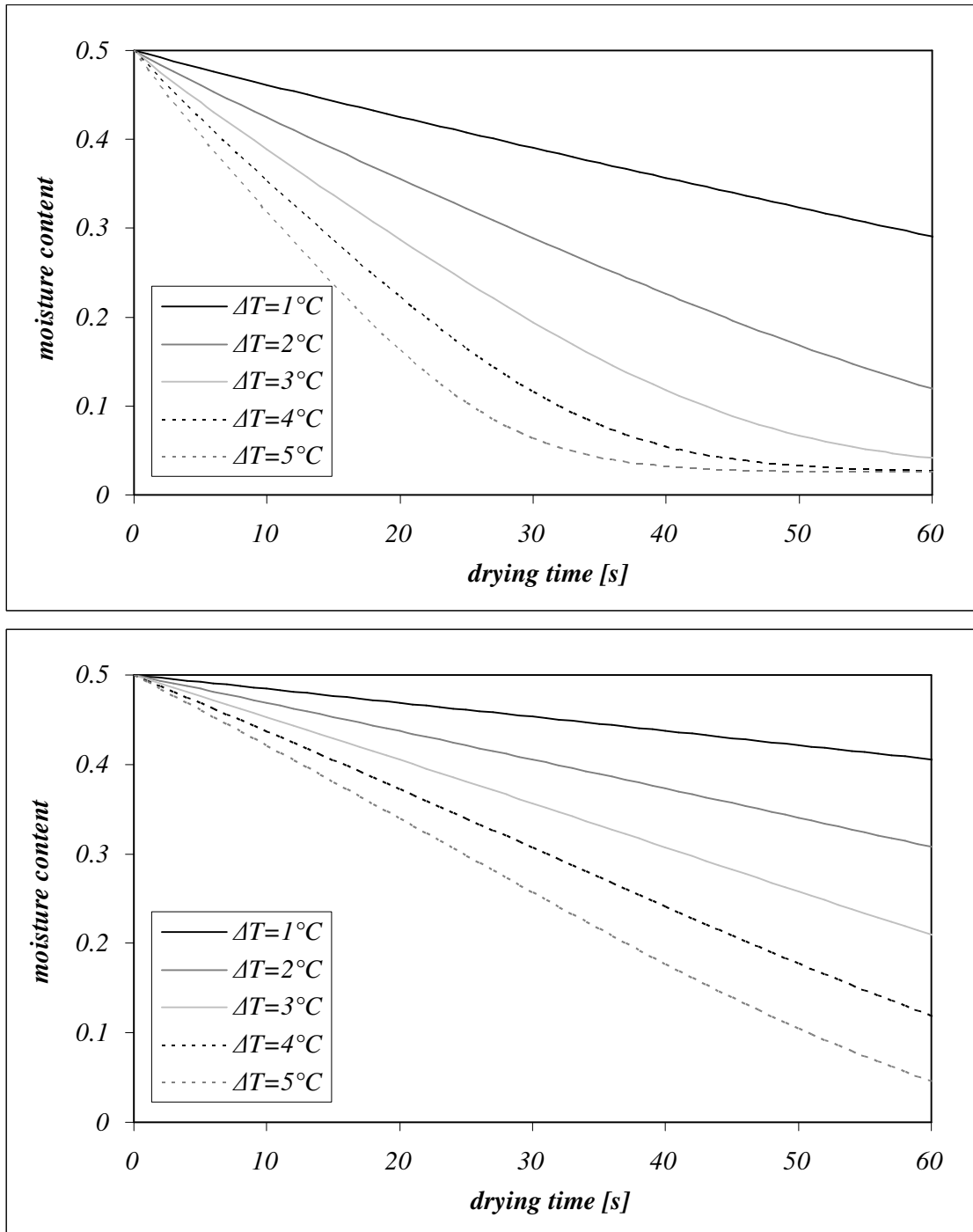


Figure 6.10: Moisture contents of dredging spoil (upper graph) and sewage cake (lower graph) versus drying time, for constant driving forces. The results are based on a standard 100:1 mass ratio between balls and evaporated cake moisture.

Chapter 7: The Delta dryer research prototype

A research prototype was developed and built with the aim to produce practical evidence on the drying principle. Beside this, it was meant to check the technical feasibility of the concept and explore various implementation details.

The principal design specifications for the installation were defined based on knowledge gained from simulation results. The chosen capacity of 1 ton of water per hour is relatively large for a laboratory set-up that is not intended to run in continuous production. This, however, appeared necessary, as the process efficiency depends on the drying capacity in relation to the heat losses. The capacity is roughly proportional to the internal volume of the installation, while the losses are controlled by the surface area. Hence, energy consumptions per kilogram of removed water increase for smaller versions.

The first paragraph of this chapter gives an impression of the prototype, introducing its main components. Four units of the apparatus are responsible for the primary operations. These are:

- The steam vessel
- The two sluices enclosing the steam vessel
- The elevator plus feeding operation
- The screw conveyors together with the condenser

The following paragraphs elaborate on the units separately. They describe the functional requirements, present selected hardware and necessary modifications, and discuss results from performance tests. The last part of this chapter shows how the units combine with secondary equipment to form an assembled installation, and how this prototype should be operated. It presents results from drying experiments and concludes with an evaluation of the test phase.

7.1 Introduction to the prototype's main components

The design capacity of 1 ton of water per hour makes the prototype a small apparatus compared to most drying plants, yet a full-scale installation of practical use. To reach this capacity, the Delta dryer moves around 120 tons/h of steel balls in a closed vacuum circuit. The balls have a diameter of 12mm, which is a compromise between surface-volume ratio and handleability. Like most parts of the installation, they are made from steel. The use of stainless steel is limited to a few components, as the equipment should not contaminate the product by abrasion. Besides, construction steel has superior thermal properties and does not corrode during normal operation as long as air is absent.

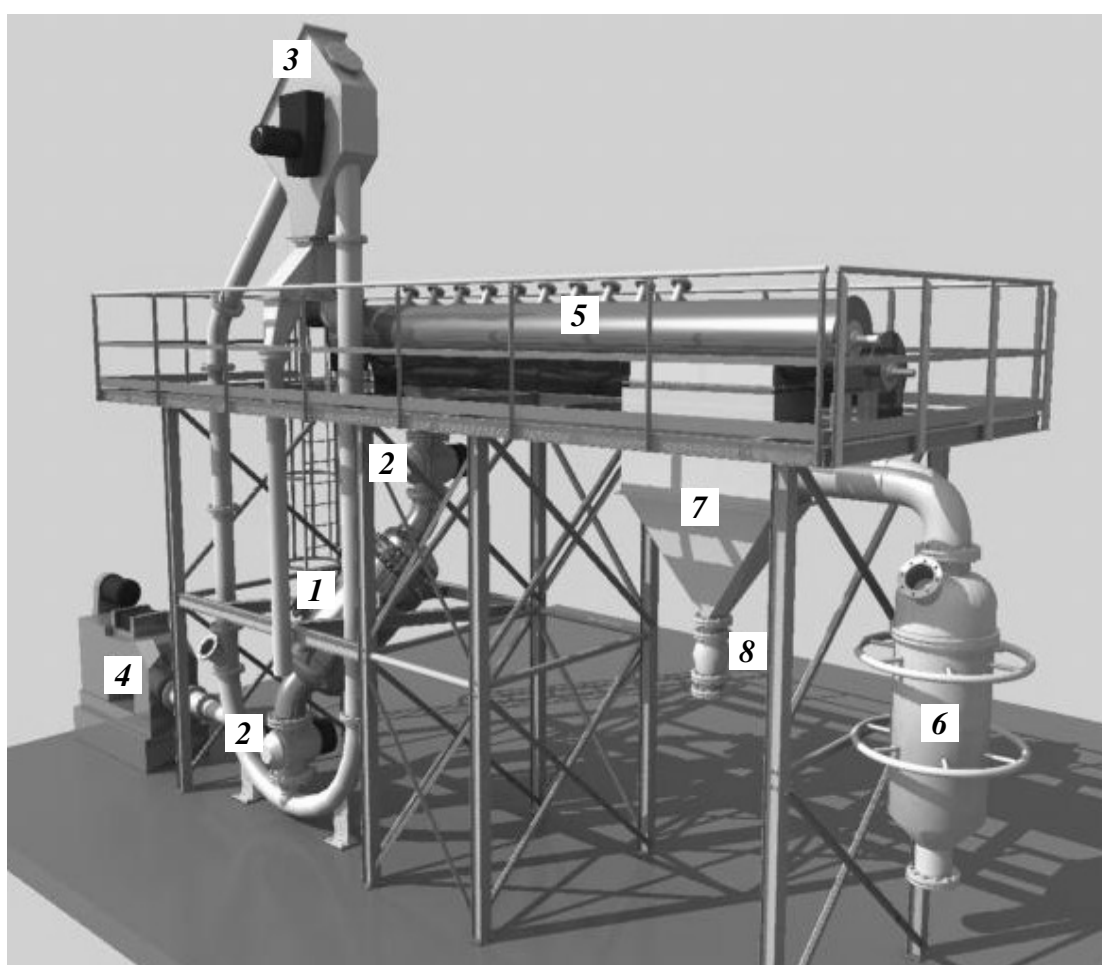


Figure 7.1: Rendering of the Delta dryer prototype (with permission of Van Tongeren Kennemer BV).

Fig.7.1 illustrates the prototype with its primary components. Heating and dewatering of the balls takes place inside the steam vessel (1), which is enclosed on both sides by sluices (2). These two sluices preserve the pressure difference between the vessel and adjacent units. After passing the lower sluice and entering the bottom of the elevator (3), the balls meet the wet cake that is pressed into this part of the installation either

by a clay extruder (4) or by a hose pump. The elevator mixes the balls and the cake and brings the mixture up to the first of two screw conveyors (5) where it dries. Released vapour is carried away via vapour bridges partly to the counter current balls stream in the second conveyor, and partly to a spray condenser (6). A drum screen housed at the end of the first screw removes the dried material from the balls. A funnel (7) collects the product. A sluice (8) transfers the product to big bags while keeping air from entering the vacuum zone.

7.2 Steam vessel

7.2.1 Heating of balls

The functions and working principle of the steam vessel were discussed in the previous chapter. One of its tasks is to supply heat from an external source and compensate the losses from the system. For the experiments, the heat was produced in a steam unit through combustion of oil. The conversion step is not taken into account, since it is not necessarily representative. Any available steam source with the right size and temperature can be used in practice.

The appropriate steam temperature is 105-120°C. Working pressures range from 1.2bar to 2bar, which, in combination with the internal volume, obliges the vessel to comply with the Pressure Equipment Directive (PED). Other parts of the installation operate at lower pressures and carry overpressure safety valves. The PED regulations are therefore irrelevant for these units.

The internals of the steam vessel are comparable to those presented and assessed in Paragraph 6.1. The steam expands from a wide contracted nozzle that disperses the steam over the full width of the balls stream. The distance from the mouth of the nozzle to the dewatering screen is only 25mm, allowing the jet to contribute to the dewatering effect of the balls.

A rubber sheet is placed at the entrance of the dewatering screen. The sheet delays the balls and increases their average residence time to a total of 2-3 seconds. This period should be long enough to bring the temperature of the balls close to the steam temperature. The conditions inside the vessel, while in operation, indicated that the process meets the expectations. Small pressure oscillations inside the vessel were observed as an instant response to variations in the balls throughput.

7.2.2 Removal and discharge of condensate

The rubber sheet inside the steam vessel has multiple functions. Besides increasing the residence time, it helps distributing the balls over the width of the screen, levels off fluctuations in balls supplies, and ensures that the balls do not bounce their way down.

The screen itself consists of 5mm thick rectangular stainless strips, directed parallel to the balls stream and held together at equal distances of 8mm. The screen has a length of 1.2m, width of 0.7m, and declines at an angle of 45 degrees. These dimensions should suffice to remove the largest fraction of remaining sprinkler water plus the condensate formed in the second screw conveyor and in the steam vessel. Tests at atmospheric conditions confirmed the dewatering action, showing fully wetted balls to loose most liquid. The final thickness of the adhered water film was around 10µm.

Excessive quantities of added water, however, resulted in film thicknesses near 25µm. The capacity of the screen is thus finite. Its efficiency decreases when it is saturated. Dewatering experiments under steam injection were found a waste of time, since the thickness of the adhered water film could not be measured without disturbing the process. Results of such experiments should differ somewhat from those conducted at atmospheric conditions, because of the condensate that forms as the balls cross the distance from the end of the screen to the sluice. On the other hand, the dewatering effect should be larger due to the presence of the steam jet and due to the lower viscosity of water at elevated temperatures.

The liquid from the balls is not clean drinking water. It is contaminated with residual solids left after the drum screen and sprinkler installation, as well as dust particles that passed the vapour bridges. A transversal slit after the end of the screen ensures that most of it eventually reaches the bottom of the steam vessel and does not seep into the lower sluice after all. Discharge of the condensate occurs via an opening in the bottom of the vessel. A standard float control valve lets the liquid pass while forming a barrier to vapour flows. A small settling tank prevents it from collecting too much dirt, as this piece of equipment is naturally sensitive to clogging. Proper functioning of the float control valve is essential. Steam could otherwise escape, or liquid could accumulate in the vessel and eventually overflow into the lower sluice. Despite the fact that the settling tank is emptied periodically by flushing and that the valve appears to work well during short experiments, a more reliable steam trap is needed in order to guarantee flawless performance for a continuously operated dryer.

After leaving the float control valve, the condensate quenches in a water container, which cuts down the temperature below 100°C. If recycling of the process stream is economically and environmentally irrelevant, as is the case for the prototype, it is disposed directly or goes to a thickener for further treatment. Such treatment facilities are usually part of the infrastructure of a sludge-generating plant. The heat content of the condensate can also serve to enhance biological activity in a wastewater treatment plant.

7.3 Sluices

7.3.1 Transport of balls

The sluices maintain the pressure difference between the steam vessel and the rest of the installation. They obstruct vapour flows from the vessel to the elevator and the second screw conveyor, and, at the same time, pass on four to five thousand balls per second across the pressure gradient. The upper sluice handles the balls coming from the end of the second screw conveyor. It arranges a continuous feed for the steam vessel. The lower sluice removes the balls from the end of the screen, which must be done quickly in order to minimise the amount of condensate formed after dewatering. Its output should not vary too much either.

The hardness and unpredictable behaviour of the balls (as part of a bulk stream) make the sluices charged with difficult tasks. The risk of moving parts getting jammed seems a fundamental problem if a single mechanism needs to repel balls and establish an airtight seal at the same spot. Physically separating these actions can avoid the obstacle, implying that the bulk stream must be drawn away first before a sluice compartment closes.

No standard sluice system directly meets the requirements. After some adaptations, however, both rotary sluices and systems relying on pneumatically driven slide valves could perform satisfactory. The working principle of the latter is superior. These types operate with stop valves that move back and forth in an alternating mode. They are usually leak proof and do not remain stuck or damage themselves in case the mechanism is blocked. Unfortunately, however, pneumatically operated sluices cause large fluctuations in the balls throughput unless they are redesigned from scratch. Rotary sluices on the other hand, are compact, relatively cheap, and more easily adaptable to meet the transport requirements. Their potential was explored first.

