Model Development and Simulating of a Spinning Cone Evaporator

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Massey University
Palmerston North
New Zealand

Xizhong Zhu

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Abstract

The idea of milk pre-concentration at the farm has attracted worldwide interest for many years. A new pilot-scale evaporator (called spinning cone evaporator), which can be operated on the farm and has a compact and efficient design, has been developed at Massey University. However, there is a shortage of knowledge on the design, operation and control of this new evaporator. The main goal of this thesis is to develop a dynamic mathematical model in order to better utilize this evaporator and make further developments.

This thesis consists of three parts. Firstly, a first-principles model of a pilot scale spinning cone evaporator is developed using the sub-system modelling techniques of the evaporator from the Laws of Thermodynamics and the general mass and energy balances. The model is dynamic and includes the evaporator, the compressor, the condenser and the product transport sections. The system model describes the dynamic relationships between the input variables (cooling water flowrate, M_c, speed of compressor, N_{comp}, feed flowrate, M_f, feed temperature, T_f and mass composition of feed dry matter, w_f) and the output variables (outlet temperature of cooling water, T_{co}, evaporating temperature, T_e, mass composition of product dry matter, w_p and product flowrate, M_p),

Secondly, the evaporator model was implemented using the software package Matlab along with its dynamic simulation environment Simulink. The differential equations for the evaporator model are embedded in a block diagram representation of the evaporator system. The evaporator Simulink model is divided into three levels, the blocks at the top represent the overall model and global constants used in it. The second level contains the individual sub-systems and the bottom level elements within each sub-system. Results of the model verification are satisfactory.

Finally, the model validation is presented for both steady state and dynamic comparisons. The product flowrate (except in the case of feed temperature changes) and evaporation temperature can be predicted at a given time, and the outlet temperature of cooling water and product dry matter composition can also be predicted at a steady state. It can be seen that the results predicted using this spinning cone evaporator model, which accounts for the varying concentrate flowrate and evaporation temperature with time, are in good agreement with experimental data. This model provides a valuable tool to predict performance in a spinning cone evaporator and to modify the design parameters.

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Chapter 1 Introduction

1.1 Introduction

Evaporation technology is an intensively used process in the dairy industry because it is well suited for concentrating food solutions (Kessler, 1981). The process usually consists of two stages with an evaporator concentrating the milk to a critical dry matter content and then a spray dryer removing the remaining water. Various types of evaporators have been used as part of the powder production process to make whole milk, powder, skim milk, lactose, whey protein concentrates, in fact a very wide range products. Nowadays, the dominant evaporators used in New Zealand are the falling film evaporators.

Although the falling film evaporators are tending to become larger and despite the efforts to improve the efficiency of the process, the evaporation process is still costly. The cost of the bulk material to be stored and transported, however, is still considerably reduced. The conventional evaporation system in the dairy industry is operated as follows: on the farm, the fresh whole milk is first collected in the buffer tank, then is chilled in a heat exchanger and then passed to the storage tank. Finally the whole milk is transported by the tanker to the dairy factory for further processing. The system is shown in Figure 1.1.

With increases in the cost of transporting milk between farms and dairy factories, the concept of the on-farm evaporation system is looking more attractive. When milk can be concentrated on farm and the pre-concentrated milk transported to a factory there will be many benefits; including reduced transportation cost, factory processing cost and effluent, overall energy requirements and refrigeration cost on the farm. If a suitable design is used, taking into account the whole energy consumption on the farm, an evaporation system would be an effective method of on-farm concentration (Jebson et al., 1993).