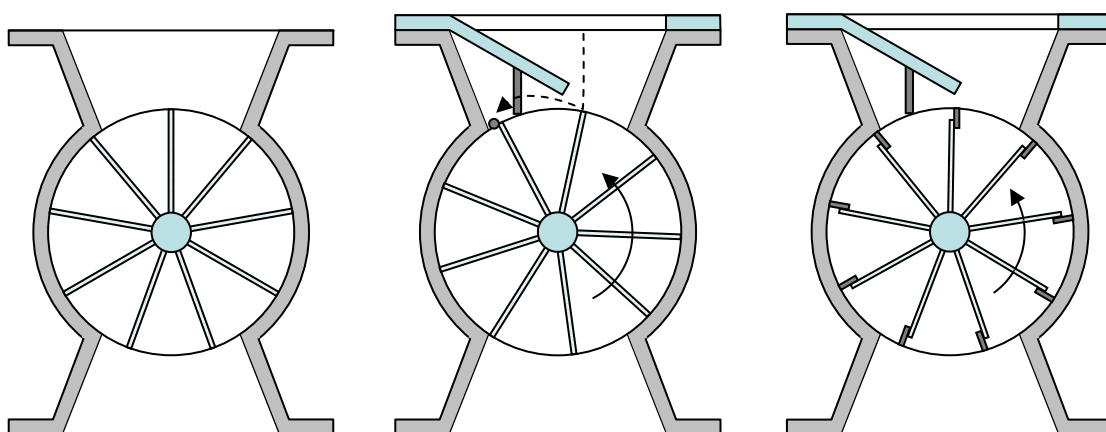


Figure 7.2: Schematic cross-sections of rotary sluices in the simplest configuration (left), furnished with an inlet plate and rubber sheet (centre), and as used in the prototype (right).

Fig.7.2 shows schematic cross-sections of rotary sluices. The left picture displays the simplest configuration. The sluice in the centre contains an inlet adapter, which keeps the majority of the balls away from the spot where the blades encounter the casing. The space between the blade and the adapter is obviously larger than a ball diameter. A rubber sheet hinders balls that jump up and attempt to follow the dashed line, so that the ones with poor timing do not trap themselves after all.

The sketched adaptations solve most problems if the fill level in the compartments is not too high and if the rotational speed is low enough to bring the balls at rest before the compartment closes. Tilting the sluice in clockwise direction shifts the point of closure upward in respect of the inlet, and further improves operation. Still, guiding every single ball smoothly through the sluice seems impossible. Further strategies are therefore to minimise the damage in case of stagnation. A simple electronic circuit, for instance, can make the rotor run shortly in the opposite direction when a peak load is detected, so that a jammed ball is liberated. Such an incidentally used resort will get the balls stream going again without impinging too much on the drying process.

The sluices of the prototype were not tilted or equipped with advanced electronic controls. Instead, some space was kept between the casing and the tips of the rotor blades. The gaps were bridged using rubber sheets mounted onto the blades (see the right drawing in Fig.7.2). These sheets made the sluices foolproof and resistant against overfeeding.

7.3.2 Steam leaks

Even the ideal sluice cannot prevent losses from the vessel to the rest of the installation, since it will not just pass on balls but also steam and condensate. These steam displacements are relatively small. They can be calculated after Equations 7.1 and 7.2. The sluices of the prototype have an internal volume (V_{free}) of 0.07m³ and revolve at a frequency (f) of 0.25Hz. The balls throughput is approximately 30kg/s. The pressure in the steam vessel (P_{in}) ranges from 1.3bar to 2bar, while the intended operating pressure at the other side (P_{out}) is around 1bar. These parameters result in steam leaks of 5-11g/s for the upper sluice and 0-4g/s for the lower sluice.

The second contribution to the losses, i.e. the amount of condensate that travels along with the balls, was evaluated in Section 6.1.4. It is in the order of 50g/s. It is not fair, however, to ascribe it fully to the sluices. The lower sluice is to some extent responsible for the liquid that leaves the steam vessel, given that it does not instantly take away the balls at the end of the dewatering screen. The large rotational velocity of the sluice ensures that the delay is only 0.25s on average. The amount of condensate formed in this period is acceptably small.

$$\text{upper sluice: } \Phi_{steam} = V_{sluice} \cdot f \cdot \frac{P_{in} \cdot M_{H_2O}}{R \cdot T_{in}} - \left(V_{sluice} \cdot f - \frac{m_{balls/s}}{\rho_{steel}} \right) \frac{P_{out} \cdot M_{H_2O}}{R \cdot T_{out}} \quad [7.1]$$

$$\text{lower sluice: } \Phi_{steam} = \left(V_{sluice} \cdot f - \frac{m_{balls/s}}{\rho_{steel}} \right) \frac{P_{in} \cdot M_{H_2O}}{R \cdot T_{in}} - V_{sluice} \cdot f \cdot \frac{P_{out} \cdot M_{H_2O}}{R \cdot T_{out}} \quad [7.2]$$

Less-ideal sluices bring additional losses thanks to the pressure gradient and the fact that the sluice compartments are not perfectly sealed. This especially counts for rotary sluices. The rotor blades are subject to a bombardment of balls as they pass the inlet. The seals will therefore wear faster than those that never cross the balls stream.

Small steam leaks through the upper sluice are of no concern. The steam simply condenses onto the return balls, heating them up before the balls enter the steam vessel. Larger leaks, however, may frustrate vapour transports in the first vapour bridge. They can even invert the flow direction in the vapour bridge if the local pressure in the second conveyor becomes higher than the pressure in the first conveyor. The steam that escapes through the lower sluice will extra moisturize the cake and can be considered lost right away.

The sluices of the prototype contained 6mm thick rubber seals mounted on top of the rotor blades. The tolerances between the sides of the rotors and the casings were within a few tenths of millimetres. The total resistance to vapour flows was investigated using the prototype installation. Hereto, the complete installation was pressurized with air, after which the elevator was suddenly opened. Whereas the elevator and second screw conveyor promptly returned atmospheric, the pressure in the vessel decreased more gradually due to the presence of the sluices. Fig.7.3 presents the monitored pressure drop as a function of time.

The gas fluxes through the sluices are roughly proportional to the momentary pressure gradient over the sluices ($P = P_{vessel} - P_{out}$). The decreasing pressure in the vessel can thus be described mathematically as in Equation 7.3, with A being a geometrical constant. When comparing the empirical experimental results of Fig.7.3 with theoretical curves for different values of A , they appear to match those for $A=0.07$ (stationary blades) and $A=0.1$ (rotating blades).

$$\left(\frac{dn}{dt}\right)_t \cdot \frac{R \cdot T}{V} = \left(\frac{dP}{dt}\right)_t = -A \cdot P_t \rightarrow P_t = P_0 \cdot e^{-A \cdot t} \quad [7.3]$$

Pressure gradients at the height of 0.3-1bar thus caused air leaks of 0.015-0.05kg/s per stationary sluice and 0.02-0.07kg/s per rotating sluice. The differences in loss between the stationary and rotating sluices are in agreement with the gas displacements of the empty sluice compartments.

Despite the fact that the experiments were done with air, without a balls throughput, and at room temperature, the results give a good indication of the steam losses to reckon with during the drying process. The kinematic viscosity of steam at 110°C closely approximates the kinematic viscosity of air at room temperature, so the steam leaks will match the measured air leaks. (Differences in gas density are corrected.) For an installation that dries 1 ton of water per hour and moves around 30kg of balls per second, the imperfect seals and the volume displacement of the sluices thus increase the specific energy consumptions with an amount of 0.15-0.5MJ per kilogram of removed product moisture. This is unacceptable.

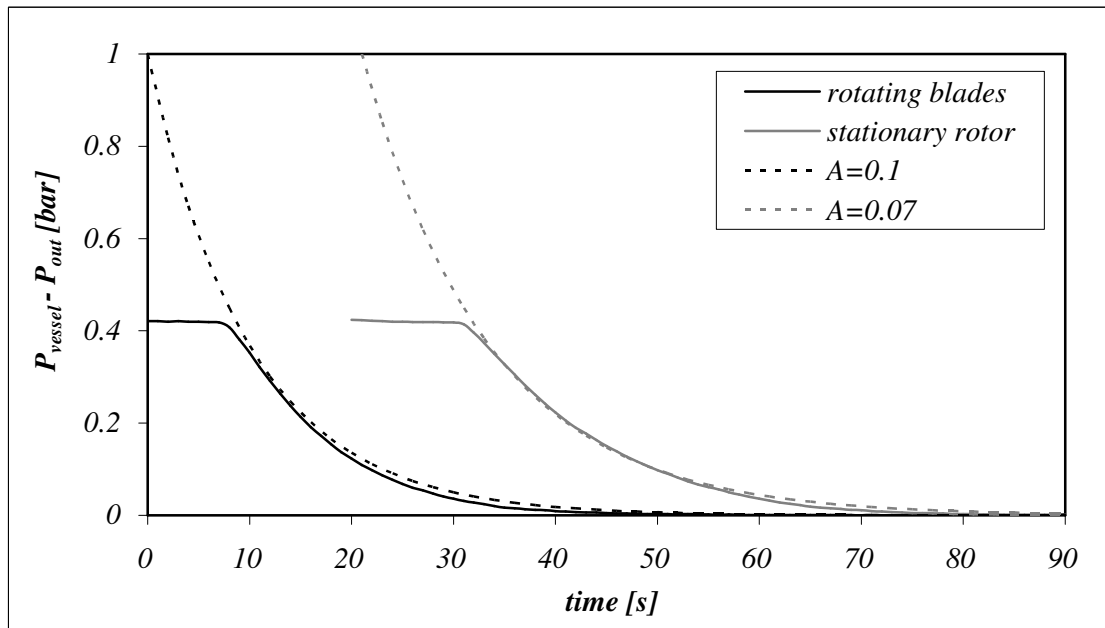


Figure 7.3: Results from leak tests. The pressure inside the steam vessel decreases with time as air escapes via the sluices to the atmosphere.

7.3.3 Flashing of sluice internals

Losses from the steam vessel also result from flashing sluice internals. Every revolution exposes the rotor blades to the steam atmosphere inside the vessel. The blades heat up but cannot get rid of the condensate. When they are subsequently exposed to the reduced pressure atmosphere in the elevator and second screw conveyor, the liquid re-evaporates, cooling the blades off again. This occurs repeatedly, involving transports of heat.

The used sluice rotors contained nine blades of 7kg each. At a rotational frequency of 0.25Hz, the steel mass subjected to the flashing mechanism was 32kg/s, i.e. comparable to the balls throughput. Despite the fact that the surface-volume ratio of a blade is 1.5 times less than for a sphere, that the contact time with the steam was shorter, and that the penetration depth of heat was thus smaller than for the balls, the losses would contribute significantly to the total energy consumption of the process.

The best way to solve this problem is to cover the sluice blades with a thermally insulating layer. The material of choice should display low values for the product of thermal conductivity, specific heat, and density. It should also be wear resistant and insensitive to the temperature and impact of the balls. High-temperature rubber is a suitable candidate that can form several millimetre thick lasting layers onto steel surfaces (Fig.7.4). Another option is to use a silicon polymer by the trademark of Speedy-con. This material does not withstand sharp objects but has the advantage of being mouldable by hand, making it easy and inexpensive to apply for small series. It performed well during many hours of thermal testing.



Figure 7.4: Thermally insulated sluice rotors.

7.4 Chain tube conveyor and feed operation

7.4.1 Input of wet cake and mixing with balls

Input of wet slurry occurs by means of a hose pump, which receives the substance from a continuously stirred bunker. The equipment is robust and easy to operate, but inappropriate if the feed is highly viscous. In that case, a clay extruder takes over the job. The extruder contains two screw conveyors. The first screw presses the cake from the entrance vertically through a perforated plate. The second forces it in horizontal direction towards an opening in the elevator. A vacuum chamber in between the screws de-aerates the cake. In this way, the apparatus can continuously supply wet feed, while acting as a seal between the air atmosphere and the water vapour atmosphere inside the installation.

The elevator accepts the feed, mixes it with hot balls coming from the steam vessel, and passes the mixture on to the first screw conveyor suspended 6m above the ground. The apparatus, known as chain tube conveyor, comprises a cylindrical duct in which a closed chain runs (Fig.7.1). Dishes that hold to the chain push the bulk load forward.

The feed enters the chain tube conveyor some distance ahead of the balls stream. Hence, it is heavily exposed to the impact and mass of the drying medium. The heat capacity of the cake cools the balls, yielding a mixture with a temperature close to 100°C. The corresponding pressure, with values a little above or below 1bar due to fluctuations in the feed rate, helps keeping air from penetrating this part of the installation. (The conditions during the drying experiments differed from the intended operating conditions, as discussed in Paragraph 7.7.)

The actual mixing operation takes place in the vertical pipe of the chain tube conveyor. An important feature making this happen, is the diameter of the dishes (0.19m) being significantly smaller than the diameter of the pipe (0.26m). The gap lets part of the content fall back alongside the wall, thus resulting in lower transport capacities compared to set-ups in which the dishes cover the full width of the pipe. The difference between the actual throughput and the theoretically maximum throughput based on chain speed and inner tube diameter is a measure for the turbulence created, i.e. the intensity of the mixing operation. It proved sufficient at least for dredged material with solids content of 50% to spread out over the balls.

7.4.2 Vertical transport

The gap between the dishes and the inner wall of the piping has also positive effects on the transport characteristics. It prevents balls from getting stuck near intersections, and it enables the elevator to level out fluctuations in the balls throughput.

The latter is illustrated in the graph below. While filling the empty elevator, the balls gather in the lower part of the vertical pipe. Only few of them are carried all the way up. The transport capacity remains low until a fill level is reached at which the column reaches the top of the pipe. An essential aspect is that the volume fraction of balls remains a function of the position in the tube: it is lower near the top and right underneath every dish. Further addition of balls fills up empty spaces and increases the throughput. In this way, the elevator acts as a buffer as long as the maximum fill level is not reached.

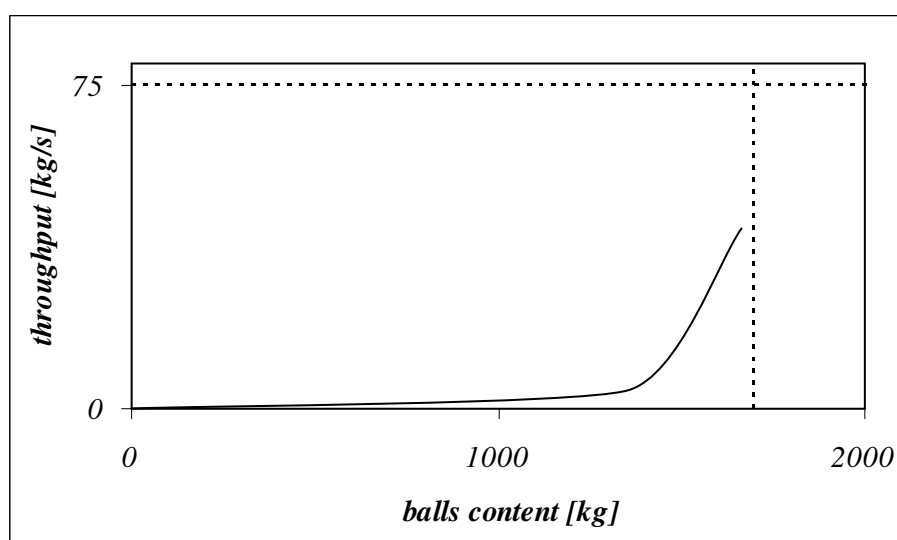


Figure 7.5: Throughput vs. fill level in the elevator (schematically). The dashed lines are theoretical maxima based on the pipe diameter and the chain speed.

The chain tube conveyor of the research prototype was a more or less standard unit with some reinforcements to sustain the heavy load. It had an opening through which it could accept new balls to be brought into the loop, as well as a lid to remove balls from the system. Since there was no desire to refill and empty the installation before and after every drying session, the drive motor was equipped with an electronic brake. Such a brake is necessary to prevent the chain from running backward by the mass of the vertical ball column when the motor shuts off. It keeps the bulk load away from the lower bend, avoiding excessive friction during a restart.

Although the chain tube conveyor performed well, a single mechanical hazard appeared hard to eliminate. The drive station, shown in Fig.7.6, forms the problem area. Most balls immediately leave the dishes when lifted above the end of the vertical pipe. However, some of them move further up. They fall, collide, and slide against the drive wheel before continuing their way to the outlet of the elevator. This gives them the opportunity to reach the exact spot where the chain engages the cogs of the drive wheel (noticeable by a knocking sound and a peak in the motor current). An entrapped specimen between chain and cogwheel pushes the chain aside as in the photograph. The following links miss the next cogs if they are too far off centre.

Although the described failure mode may seem improbable, the results of it were faced several times during a limited amount of test hours on the equipment. Guiding sheets positioned above the end of the vertical pipe and an adapted form of the dishes increased the time between failures with at least a factor of 10. However, they could not eliminate the problem.

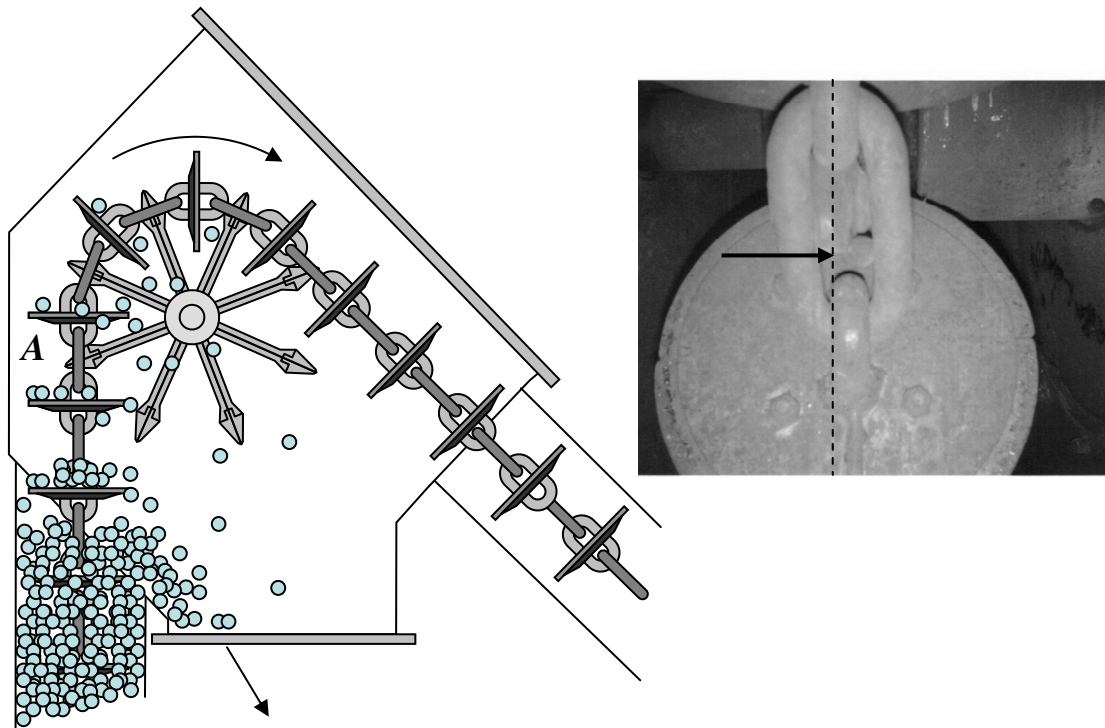


Figure 7.6: Cross-section of the drive station of the chain tube conveyor. The photograph was taken horizontally from the left side (A). It shows an unfortunate ball stuck between the chain and the cogwheel.

7.5 Screw conveyors and condenser

7.5.1 Handling of balls and cake

The geometry of the screw conveyors was selected based on simulation results, some of which discussed in Chapter 5. The conveyors had a length of 6m, hull diameter of 0.5m, and screw axis diameter of 0.27m. The pitch length was 0.2m, dividing each screw into 30 compartments.

Fig.7.7 shows the screw conveyors of the prototype. The balls-cake mixture supplied by the elevator (just hidden from view here) enters the right conveyor. The screw segments next to the entrance are reserved for the drying process. They cover the length of 20 pitches and give place to 10 vapour bridges spread out at equal distances. The end of the conveyor houses a 2m long drum screen that rotates concentrically with the screw axis. The part of the hull surrounding the drum screen displays a larger diameter than the rest of the hull, offering more room to the screen. It also allows smooth transitions for the balls.

Gravity makes the balls leave the right conveyor and enter the left, lower conveyor. The first part (the furthest end here) contains three sprinkler nozzles and a screen. These, however, are preferably redundant. They are to be left aside whenever possible because of the heat leaks that they generate (see Section 5.3.1). The last 20 pitches of the right conveyor connect with the vapour bridges.



Fig.7.7: The two screw conveyors of the prototype, partly wrapped in insulation material. The vapour bridges were equipped with pressure transmitters and thermocouples.

The screw conveyors of the prototype were made to measure on a lathe. The one-sided tolerance between the blades and the hull was 2mm. Aligning the axis to the hull was a delicate task. Despite its stiffness, the axis inclined to bow under its own weight, to be pushed upward and sideways by the balls. Hence, it rarely ran freely and thus wore faster at certain places.

The tolerances between the blades and the hull were significantly larger near the inlet and outlet of the screw conveyors. The extra space was meant to prevent balls from getting stuck. The balls stack visible in Fig.7.8 indicates that not any distance larger than one ball diameter satisfied. Although such stacks usually caused small dents at the most, a few of those together were found stable enough to bring the screws to a halt. An appropriate gap width between the blades and the hull appeared to be 25mm, i.e. a little larger than twice the ball diameter.

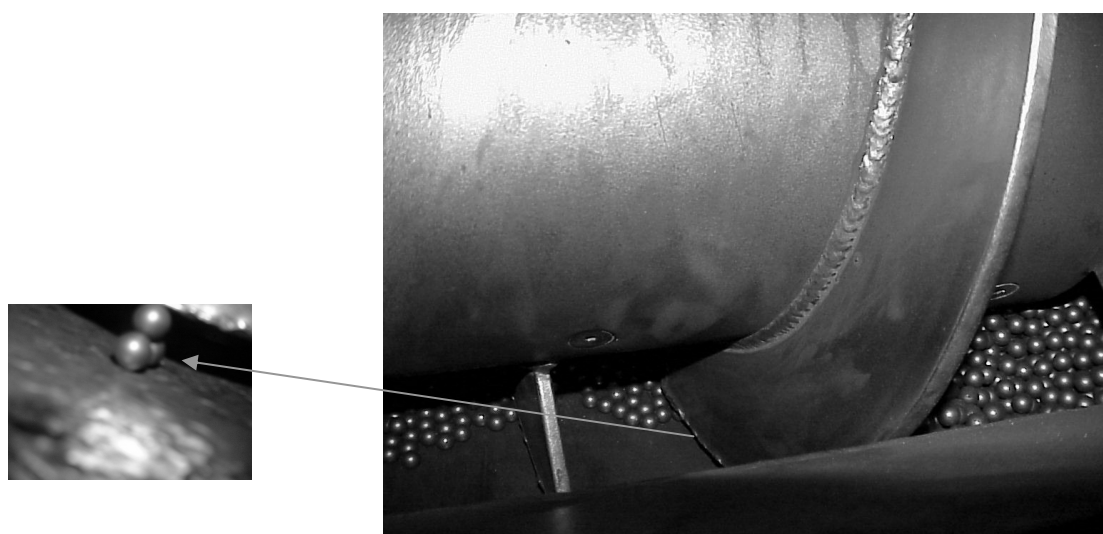


Figure 7.8: Photograph taken from the inlet of the first screw conveyor. The close-up shows a set of three balls forming a stable bridge between the screw blade and the hull.

7.5.2 Accommodating the drying process

Fig.7.1 also shows the spray condenser. A haze of cold water inside this piece of equipment generates low pressure, making the vapour condense at the site. By acting as a heat sink, the condenser is responsible for the temperature differences between the contents of the first and second conveyor, for the vapour flows through the bridges, and thus for the primary conditions for the MSFE-process.

The connection between the end of the first screw conveyor and the condenser was accomplished by means of a large diameter pipe. This pipe protruded from a double-walled funnel, positioned right underneath the drum screen. The funnel had a large volume, resulting in relatively low flow velocities. This, in combination with its double wall, helped removing the fines from the gas stream: the vapour moved together with the dried product down the funnel, entered the space between the inner and outer wall, flew upwards to the pipe, and separated from the solid particles that continued their way down.

The MSFE-process relies on the pressure differential across the length of the screw conveyors. For the latter to establish, the leak streams in longitudinal direction must be small in relation to the amount of vapour produced.

The leak streams through the slit between the screw blades and conveyor hull were reduced by the narrow tolerances applied. Pieces of metal sheet prevented short-circuiting when the blades passed the vapour bridge openings (see Section 5.1.4). Although simulation results indicated that a brush on top of the blades is beneficial, it was absent in the prototype for reasons of simplicity. Without a brush, the Delta dryer should still meet the minimum demands, yet display significantly lower drying efficiencies.

The other important leak streams, i.e. the vapour flows through the ball bed, depend on the length of the conveyor, the pitch length, the diameters of screw axis and hull, and on the fill level. The dimensions of the screws are obviously linked to the operational parameters. The intended drying capacity and steam temperature required a balls throughput of 33kg/s, whereas a drying time of 40-60s was attained using a rotational frequency of 0.5Hz. This made the ball bed occupy half the internal volume of the screw conveyors.

Unfortunately, a fill level of 50% appeared beyond reach in practice. The maximum ball mass per pitch was 42kg instead of the desired 66kg. Additional balls simply ran over the axis due to the momentum generated by the pushing side of the screws. (A pressure gradient anti-parallel to the transport direction contributes to this effect.) A smaller ball bed volume has direct consequences to the size of the leak streams. The impact depends also on the shape of the ball bed, as visualised in Fig.7.9: if the surface of the bed is flat or convex, a ball mass of 42kg will still reach the screw axis; if the surface is concave, the vapour leaks between the axis and the bed will be significant. Unfortunately, the latter was observed in the prototype.

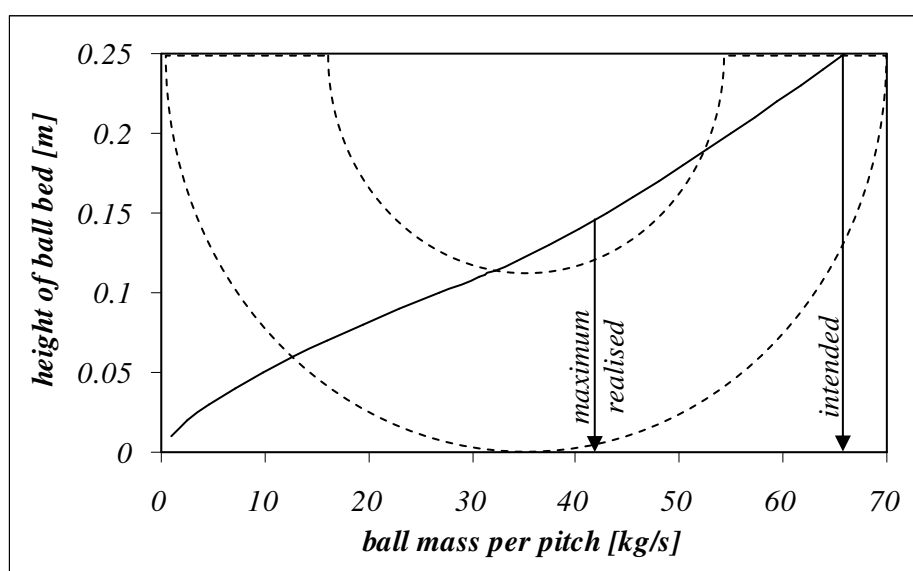


Figure 7.9: Height of a flat-surfaced ball bed as function of the fill level in the screw conveyors. The dashed lines indicate the diameters of the axis and the hull.

7.6 The assembled installation

7.6.1 Main components, appendages, and process streams

The main units discussed in the previous paragraphs were combined to form the Delta dryer research prototype. Fig.7.10 shows this installation. Beside the steam vessel, sluices, elevator, condenser, and screw conveyors, the prototype comprises transitions between components, some secondary equipment, and various ducts.



Figure 7.10: The Delta dryer prototype without insulation. The installation has a footprint of 10m x 3m, and height of 8m.

The flow diagram in Fig7.11 shows the various components and process streams. The dashed arrows indicate the liquid streams, the grey arrows the various gas flows, and the black arrows the transport directions of balls and cake. The presented temperatures and pressures refer to intended operating conditions.

Since the pressure in most parts is below atmospheric pressure, a few pumps are required at places where water is supposed to leave the installation. These pumps operate autonomously by means of float switches that monitor water levels in small reservoirs along the ducts. One-way valves prevent backflows of liquid. A liquid ring

7.6.2 Start up and shutdown procedures

When putting the Delta dryer into operation, the first in a sequence of prescribed actions is to set the balls in motion. This obliges synchronisation of the equipment involved, which is why the sluices, elevator, and screw conveyors are controlled by means of a single switch. It is not possible to activate any of these units separately. If one of the units comes to a halt, the whole installation shuts off automatically in order to prevent accumulation of balls.

The mechanical start up phase is followed by a de-aeration procedure. Instead of using the vacuum pump for this purpose, it is preferred to displace air by means of steam injection. This favours subsequent steps as no low pressure is generated yet. While the steam enters the steam vessel, the air flows through a temporary opening near the condenser, which is the furthest point in the installation. It runs through a duct that ends below the surface of a water reservoir (see Fig.7.11), so that the exhaust is clearly visible. The transition from a bubble bath to a calm water basin denotes the end of the de-aeration phase. The steam, which has replaced the air, implodes audibly below the surface of the water basin.

The installation obviously heats up during steam injection. The balls leave the vessel at elevated temperature and transfer heat as they circulate through the installation. Steam that escapes from the vessel or that forms after re-evaporation of liquid also raises the temperature of parts that are inaccessible to the balls. Significant amounts of condensate are thus produced until a uniform temperature of 100°C is reached.

The lower part of the chain tube conveyor is one of the places where condensate collects. The pressure inside the installation rises above atmospheric pressure as the exhaust air duct is closed. Hence, the condensate can be removed relatively easy. It leaves the installation via a drain at the bottom of the chain tube conveyor, via the discharge sluice, and via the sprinkler water outlet.

The water supply to the condenser is opened after completion of the dewatering phase. It causes an instant pressure drop. Drying conditions establish so that remaining condensate evaporates. The vacuum pump is switched on almost simultaneously to prevent new build up of air in the reduced pressure atmosphere. Finally, the extruder or hose pump is set in motion, initiating the actual drying treatment.

The purpose of the shutdown procedure is twofold: to clean and dry the balls and the internals of the installation; and to stop the balls stream in a controlled way.

The first step is to switch off the extruder or hose pump and to maximize the sprinkler water supply. As soon as the balls are dirt-free, the sprinkler water supply is closed, followed by the steam supply and water supply of condenser. The vacuum pump is switched off as well.

The balls, which keep on circulating through the installation, may be protected against oxidation by the addition of oil. They may also be removed from the system via a lid at the top of the chain tube conveyor. A single switch finally makes the main units shut down in sequence, starting with the screw conveyors, then the sluices, and ending with the elevator. Short time intervals between the units empty the ball volumes above the sluices and in the lower bend of the chain tube conveyor, which could otherwise impede a restart. The surplus of balls as result of this shutdown procedure is stored in the first pitch of the first screw conveyor. It levels off quickly when the screw is put in motion again.

7.7 Performance of the prototype as a whole

7.7.1 Test results

Various leak tests and thermal tests were performed on the research prototype. To check whether the installation was sufficiently airtight, low pressure was created using a vacuum pump. The pressure rise was subsequently registered as function of time. Fig.7.12 presents results of this test. It shows that the pressure rise at a uniform pressure of 0.3bar was approximately 0.3bar/h. This, based on an internal free volume of 5m³, corresponds with an air leak of 0.5g/s. Simulation results presented in Section 5.3.3 indicate that such air leaks are too small to have a noticeable effect on the drying process. In fact, the leaks will even be smaller in practice, since the pressure in some critical units will be higher than 0.3bar.

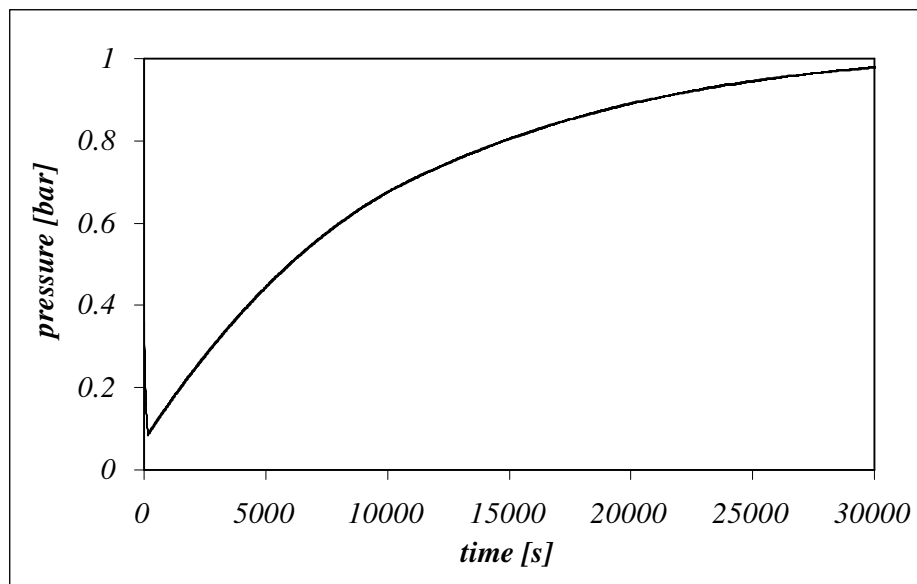


Figure 7.12: Results from vacuum leak test, showing the pressure rise as function of time.

The graphs of Fig.7.13 show typical conditions in different parts of the installation during the start up period of a drying experiment. The upper graph displays temperatures in the vapour bridges, in the steam vessel, and in the condenser. The lower graph indicates pressures at these locations, as well as the free vapour pressure corresponding with the temperature in the third vapour bridge ($P(T3)$). The cumulative steam consumption has been plot along the secondary axis.

The time line in the graphs distinguishes five phases. It starts as the balls are running through the installation. The steam supply opens at the first mark (1), making the air leave and the installation heat up. In the period from (1) to (2), around 200kg of steam is consumed. Most air has been removed near the second time mark (2), which is noticed by the small difference between total pressure and vapour pressure ($P(T3)$). The exhaust air duct is closed now and the installation is dewatered, after which the temperature starts to rise again. The pressure inside the steam vessel reaches its ultimate value of 1.7bar at the third time mark (3). The maximum pressure in other parts of the installation is 1.4bar, set by overpressure safety valves.

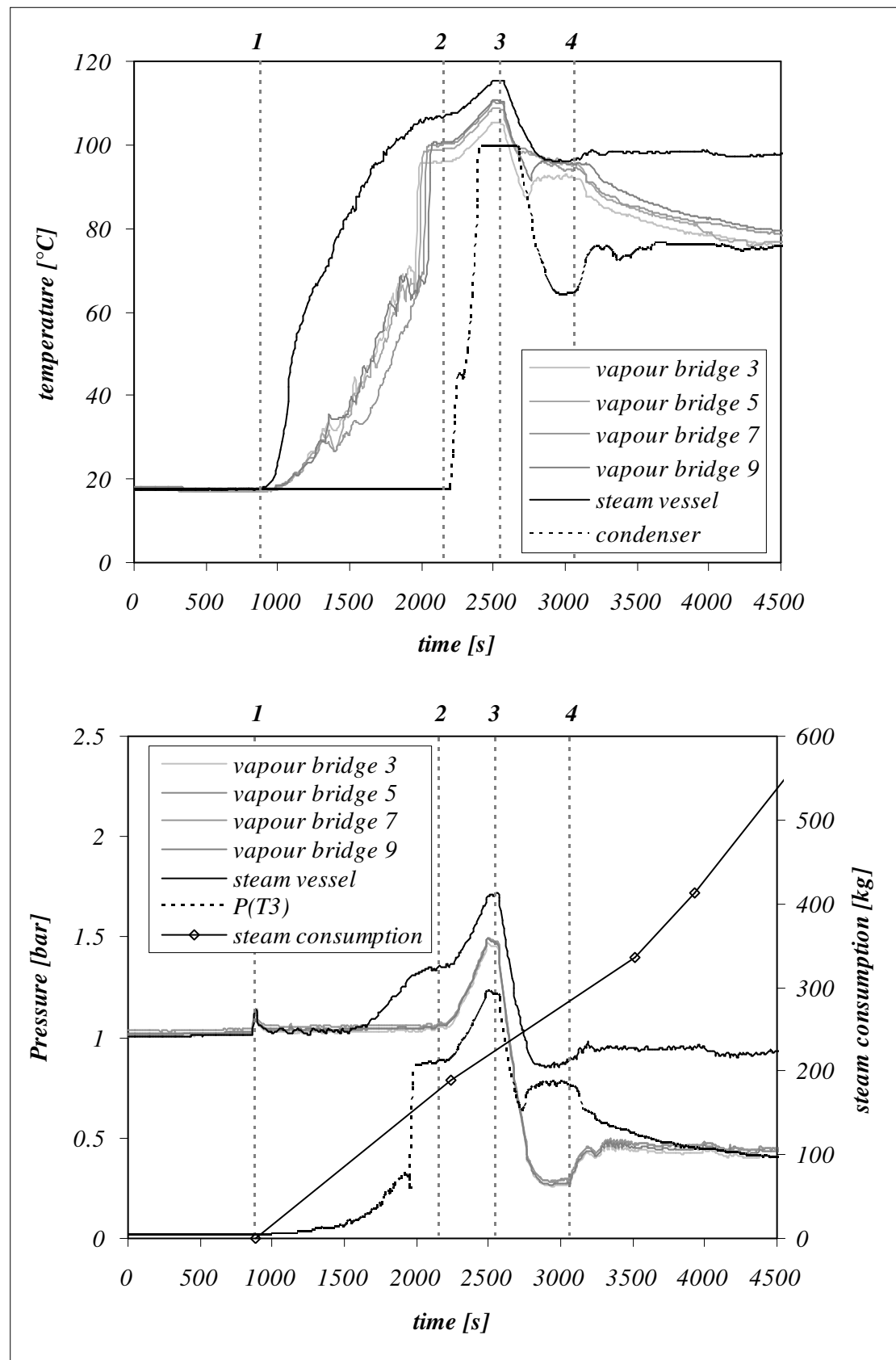


Figure 7.13: Temperatures and pressures during the start up phase of a drying experiment. $P(T3)$ is the free vapour pressure belonging to the temperature in the third vapour bridge. (Note: the sluice blades were only one-sidedly insulated prior to the experiment.)

The water supply to the condenser opens and the vacuum pump switches on right after the third time mark (3) in the graph. The pressure inside the installation drops rapidly, while the temperature decreases as remaining condensate evaporates and flows to the heat sink. The steady period after the temperature drop ($t=2700\text{s}$) is worth mentioning. It is explained by the absence of liquid water. The pressure inside the screws is significantly lower now than the free vapour pressure corresponding with the temperature of the balls ($P(T_3)$), so the driving forces for evaporation are large. The fact that the temperature of the balls approaches the temperature in the steam vessel indicates that the heating and dewatering process is efficient. Evaporation of a $10\mu\text{m}$ thick liquid film from steel balls with a diameter of 12mm would cause a temperature drop of 3°C . Therefore, if the conditions were truly stationary at this point, the thickness of the adhered water film would be small.

The fourth mark (4) denotes the start of the actual drying treatment, as from this moment, an amount of 0.1kg/s of water is sprayed onto the balls in the chain tube conveyor. It coincides with a temperature and pressure rise inside the condenser. The balls cool off as the liquid evaporates, which should eventually yield a certain temperature profile across the length of the screw conveyors.

The anticipated temperature drop of the balls is 22°C , which is in accordance with the temperature difference between the screw conveyors and the steam vessel, observed near the end of the time line. However, two problems showed up during this particular experiment, making the presented results less favourable than they seem. The first issue is the relatively high temperature in the condenser ($\sim 65^\circ\text{C}$ at $t=300\text{s}$) prior to the actual drying operation. Since the water supply to the condenser was $6.5\text{m}^3/\text{h}$, the temperature rise corresponds with a heat loss of 340kW , equivalent to a condensation rate of 0.14kg/s of steam. The second issue is the low pressure inside the steam vessel for $t > 2700\text{s}$. The pressure reaches values below atmospheric pressure, so that the condensate cannot pass the float control valve. The liquid collects in the steam vessel until it runs over into the lower sluice. As a result, the balls cool off further and further to the temperature of the condenser ($t > 3500\text{s}$). The driving forces for evaporation diminish, and the drying process comes to a halt.

The evaluated experiment was short-lived because of the high temperature in the condenser and the reduced pressure in the steam vessel. Additional drying experiments were carried out on an improved installation using a mixture of silver sand and water as feed. Fig.7.14 presents results from temperature and pressure measurements during the start up phase of one of these experiments.

The trends observed in Fig.7.14 are similar to those in Fig.7.13. The different phases are comparable. However, an extra phase is distinguished here, during which the steam supply to the steam vessel increases. It extends from the fifth mark (5) to the end of the graph. The pressure in the steam vessel is above atmospheric pressure this time, so that the condensate flows through the float control valve and dried material is received through the product sluice.

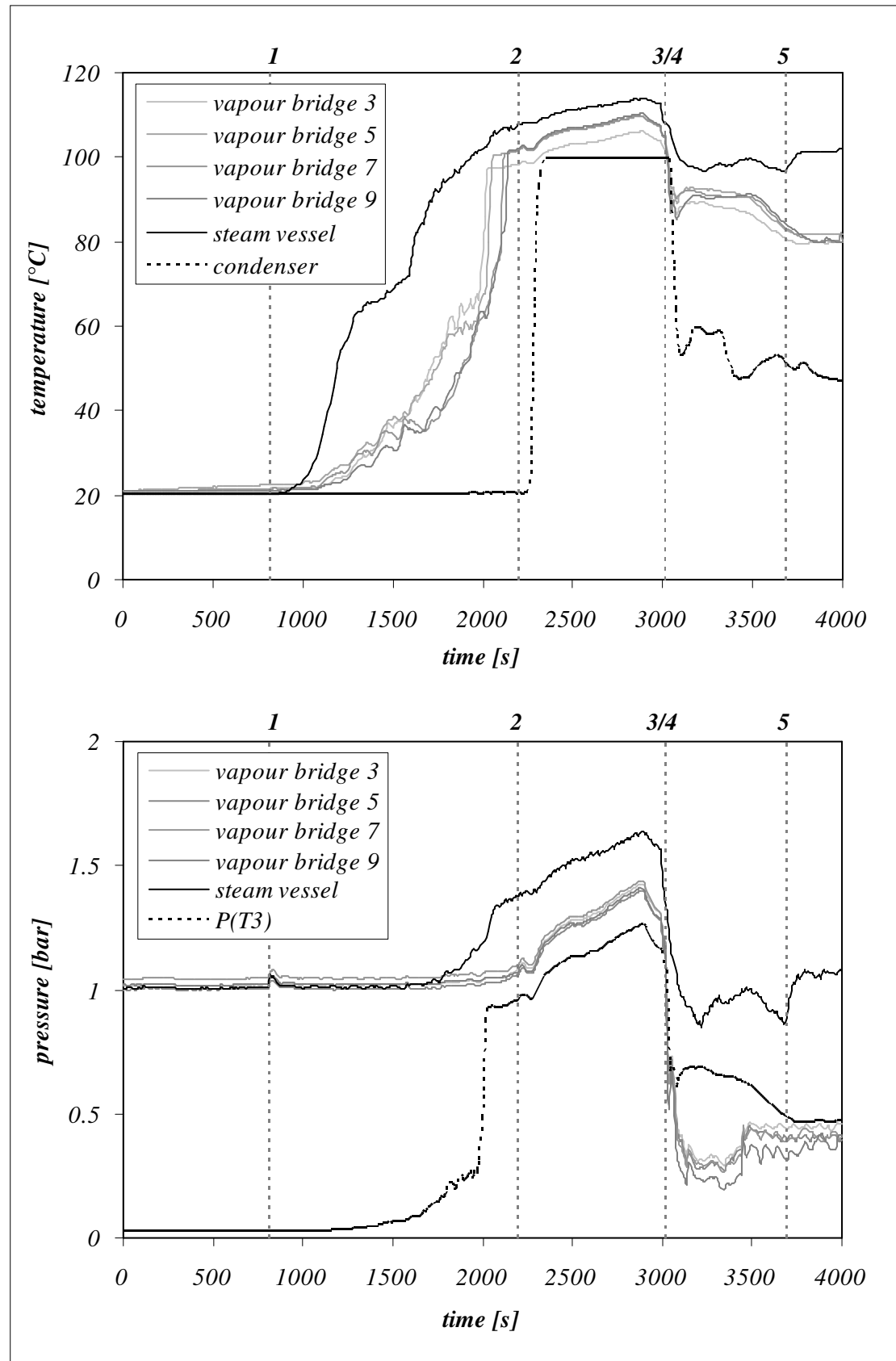


Figure 7.14: Temperatures and pressures during the start up phase of a drying experiment. $P(T3)$ is the free vapour pressure belonging to the temperature in the third vapour bridge. (Note: the steam supply and water supply of the condenser were adjusted a few times.)

7.7.2 Evaluation of the experimental phase

The tests on the Delta dryer prototype enhanced insights into the technical feasibility of the concept. Four conclusions can be drawn.

- Circulation of large quantities of balls through the units involves difficulties that should not be underestimated. The mechanical requirements pose a threat to the process.
- As long as the installation is sufficiently airtight, good drying conditions establish as result of a temperature difference between the balls and the condenser.
- Internal heat leaks from the steam vessel to the condenser contribute significantly to the total energy consumption of the drying process.
- Recovery of heat is impossible in the tested configuration due to the low maximum fill level in the screw conveyors.

Regarding the first conclusion, the prototype displayed its mechanical vulnerability throughout the test phase. Many modifications were made to enhance its robustness, but these paid off only partly. No fundamentally unsolvable problem area was identified though. The transportation of balls through the dryer is considered doable when looking at the components individually. It is rather the thousands of balls per second, their unpredictable behaviour as part of a bulk stream, and the many problem areas that make high demands on the design and workmanship of the installation. Any lack of robustness in the chain of components is bound to result in breakdown, so it is necessary to eliminate all possible failure modes. The latter is something that has just not been practicable so far.

The total number of thermal test hours on the prototype was limited as result of the many mechanical problems faced. Despite the limitations, excellent drying conditions established after a start up period of approximately 30 minutes. It was demonstrated by the low pressure in the screw conveyors compared to the equilibrium vapour pressure belonging to the temperature of the balls, and obviously by the fact that dried material was received at the discharge end.

Beside this positive result, the experiments revealed important shortcomings of the research prototype. After all, the intended operating conditions could not be approached. Two striking issues emerge while evaluating the measurement data. The first issue is the high temperature of the condenser prior to the input of water, which was mentioned in the previous section. The heat loss involved corresponds with the evaporation enthalpy of 0.14kg/s of water. The vapour flow to the vacuum pump adds to this. It is estimated at 0.003kg/s, based on the partial vapour pressure in the condenser and the air leaks according to Fig.7.12. The internal leak streams are unacceptable for an installation with an intended drying capacity of 0.3kg/s.

The leak streams were caused by vapour losses from the steam vessel, by the amount of condensate that travels along with the balls through the lower sluice, and by flashing sluice blades that were only one-sidedly insulated on this particular occasion. (Note that both sides of the sluice blades were thermally insulated in the experiment of Fig.7.14.) Table 7.1 lists the three contributions. The steam losses from the vessel through the sluices were estimated from leak tests (Fig.7.3); the amount of liquid water escaping from the lower sluice was determined from the temperature difference between the screw conveyors and the steam vessel (Fig.7.13 at $t=2500s$); and the losses due to the flashing mechanism simply followed from a heat balance.

Table 7.1: Internal heat losses during the 'steady state' period in Fig.7.13 ($t=3000s$).

<i>source</i>	<i>heat leak [kW]</i>	<i>steam equivalent [kg/s]</i>
<i>steam through sluices</i>	184	0.08
<i>adhered water</i>	<50	<0.02
<i>flashing of sluice rotors</i>	>123	>0.053
<i>destination</i>	<i>heat leak [kW]</i>	<i>steam equivalent [kg/s]</i>
<i>condenser</i>	350	0.15
<i>vacuum pump</i>	7	0.003

The second vital issue that emerged is the absence of pressure differences between the vapour bridges. The measured pressure in each vapour bridge just tended to follow the pressure in the condenser regardless of the local temperature. Hence, no heat was recovered. All vapour disappeared to the condenser.

The poor resistance of the ball bed was the main shortcoming responsible for the absence of the pressure gradient in axial direction of the screw conveyors. The relatively low fill level in the screw conveyors in combination with an unfavourable shape of the bed apparently led to a short cut between ball compartments. To make things worse, this short cut impinged on the pressure gradient over the sluices. The pressure at the entrance and exit of the steam vessel became lower than anticipated, which partly explains the magnitude of the steam losses through the sluices.

Chapter 8: Towards an improved design

The previous chapters were dedicated to various aspects of the Delta drying process. The theoretical potential of the concept was investigated, as well as the sensitivity of the process to various design issues and operational parameters. Laboratory experiments took away uncertainties due to modelling assumptions and unknown physical properties. Results of these experiments leave no room for fundamental objections. They support the confidence in the new process concept.

The development of the research prototype was a first attempt to translate the concept into an installation in steel. Its full-scale dimensions enhanced insights in specific process requirements, revealed strengths and weaknesses of selected hardware, and helped identifying possibilities for improvements. Despite the fact that most process steps actually met the expectations, several weaknesses emerged during the drying exercises. These were not all just early flaws waiting to be solved in future versions: one in particular is the result of a more elemental problem. The observed limitations of the prototype even put question marks to the viability of the configuration that has been the standard since the early stage of the development.

The shortcomings of the prototype are addressed in this chapter. A summary of the basic process steps, successes, and disappointments gives a view upon the present state of technology and upon the amount of progress to be made. Certain room for improvement exists within the constraints of the original design. Beside adaptations to existing hardware, two new development directions are presented. These are promising in a sense that they may exploit the advantages of the concept, while avoiding vital problems of the original design.

The chapter concludes with a short discussion of the thermal and economic potential of the Delta dryer concept.

8.1 Evaluation of the early Delta dryer design

8.1.1 Present state of technology

The Delta dryer, in essence, is an installation that heats metal balls, mixes them with wet cake, divides the hot mixture into compartments, lets the moisture evaporate in multiple pressure stages, and collects the dried product. It then makes released vapour condense onto the return stream of clean balls, carries the balls stream back to the heating stage, and removes the condensate (Fig.8.1). Two early design choices are responsible for the original outline of the dryer. The first decision was to use steam injection to heat the balls to their ultimate temperature, as this method is efficient and universally applicable. Consequently, the heating stage needs to take place in an isolated space, shielded from the low-pressure atmosphere in the rest of the installation. It thus necessitates some type of sluices. The second decision was to accommodate the evaporation and condensation steps in two parallel, interconnected screw conveyors. The elongated ball bed in the screw conveyors was aimed to provide sufficient resistance and allow a pressure differential in axial direction.

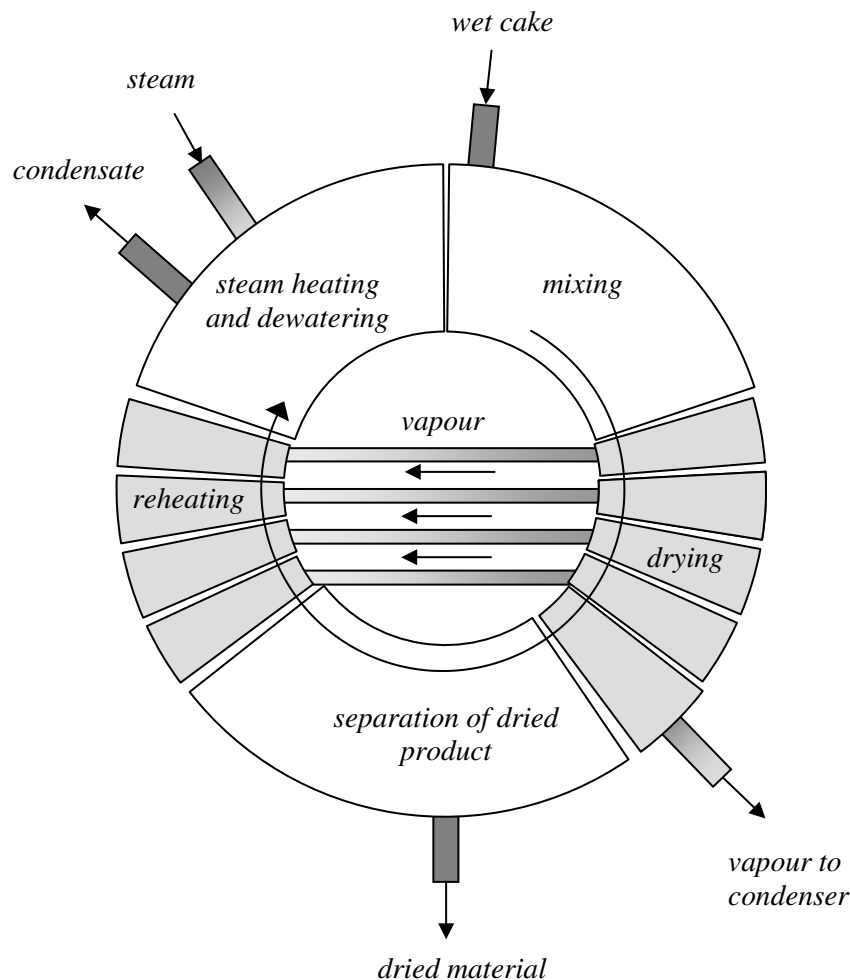


Figure 8.1: Scheme with elementary process steps of the Delta dryer. The balls cycle through the stages while various heat and mass flows are generated.

The Delta dryer technology was developed far enough to consider most process steps of Fig.8.1 either mastered or technically feasible. The research prototype and laboratory experiments provided evidence on the following successes.

- The continuous stream of metal balls can be steam heated to the desired temperature within a few seconds.
- The dewatering effect of balls running over an inclined screen is large enough to remove most condensate.
- A chain tube conveyor can transport the balls in vertical direction, which is necessary because of height losses in other equipment and transitions.
- A chain tube conveyor can create a turbulence in which the balls and the wet feed intensively mix. It makes a separate mixing operation redundant.
- Rotary sluices can carry the balls stream to the steam vessel and vice versa, while hindering vapour flows.
- Despite continuous feed operations and semi-continuous discharge operations, the partial air pressure inside the installation remains below critical limits.
- The vapour pressure difference between the balls-cake mixture and water haze inside the condenser results in excellent drying conditions.
- The large heat-transferring surface area of the balls results in high drying rates compared to conventional contact dryers. This makes it possible, in principle, to separate the stages of the MSFE-process by a packed ball bed of acceptable length.
- Once being dried, the cake is relatively easy to recover from the balls stream by means of a drum screen.

Beside these successes, the experiments raised awareness of some problem areas. Amongst a few minor issues are five shortcomings of concern.

- The low maximum fill level in the screw conveyors results in a highly permeable ball bed. The absence of the MSFE-process wipes out the prospects on any recovery of heat.
- Losses from the steam vessel to the condenser can spoil the internal energy economy of the installation. The rotary sluices of the standard design offer too little resistance to vapour flows.
- Screw conveyors are far from ideal for the transport of steel balls. The tolerances between the screw blades and the hull make them also vulnerable to hard objects that may come along with a low-valued product such as dewatered sludge cake.
- The chain tube conveyor is robust, except for the top part where the chain risks running off the drive wheel.
- Discharge of condensate occurs via an opening in the steam vessel and must be regulated to prevent steam leaks. Standard float control valves are sensitive to clogging.

It should also be mentioned that malfunctioning of the Delta dryer cannot always be put down to a single failing component. As part of a string of critical process steps, a supposedly harmless piece of equipment can have a negative impact on the performance of its colleagues and thus initiate a chain reaction. Several unforeseen series of events were experienced during the drying experiments. One of these, for instance, was triggered by the low fill level in the screw conveyors. The high permeability of the ball bed (a result of the low fill level) made recovery of heat impossible, putting high stress on the condenser. The design capacity of the condenser was too small for this unexpected task, so that the cake dried only partially. As the

finished product was harder to liberate from the balls, a significant fraction of it travelled to the steam vessel, fouling the float control valve. After a while, the float control valve clogged. The drying capacity decreased further and further until the process came to a halt.

Many other failure modes exist because of the number of components involved, and because of the high mechanical and thermal demands. Hence, the equipment should not just perform well during normal operation, but also in worst-case scenarios.

8.1.2 Improvements within the basic design of the prototype

The fact that the research prototype lagged behind expectations had obviously to do with its explorative function. Rather than bringing conclusive results in terms of ultimate drying efficiencies, it was meant to show the working principle and explore the performance of equipment. Many possibilities for improvements were identified. The float control valve, for instance, is replaceable as a whole by an advanced version that can cope with foul water. The rotary sluices, chain tube conveyor, and screw conveyors can be further customised as discussed below.

The disappointing performance of the rotary sluices is the result of a poor compromise between mechanical and thermal requirements. The transport properties are excellent, but obviously at the cost of the quality of the sealing on top of the rotor blades. Seals that are more rigid will reduce steam losses and are considered practicable, although a minimum level of flexibility is required to avoid ball jamming. Less rapidly to implement, but also attractive, is to increase the diameter of the sluices and to reduce the width of the compartments. The geometric change shortens the gap to be bridged by the seals, gives place to a few extra rotor blades, and does not need to affect the transport capacity. Steam losses along the flanks of the blades can be taken care of easily by closing the sides of the sluice compartments.

While the adaptations combined will make the sluices significantly more leak proof, further progress is possible on a level of process operation. The use of lower steam temperatures yields smaller pressure gradients and is profitable here. (Adverse effects of these temperatures on the kinetics of the drying process must be taken into account though.) Since the pressure gradient is partly due to the temperature drop by the input of cold feed, preheating the cake is also worth considering. The excess condensate from the steam vessel, which is readily available, forms a suitable heat source.

If, after all these adaptations, the rotary sluices would still not perform well enough or show too little wear resistance, a range of alternative sluice designs exists that has been left uncovered so far but could be explored as well. The most interesting of these designs are pneumatically operated in a reciprocating mode.

The chain tube conveyor as used in the prototype has one major flaw in that the chain risks running off the drive wheel. For a transport capacity of thousands of balls per second, this risk should not just be minimised but fully eliminated. There are two options to tackle the problem. The first is to ensure that the system can handle a ball stuck getting between the chain and the cogwheel, e.g. by increasing the tolerances within the driveline. Propelling the dishes instead of the chain is an interesting idea, using a cogwheel with bifurcated extensions. A ball that manages to get in between a dish and a fork will cause no problem, as the next dish in line will never miss the following fork. A difficulty of this idea, unfortunately, is the weight of the ball column putting huge stress on the dishes.

The second option is to keep balls away from the danger zone. One of the suggestions is to extend the sides of the cogwheel parallel to the axis like a fan. Most balls falling off the dishes move over the axis now and do not have a chance anymore to reach the spot where the chain links and the cogs meet. Another suggestion is to introduce a second chain wheel, as illustrated in Fig.8.2. The wheel shown in close-up has no cogs and is only to bring the chain into horizontal position. It bears resemblance to the fan-typed chain wheel mentioned before, but without cogs and with a curved race that centres the chain. An ordinary drive wheel, located at the end of the horizontal section, is virtually unreachable for the balls.

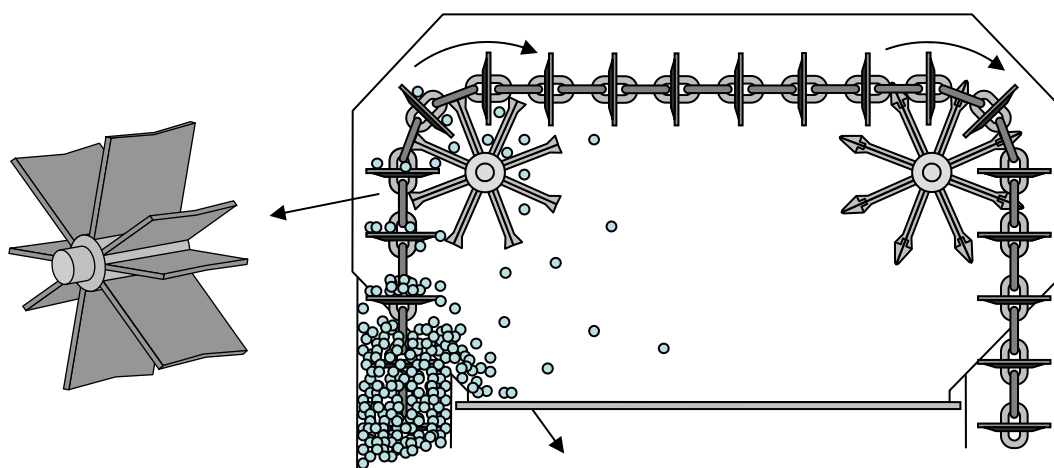


Figure 8.2: Modified drive station of a chain tube conveyor. The two chain wheels interconnect by means of an external synchronising belt.

The screw conveyors, being the most disappointing units of the prototype, display room for improvements as well. Reducing the lift by the propelling side of the screw blades is an option that results in higher maximum fill levels, as fewer balls are paddled over the axis. Smaller pitch lengths and increased diameters of the hull and the axis are profitable in this perspective. They bring down the angle at which the blades push against the balls, and as an additional advantage, elongate the ball bed per unit of conveyor length. Besides, a shorter pitch decreases the cross-sectional area of the bed, whilst a more oversized axis obviously forms a larger obstacle to the balls.

A small downward inclination of the screw conveyors can also help. An inclination angle of 2 degrees, for instance, allowed fill levels up to 46kg per pitch, whereas the maximum value in the horizontal set-up was 41kg per pitch. A rotational frequency of 0.25Hz instead of 0.5Hz even raised the upper limit to almost 50kg per pitch, although this measure halves the drying capacity, increasing heat losses per kilogram of evaporated moisture in a different way.

Despite these achievements, however, it is doubtful if such adaptations will eventually result in a ball bed with sufficient resistance. The disappointing performance of the screw conveyors in the prototype could just be the incidental effect of a poor design, but it is more likely a first sign of their unsuitability for this purpose. The more so as higher fill levels would not solve other issues such as the large manufacturing efforts, the effect of wear on process efficiency, and the mechanical vulnerability of screws.

8.2 Compartmentalization by means of physical barriers

The limitations of the screw conveyors make the Delta dryer in its original configuration very difficult to realize. It raises the question whether the MSFE-process cannot better take place inside an alternative piece of equipment. Finding a suitable replacement for the screw conveyors, however, is not easy, since the mechanical and thermal requirements are stringent and naturally conflicting. On the one hand, the flash stages should be well isolated from each other, allowing just predefined exchanges of vapour. On the other hand, the transports of balls do not benefit from attempts to restrain undesirable vapour leaks. Any seal or tolerance between moving elements poses a threat in the presence of the balls stream.

A few promising development directions were identified that may be sound from a mechanical and thermal perspective. One of these entails the use of physical barriers as a rigorous measure against vapour leaks between the stages of the MSFE-process. The physical barriers may move together with the ball compartments along the evaporation and condensation steps, so that compartmentalisation may only occur once during a full heating cycle or cooling cycle. Alternatively, the barriers may remain stationary. The balls then simply run from one compartment to the next via a series of sluices. In this case, the equipment itself is not involved in the heating and cooling cycle. This is an advantage since the parasitic heat capacity of the equipment would otherwise lower the thermal performance of the dryer.

8.2.1 Continuous operation

The scheme of Fig.8.3 shows a variant with stationary barriers. This variant comprises eight sluices, four chain tube conveyors, a steam vessel, and a screening section, thus creating four MSFE-stages. After leaving the steam vessel, the balls enter a chain tube conveyor, mix with wet feed, and move up, while released vapour escapes through a first vapour bridge. The mixture then tumbles down the second drying stage and passes another chain tube conveyor (the third drying stage). It subsequently enters a screen (fourth drying stage), where remaining moisture evaporates to a condenser and dried cake separates from the balls. Two additional chain tube conveyors bring the balls back to the steam vessel.

The leak streams between the flash stages should be small due to the serial arrangement of the sluices. This could make the presented scheme superior in terms of energy economy, even though the number of flash stages is limited. The physical barriers also put less stress on the evaporation kinetics than the ball bed in the original configuration. As long as they are sufficiently leak proof, the sluices allow lower drying rates, so that heat sources at even lower temperature may be used. The dynamic ball bed also plays a role in this, as it continuously restores the contact between the balls and the partially dried cake. The formation of a boundary layer of air that may obstruct the condensation process should not be a real issue: if necessary, a controlled sweep of the vapour-air mixture from the compartments to the condenser can prevent this.

As mentioned, considerable effort needs to be put into the development of rotary sluices in order to get a grip on their leak resistance. Still, it is an interesting thought to use such sluices in this particular scheme. The four sluice rotors in line could be put together into one housing, yielding a relatively simple and compact installation with only three main drivelines: one shared driveline serving the four parallel chain tube conveyors plus two others for the two sluice-batteries.

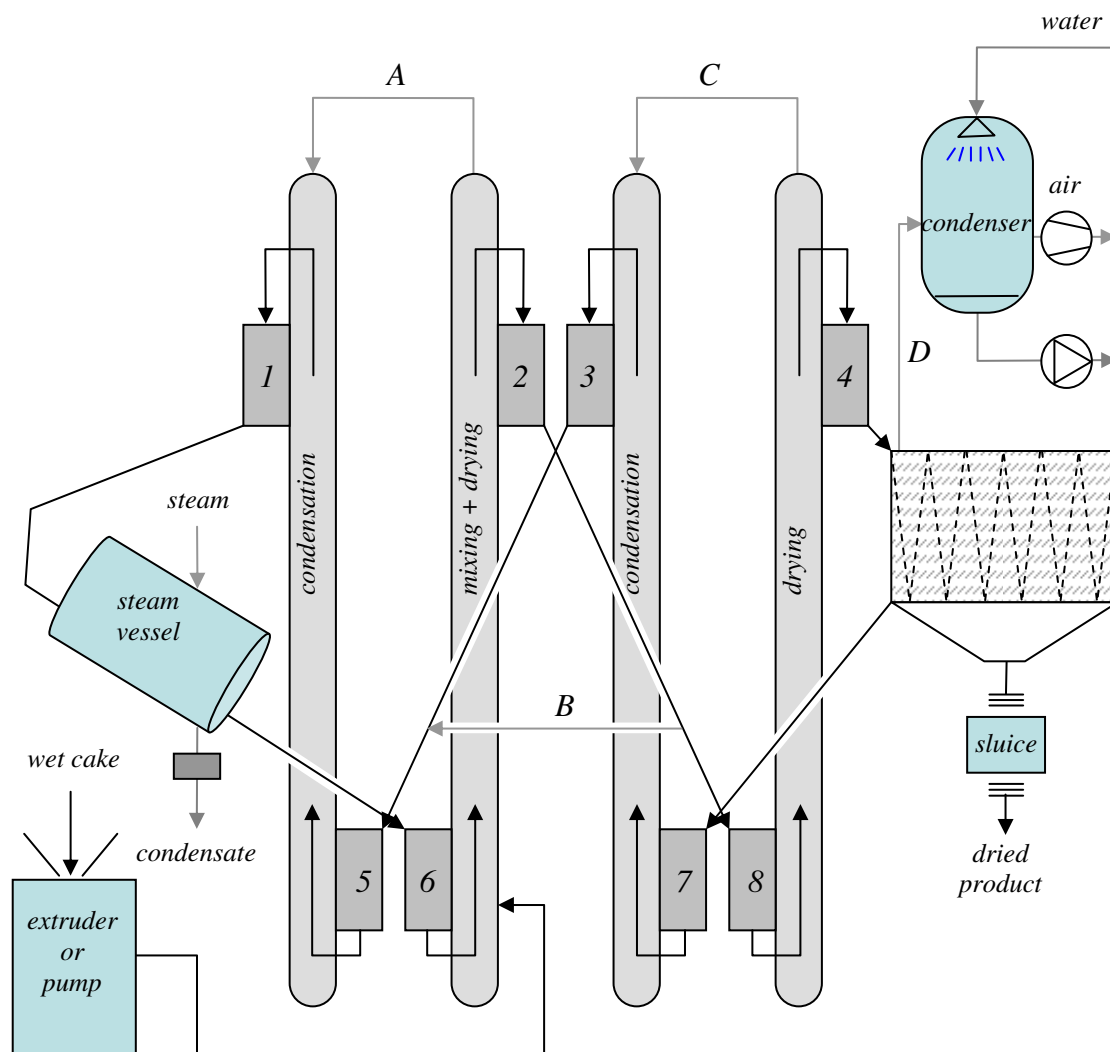


Figure 8.3: Process scheme of a Delta dryer with four chain tube elevators and eight sluices. This arrangement yields four isolated flash stages (A, B, C, D).

8.2.2 Semi-continuous operation

Pneumatically operated slide valves and pinch valves are leak proof and thus more advantageous from a thermodynamic point of view than rotary sluices. As their reciprocating motion makes them robust and reliable, they seem perfectly suitable for the job. Unfortunately, however, their relatively low operating frequencies cause large fluctuations in the balls stream. Buffer volumes would be necessary to arrange constant balls supplies to parts of the installation that function better when running continuously instead of in a (semi-)batch mode. The inlet of the steam vessel is a perfect location for a buffer volume, since an increased residence time in this unit is beneficial anyhow. To some extent, the chain tube conveyors will act as buffers as well, although other pieces of equipment are thinkable of performing better here.

Semi-continuous operation of the Delta dryer process is attractive especially if the compartments are fully emptied and refilled after completion of each flash stage. It allows the use of efficient drying schemes based on the principle of Multiple Stage Double Flash Evaporation. This principle, derived from the well-known MSFE-principle, arranges two vapour bridge connections per compartment, so that every stage flashes twice: first using one vapour bridge (the grey ones in Fig.8.4) and then the other (the black ducts in Fig.8.4). On time-average, half the number of vapour bridges is left unused. These bridges are closed off by means of valves. The result of the double flash is a reduction of the theoretical minimum power consumption by a factor of two compared to the standard arrangement with equal number of compartments (Equation 3.1).

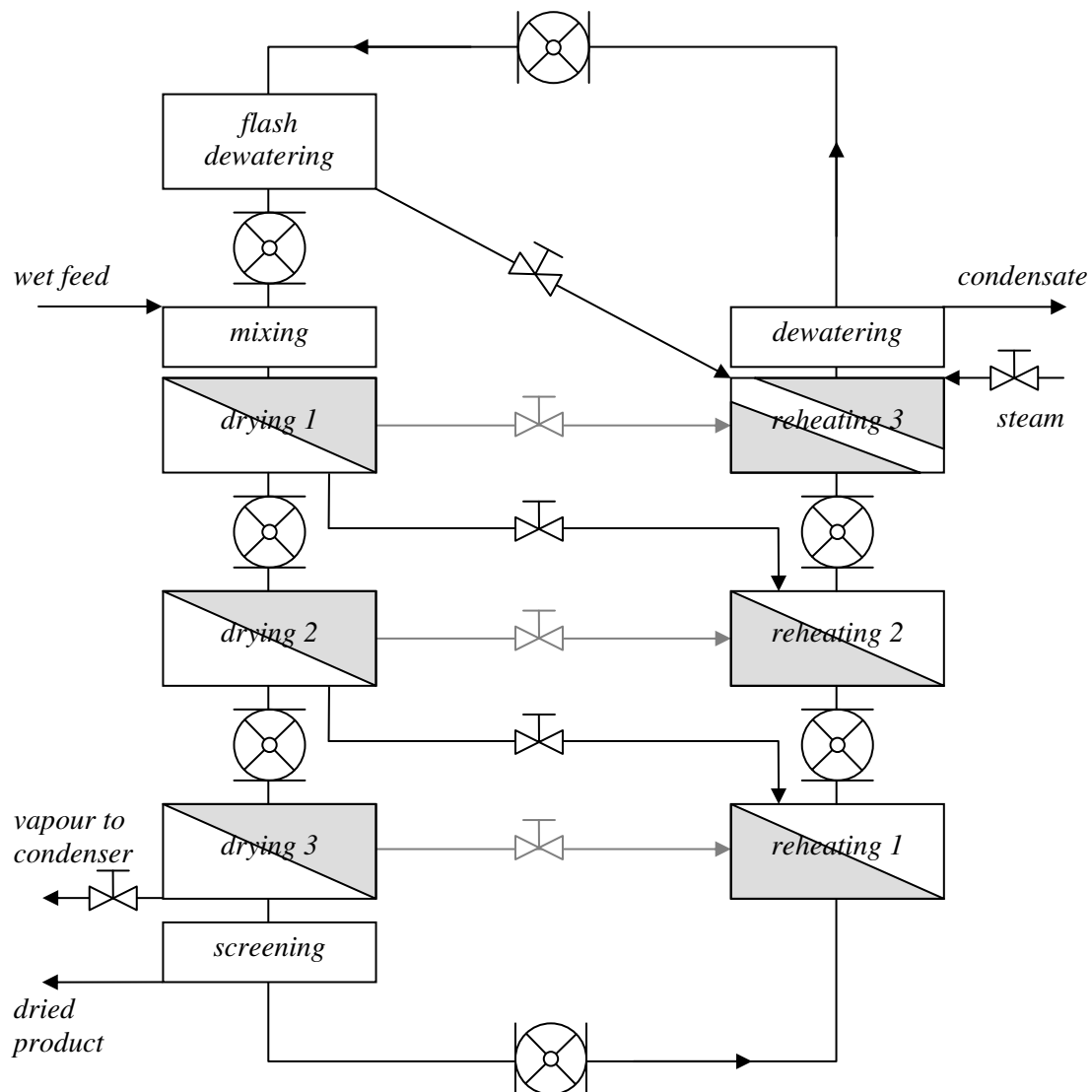


Figure 8.4: Scheme of an MSDFE-process in which the various vapour ducts open and close at predefined time intervals. The scheme includes a final heating stage comprising three steps: it uses vapour from the first drying stage, then flash steam from remaining liquid after the dewatering stage, and finally some steam from an external source. This allows the heat of evaporation of the remaining water to be recovered at the cost of just one extra compartment.

8.3 Compartmentalization inside a chain tube conveyor

A second new development direction entails the use of a chain tube conveyor to accommodate the MSFE-process (Fig.8.5). The resistance of the ball bed separates the flash stages, just as in the early configuration based on screw conveyors. A large chain tube conveyor mixes the balls with cake and allows the mixture to dry as it moves towards the drum screen. After leaving the drum screen, the balls go down via the parallel conveyor tube and reheat by released vapour from the cake. A second elevator compensates the height loss in the sluices and the steam vessel. It helps reducing the vertical dimensions of the installation.

A chain tube conveyor as alternative to the screw conveyors is interesting, as it could result in a dryer that excels in simplicity, compactness, and low production costs. The installation can also outperform the original design mechanically. The next paragraph elaborates on this configuration, showing elementary features, dimensional aspects, design choices, and a few problem areas.

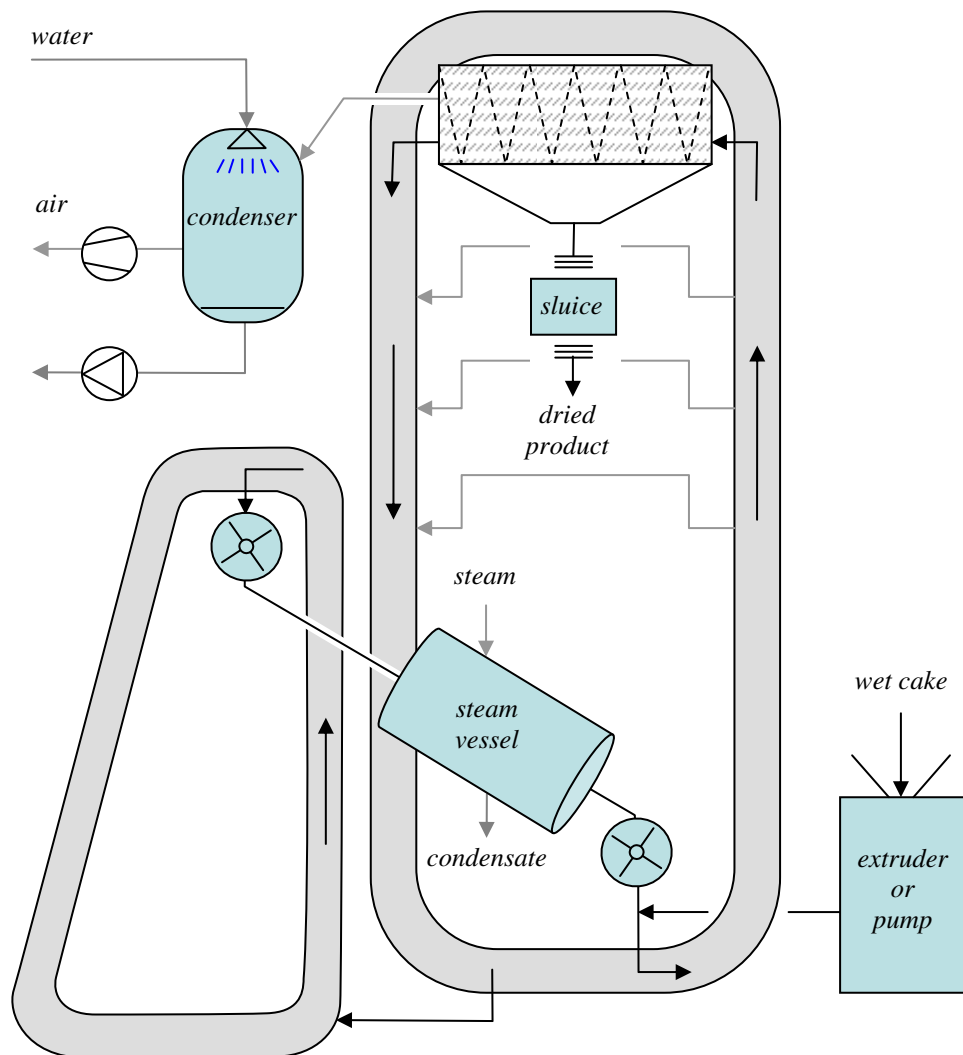


Figure 8.5: Scheme of a Delta dryer in which a large chain tube conveyor accommodates the MSFE-process.

8.3.1 Resistance inside the vertical sections of a chain tube conveyor

Recovery of heat within the chain tube conveyors is not a matter of course. A crucial condition is the presence of a pressure gradient in axial direction of the vertical pipe sections. In analogy to the intended conditions in the screw conveyors of the original design, the pressure gradient should naturally exist due to the changing temperature of the ball bed. In order to sustain the gradient, the resistance to vapour flows must be large enough, the leak streams towards the condenser remaining acceptably small.

The resistance inside the vertical tube sections depends on the balls content and the presence of the dishes. As mentioned in Paragraph 7.4, the diameter of the dishes is significantly smaller than the inner diameter of the tube, guaranteeing a minimum fill level in the up going part of the apparatus. It makes the balls gather until the column reaches the top of the vertical pipe. When in operation, the length of the ball bed is thus constant. The void fraction is not necessarily fixed. Further addition of balls can decrease empty volumes underneath the dishes and make the bed more compact.

To check the permeability of the ball bed and dishes, two intersections were made halfway down the 6m long vertical pipe of the chain tube conveyor in the research prototype. One of these intersections led to a vacuum pump, whose discharge airflow was measured. The other was equipped with a pressure transmitter. Large openings at the top and bottom of the elevator ensured that the ends of the ball bed remained at atmospheric pressure, while the vacuum pump generated airflow, causing lower pressure in the middle.

Table 8.1 presents results of the measurements for a static bed and a moving bed. The last two columns of the table contain superficial flow velocities and values for the void fraction that were calculated as if the results would relate to a uniformly packed bed without dishes. Whereas the actual void fraction in the conveyor tube was 0.45 to 0.5, the apparent void fraction approached values for a loosely packed ball bed. One would expect the values to become even lower for a static bed, since the absence of turbulence increases the density of the bed. This, however, was not observed. It did not make a significant difference whether the installation ran or stood still.

Table 8.1: Pressure gradient and airflow through a 3m long ball bed inside the chain tube conveyor of the prototype. The last column contains apparent void fractions for the chain tube conveyor and for the simulated screw conveyors that were analysed in Chapter 5.

	ΔP [mbar]	airflow [m ³ /s]	$v_{\text{superficial}}$ [m/s]	'void fraction'
<i>elevator, static bed</i>	3.5±0.5	0.023	0.4	0.4
<i>elevator, dishes moving up at 0.32m/s</i>	3.0±0.5	0.023	0.4	0.4
<i>screw conveyor, 2mm gap between blade and hull</i>				0.5
<i>screw conveyors, 2mm gap bridged by a brush (reference simulation)</i>				0.4

8.3.2 Dimensional aspects

When expressed in terms of apparent void fraction, the measured permeability in the vertical pipe section of the chain tube conveyor matches the overall permeability in the simulated screw conveyors that served as a reference in Chapter 5 (see Table 8.1). These screws performed well on paper. A chain tube conveyor with the same effective dimensions, i.e. a one-sided length of 15m, could thus successfully replace them. Unlike the resistance in the screw conveyors, the resistance in the chain tube conveyors does not rely on narrow tolerances, so the energy economy will not drop because of wear.

Whether or not high drying efficiencies can be obtained with the new design depends on a few uncertainties. One of those concerns the permeability of the ball column that moves in downward direction. (It could not be determined using the existing set-up.) Part of the content will drop through the gap between the dishes and the inner wall of the tube, so that the average vertical velocity of the balls stream is larger than the velocity of the dishes. At equal dimensions, the fill level and the resistance to vapour flows will be lower than at the other side of the chain tube conveyor where the balls move upwards.

The dimensions of the vertical tube section should be chosen to minimise the influence of the transport direction on the permeability of the ball bed. Reducing the local diameter of the piping in relation to the diameter of the dishes is a first step. The gap between the dishes and the inner hull should just be wide enough to avoid mechanical problems. Lowering the vapour bridge connections on the descending side is another change for the better: it increases the effectual length of the bed, as the area below the last vapour bridge is functionless anyway. In addition to these measures, it is obviously possible to increase the length of the chain tube conveyor, or to arrange three elevators in a series as visualised in the scheme of Fig.8.6. The latter requires additional equipment and involves more transitions. On the other hand, it allows one of the units to be assigned exclusively to the mixing operation. It also decreases the overall height of the chain tube conveyors, so that the core of the installation may fit inside a vertical sea container.

Key factors other than the permeability of the ball bed are the evaporation capacity and the residence time of the cake inside the chain tube conveyor. These parameters are interlinked by the length and diameter of the vertical pipe sections, by the velocity of the balls mixture, and by the balls-cake ratio. The residence time should be long enough to let the product dry to the desired moisture content and at acceptably low temperature gradients. The evaporation capacity in relation to the heat losses determines the thermal performance of the installation as well as the operating costs.

An important point of attention is the cross-sectional ball bed area in the elevator of the research prototype being twice as large as the ball bed area in the screw conveyors, while the balls throughput is the same. Hence, it will be difficult to reach similar thermal efficiencies by simply translating the effective length of the simulated screws to the chain tube conveyor in the new design. In fact, Ergun's relationship suggests that the vapour flow rate through the bed is just about reversely proportional to the square root of the length of the ball bed. (An increase of the cross-sectional area with a factor of two requires a ball column of approximately four times as long.) The chain tube conveyor to be developed should therefore exhibit high values for the quotient between drying capacity and ball bed area. This necessitates high chain velocities, and is another reason to narrow the gap between the dishes and the inner hull.

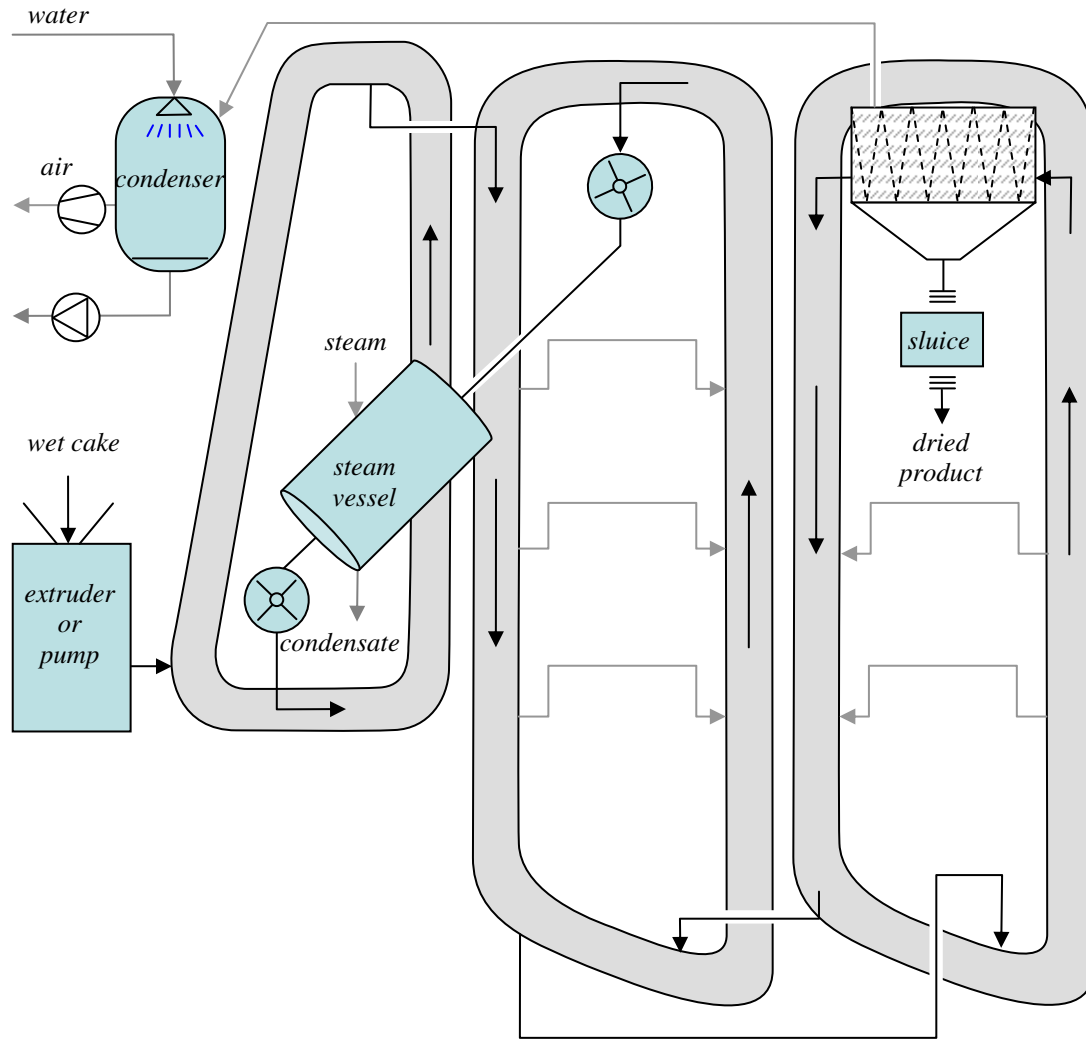


Figure 8.6: Scheme of a Delta dryer with three chain tube conveyors. The small conveyor is responsible for the mixing operation. The two others accommodate the MSFE-process.

8.3.3 Design implications

Fig.8.7 shows an artist impression of an installation based on the scheme of Fig.8.5. Whereas the small chain tube conveyor, the steam vessel, and the rotary sluices are roughly comparable to those of the research prototype, the large conveyor is a modified unit. It is heavily reinforced, comprising a second chain wheel to eliminate excessive friction between the dishes and the tube (see also Fig.8.2). The feed entrance is positioned between the lower sluice and the conveyor.

The large chain tube conveyor contains four vapour bridges including the connection to the condenser, and excluding the lower bend of the conveyor. There are no vapour bridges located in the region where the sludge has already dried substantially. This is because of the dust that can form and carry along with the gas stream. Those bridges would contribute only very little anyhow, since evaporation rates at the end of the drying process are relatively low. Tests confirmed that the vapour bridge openings form no obstacles to the balls transport, which is thanks to the large tolerances between the dishes and the inner wall.

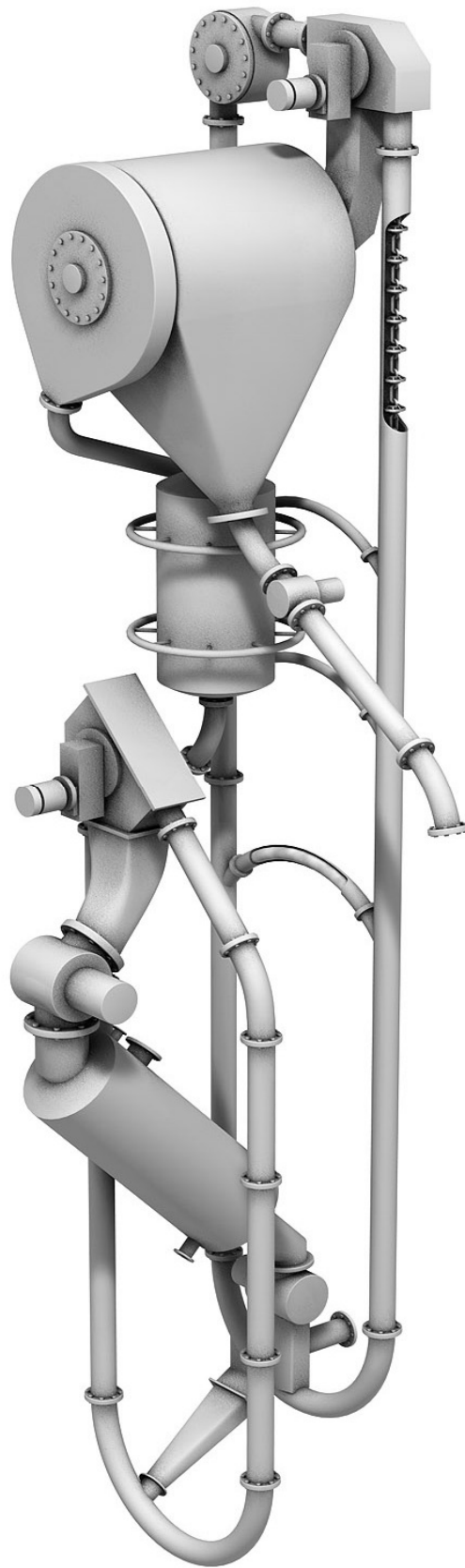


Figure 8.7: Rendering of a Delta dryer based on chain tube conveyors (with permission of Van Tongeren Kennemer BV).

As essential difference between the presented design and the early configuration, the dishes and the chain move together with the balls along the heating and cooling cycles. (The screw blades obviously never left their place.) The dishes will collect water when passing through the condensation stages, but unlike the balls, do not dewater at any point. They moisturise incoming fresh feed during a successive mixing operation, thus reducing the process efficiency.

The shape and surface condition of the dishes should facilitate the condensate to trickle off in order to minimise the amount of liquid that circulates within the system. A brush station may be necessary to remove dirt from the dishes and reduce the uptake of moisture. A logical location for this cleaning station exists between the chain wheels of the central conveyor (Fig.8.6).

Apart from the dishes, the balls should be dirt-free when leaving the drum screen. This is to avoid fouling problems down the line, as the design offers no room for a sprinkler installation. A cleaning step may be superfluous when adapting the geometry and revolution speed of the drum screen to the specific demands of the product. An additional screening operation can be helpful if the balls stream contains larger pieces as well. It avoids accumulation of re-circulated coarse fractions that may incidentally come along with the cake and do not pass the regular finer-meshed screen. It also keeps hard oversized particles away from the tolerances of the sluices.

8.4 Thermal and economic potential of the Delta dryer concept

As mentioned, the thermal potential of the Delta dryer concept depends primarily on the vapour losses within the stages of the MSFE-process. Secondary issues are the recovery of dried product, the magnitude of steam leaks through the sluices, and the amount of condensate travelling along with the balls as they leave the steam vessel. A rough performance estimate for a well-improved installation can be made based on Equation 8.1 and the assumptions below. These assumptions result in specific energy consumptions (e) at the amount of 2MJ per kilogram of evaporated cake moisture.

$$e = (A + B) \cdot (1 + C + D + E) \cdot \frac{1}{F} \cdot \left(1 + G \cdot \frac{Cp_{water} \cdot (T_{steam} - T_{room})}{H_{evap}} \right) \quad [8.1]$$

A: Theoretical energy consumption of the MSFE-process. According to the simulation results presented in Chapter 5, values as low as 1MJ per kg of evaporated cake moisture come within reach when using the resistance of a packed ball bed to separate the flash stages. The drying capacity and overall resistance of the ball bed, however, must be comparable to those of the reference simulation discussed in Paragraph 5.2.

Similar theoretical minima are applicable when using physical boundaries between the flash stages: according to the grey line in the graph of Fig.3.7, a scheme with four ideally separated flash stages (three vapour bridges as in Fig.8.3) yields energy consumptions around 1MJ/kg.

B: Contribution to the energy consumption as result of the temperature difference between balls of the first cooling step and last heating step. The temperature gradient is required to realize acceptable evaporation and condensation rates. It corresponds with energy consumptions around 0.3MJ per kg of evaporated moisture, based on the analysis in Paragraph 6.2.

C: Steam losses through the sluices in relation to the drying capacity. The pressure differential across the sluices is below 0.5bar under normal operating conditions. It would have resulted in steam losses around 0.1kg per kg of evaporated cake moisture for each of the sluices in the research prototype (Paragraph 7.3). Reduction of these losses by at least a factor of two is considered feasible using improved sluices.

D: Amount of condensate travelling together with the balls through the lower rotary sluice. A representative value for *D* is 0.2kg per kg of evaporated cake moisture, according to the model of Paragraph 6.1. It may be possible to reduce this number to a minimum of 0.1kg/kg.

E: Moisture uptake by the dishes of a chain tube conveyor in case that conveyor accommodates both evaporation stages and condensation stages. It is assumed to be 20% of the condensate formed in this unit, so that *E* approximates $0.2 \cdot (A+B)/H_{evap}$.

F: Fraction of dried solids that is actually recovered from the balls. This fraction is less than unity because of the limited efficiency of the screening operation as well as the dust that may form and travel through the vapour bridges or towards the condenser. It is approximately 0.9.

G: Fraction of hot condensate from the steam vessel that is not reused in the process. Energy consumptions increase if the output of the steam vessel is just disposed off. This factor is ignored in the calculations.

Given the present state of the Delta dryer technology, it is premature to make accurate predictions on its economic potential. Based on an energy consumption of 2MJ/kg and approximations for capital investments, life expectancy, and maintenance intensity, a rough cost estimate was made. It is presented in the graph of Fig.8.8 together with the costs of conventional drying techniques discussed in Chapter 2.

The graph shows that the Delta dryer could economically compete with the best commercial techniques. A few remarks are due here though that let the Delta dryer score even better. To begin with, the considered installation has a capacity of only 1.5 tons of water per hour, which makes it perfectly suitable for small-scale, decentralized drying of e.g. sewage cake. It is not fair to judge this installation against drying plants with evaporation capacities of four times as large (such as the Sulzer installation in Beverwijk), since up scaling results in reduced labour and maintenance costs and often in lower capital investments per unit of dried material. When comparing the expenses, it would at least be reasonable to take into account the distinct settings and to consider logistic differences between centralised and decentralised drying.

A second argument is that the costs of steam production are based on a generator with a standard efficiency of 80% and a gas price of €5.60 per GJ. It does not show from the graph that the Delta drying technique offers unique opportunities to use low-grade heat sources. Local availability of rest heat would bring a substantial change to the graph in favour of the Delta dryer.

Finally, the calculations underlying Fig.8.8 disregard infrastructural aspects and space requirements; neither do they reckon with equipment and labour that are necessary for secondary operations such as exhaust gas cleaning, preconditioning of the feed, and aftercare of dried product. These operations are known to contribute significantly to the total drying costs of conventional plants, but not so to the costs of the Delta dryer.

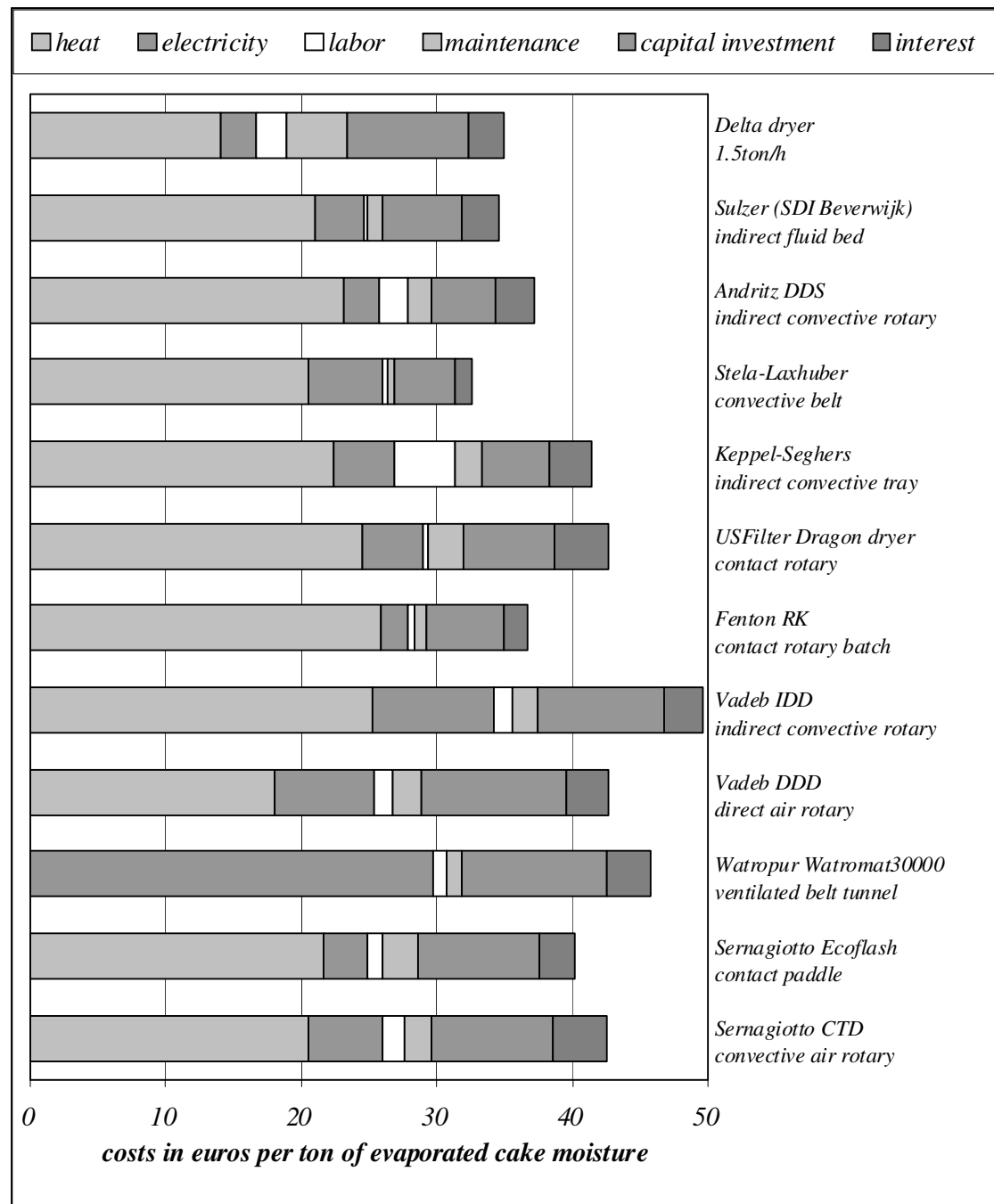


Figure 8.8: Cost comparison between the Delta dryer and the techniques discussed in Chapter 2 [Zonneveld, 2005], [Kiezewetter, 2002].

Chapter 9: Conclusions and recommendations

The thermodynamics of the multiple stage flash evaporation process, the metal balls acting as a drying medium, and the resistance of the packed bed separating the various flash stages are unique features of the Delta dryer concept. They can eliminate drawbacks of existing drying techniques and shift thermal efficiencies toward higher levels.

The work described in the previous chapters enhanced insights in the Delta drying process. The theoretical potential of the concept has become evident, as well as the practical hurdles that handicap the chances of success of an operational installation. The first paragraph of this final chapter summarises all main conclusions based on the findings described. These conclusions are presented per chapter. Since this dissertation does not pretend to cover every aspect involved, it leaves some questions open to further research. The second paragraph makes suggestions for future work that can provide the answers and bring the concept towards commercialization.

9.1 Conclusions

Chapter 1

The production of watery waste streams is significant. Treatment of these streams is self-evident. It solves problems associated with storage and disposal, and yields valuable resources. Of recent interest is the use of wet organic residues for the production of bio-fuel, biogas and green energy, which saves fossil fuels and energy crops. Most conversion routes, such as flash pyrolysis, involve a thermal drying step. In order to make the treatments attractive, this thermal drying step needs to be energy-efficient and cost-effective.

Chapter 2

Existing drying techniques use 3-6MJ per kg of evaporated water. A few exceptions aside, these techniques have ancient roots. They evolved over many years through a series of incremental improvements and approach their upper limits of performance. Substantial progress in terms of energy economy therefore relies on system innovations. Unfortunately, newly developed technologies often run up against added complexities, large (and more risky) investments, and technical constraints that make them commercially less attractive.

Chapter 3

The Delta drying concept uses hot metal balls as heat transferring medium and vacuum drying to reduce off-gas and to recycle part of the heat of evaporation. This combination can avoid disadvantages of existing techniques. The thermodynamic principle laying at the basis of the concept raises the theoretically upper limit of performance to high levels.

Chapter 4

Mechanically dewatered sludge cakes are difficult to dry by thermal means, as their physical properties are unfavourable. The Delta drying technique addresses this issue.

- Whereas other techniques are susceptible to the way the sludge transforms from a wet substance through a sticky phase into a solid, the Delta dryer benefits from the changing consistency of the cake. During the initial phase, when the cake is wet and sticky, the material spreads out over the large heat-transferring surface of the balls. At the end of the process, when the dried material detaches from the balls, more than 90% of the product can be recovered with relatively little effort.
- The large surface area of the balls allows the cake to dry in layer thicknesses of typically 0.5mm, while keeping the installation compact in terms of dimensions. These thin layers are advantageous since heat transfer within partially dried porous media is generally poor. The responsible heat and mass transfer mechanisms are comparable to those observed in high intensity drying of paper. They result in heat transfer rates up to 5000W/m²K, depending on the composition of the cake. Hence, the process can take place at relatively low driving forces. The ultimate temperature of the balls can be as low as 100°C, so that a wide range of heat sources can be applied.
- The thermal efficiency of the process decreases significantly in case the feed is strongly hygroscopic. This, however, appears of no concern for the investigated sludge types. The hygroscopic activity of these samples has very little effect as long as the solids content of the finished product is below 90%.

Chapter 5

The kinetic experiments described in Chapter 4 indicate that a total drying time of one minute is realistic. This period is short enough to allow the stages of the MSFE-process to be separated by the resistance of an elongated packed ball bed. The required length of the bed remains practicable and could, in principle, be arranged in two parallel screw conveyors that are interconnected by means of vapour bridges.

The thermal potential of this configuration with two screw conveyors was investigated with the help of a detailed physical model, the results showing more than 50% of the heat of evaporation to be recovered. Major issues for realizing minimal energy consumptions are the resistances of the ball bed segments and of the slit between the screw blades and the conveyor hull. These resistances depend on the geometry of the screws, on the applied tolerances and seals, and on the fill level of the balls-cake mixture. They are to cut down leak streams to an acceptable level. At the same time, the drying capacity should be large enough to generate a pressure gradient along the axial direction of screws.

Since the MSFE-process relies on temperature induced vapour pressure differences, the presence of air leaks and the hygroscopic nature of the cake can be detrimental to the thermal performance of the Delta dryer. These, however, appeared non-critical at realistic circumstances. The number of vapour bridges (twelve of them were defined in standard simulations) is neither a big issue. It is not worth using more than eight vapour bridges, as the last ones contribute very little to the process. A sprinkler installation, which may be considered to clean the balls after the screening step, is not recommended. It gives rise to significant internal heat leaks. Hence, the screening step must be efficient, making an additional cleaning stage superfluous.

Chapter 6

Increased drying capacities result in reduced heat losses per unit of evaporated water, thus improving process efficiencies. On the other hand, higher drying rates require larger driving forces. They involve larger temperature gradients within the flash stages, and cost energy as well. The predicted energy consumptions in Chapter 5 were based on infinite evaporation and condensation kinetics. When taking into account all heat resistances within the flash stages, these energy consumptions increase by 0.15-0.3MJ per kg of evaporated water, depending on the substance to be dried.

While focusing on the MSFE-process, the model of Chapter 5 ignored the final heating and dewatering stage of the balls. This stage, which takes place inside a steam vessel, is essential. Most condensate should be removed before the balls are reintroduced to the drying loop. Residual water travelling along with the balls causes a heat loss of 0.3-0.6MJ per kg of evaporated cake moisture. (Part of the heat can be recovered in the MSFE-process though.) In order to attain low loss values, it pays to increase the residence time inside the steam vessel and to remove the balls quickly after the dewatering action.

Chapter 7

Tests performed on a research prototype confirmed the balls to heat up and dewater inside the steam vessel, and to mix with the cake during their vertical transport inside a chain tube conveyor. Despite the fact that good drying conditions established, the installation could not provide practical evidence on the working principle of the MSFE-process. Instead, it showed several difficulties during its development, demonstrating the poor technical feasibility of a configuration based on screw conveyors.

- Circulation of the bulk mass along the process steps is a difficult task. Thousands of balls need to pass transitions and moving objects in every second. The unpredictability of their individual paths results in many possible failure modes, making the risk of jamming and accumulation hard to eliminate.
- Internal vapour leaks from the steam vessel to the condenser pose a threat to the energy economy of the installation. Measures against these leaks are problematic without compromising the mechanical robustness of the equipment.
- Screw conveyors proved unsuitable to accommodate the MSFE-process. Their maximum fill level is too low, yielding a highly permeable ball bed.

Chapter 8

The Delta drying technique can dry material at a heat input that is less than the heat of evaporation of the water that is removed. Based on the present knowledge, realistic energy consumptions are around 2MJ per kg of evaporated water. A rough cost analysis shows that the technology can be economically competitive with the best commercial installations.

Although the original design seems promising on paper, the process requirements complicate the development of a practical installation. The difficulties are serious enough to abandon this design and explore alternative directions based on the same thermodynamic principle. One of the presented alternatives uses the resistance of the ball bed inside a chain tube conveyor in order to separate the flash stages. It is interesting but shows too many obstacles to be a serious contender at this point. Another, more promising development direction uses a relatively small number of flash stages and a series of physical barriers (sluices) to minimise leak streams between them. It brings mechanical, kinetic, and thermodynamic benefits over the original design.

- Transports of balls between the flash stages can rely on chain tube conveyors and gravity. The equipment was used successfully in the research prototype.
- The dynamic bed inside the chain tube conveyor intensifies the contact between the balls and the cake. It will enhance evaporation rates and allow non-sticky fines to dry as well (although dust formation may be an issue).
- Just four perfectly isolated flash stages, i.e. three vapour bridges, would yield drying efficiencies that are comparable to the theoretical efficiency of the early configuration. Enhanced process stability ensures that energy consumptions do not increase much if the dryer is operated below its design capacity or if drying times are increased.

9.2 Recommendations

The large heat transferring area of the balls makes the cake spread out in thin layers and dry up rapidly with relatively little effort. It provides the opportunity to use low-quality rest heat, which is one of the main assets of the Delta dryer. Until now, the Delta drying process was assumed to take place preferably at a temperature range of 100°C to 50°C. Investigation of most economic combinations of drying temperature and drying capacity as a function of feed properties, design specifications, and other operational parameters requires an advanced model in which thermodynamic aspects, kinetic issues, and practical imperfections of equipment are combined. Such a model could have the same structure as the local equilibrium model of Chapter 5, with the gas temperature introduced as an extra variable of each section.

For the early configuration based on screw conveyors, the advanced model will show the temperature range of 100°C to 50°C to be close to optimal. Lower temperatures are harmful for the process kinetics, whereas higher values are disadvantageous from a thermodynamic perspective, causing increased vapour pressure gradients and thus volumetric leak streams through the ball bed (Section 6.2.2). A design based on physical barriers, however, opens new perspectives. Sluices between the flash stages improve process stability, allowing a wider range of operational settings. It is therefore worth the effort to perform additional kinetic experiments at various temperatures using the laboratory set-up of Paragraph 4.3. Based on the experimental results, it can be decided whether to increase steam temperatures (in order to raise drying capacities), or whether to exploit alternative heat sources such as geothermal heat at 80°C.

The set-up of Paragraph 4.3 can also be used to investigate the relationship between drying rate and temperature gradient. This relationship was assumed to be linear up to a certain threshold (Section 4.3.3). It is of scientific relevance, since little is known yet of the heat and mass transfer mechanisms involved. Besides, kinetic experiments can be part of a standard test procedure that serves to determine if a particular sludge sample can be dried successfully using the Delta drying technique.

The design based on physical barriers seems most promising of all evaluated configurations. This variant could bring the Delta dryer technology closer to a commercial installation. Still, it requires further investigation and development. To begin with, some important parts must be re-engineered up to desired standards. Special attention must be paid to abrasion resistance as well. A pilot installation is then to provide answers to questions regarding long term reliability, wear, dust formation, composition of output streams, energy economy and operating costs of a real drying facility.

While further developing the technology, the Delta dryer should not be seen as an isolated unit but as a link in a series of operations. It is therefore wise to follow the emerging biorefinery sector, as production of fuels and other valuable products from organic residues raises the need for efficient (decentralised) dryers. The Delta dryer seems economically viable and ecologically sound for applications in this field.

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