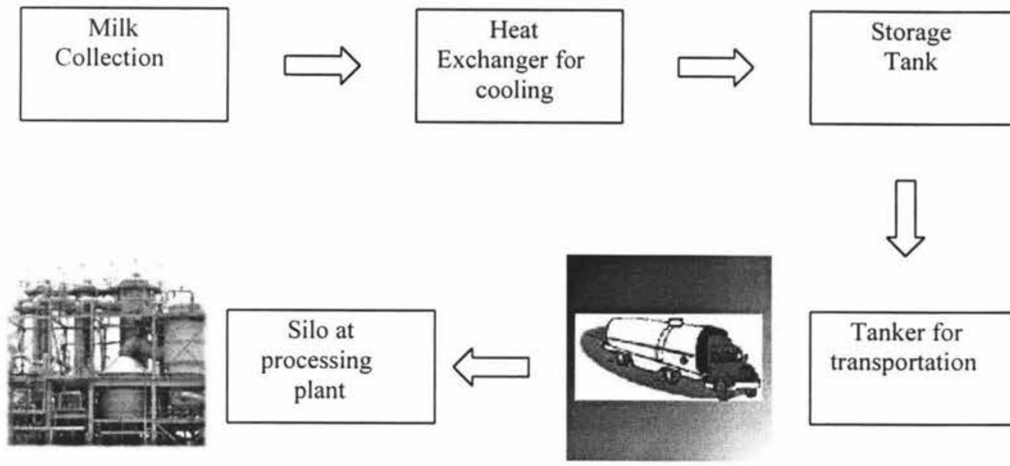


Figure 1.1: The convenient evaporation system in the dairy industry

The process system at the heart of this thesis is a pilot-scale, thin-film evaporator with the rotating heat transfer surface resident in the Institute of Technology and Engineering, Massey University, called the spinning cone evaporator. This plant is a scaled version of the type of process that will be used in the dairy farms and was designed to research the on-farm evaporation system.

The spinning cone evaporator is a compact, on-farm method for pre-concentrating milk. There have been very few studies on this new evaporator, but to better understand this evaporator, a comprehensive study of the model (including a simulation) is needed. This paper presents an application of process modelling to simulate dynamic behaviour of a concentration process in a spinning cone evaporator.

1.2 Objectives of this thesis

The production of milk powders is a major part of the New Zealand dairy industry. The spinning cone evaporator is a newly developed, compact, on-farm method for pre-concentrating milk. It will be used as an important part of an on-farm evaporation system to remove most of the water in the raw milk (Chen, 1997). However, there is a shortage of knowledge on the design, operation and control of this new evaporator. To better utilize this evaporator and make further developments, a comprehensive study of the model and simulation of this evaporator is well motivated. The work, discussed, in this thesis was initiated with the aim of improving this situation.

The objectives of this work can be split into three parts.

1.2.1 Develop Model

A first principle model of a pilot scale spinning cone evaporator will be developed from the Laws of Thermodynamics and the general mass and energy balances. The model is dynamic and includes the evaporator, compressor, condenser and product transportation sections. Considerable attention will be focused on the derivation of the model.

1.2.2 Implement Model

The evaporator model was implemented using the software package MATLAB, along with its dynamic simulation environment SIMULINK. The differential equations for the evaporator model are embedded in a block diagram representation of the evaporator system.

1.2.3 Validate Model

The model validation is to determine whether the model is an adequate representation of the physical process. Some of the model parameters are likely to need adjusting from the theoretical or experimental values in order to improve the model.

A pilot-scale spinning cone evaporator was used for the experimental approach. The research work carried out in this project was intended to satisfy the requirements for the development and control of the spinning cone evaporation system.

1.3 Overview of thesis

The six subsequent chapters in this thesis are:

Chapter 2 reviews the liquid film flow and heat transfer enhancement by surface rotation, spinning cone evaporator, modelling and simulation techniques. It focuses mainly on the evaporation system and application dynamic models, modelling and simulating methodologies applied in related fields.

Chapter 3 presents a detailed explanation of the analytical model that has been formulated for the proposed spinning cone evaporator. The sub-system modelling techniques of the evaporator are used to create the dynamic model that includes the evaporation sub-system, compressor sub-system, condenser sub-system and product sub-system. Each sub-system is analysed and their equations are derived from the first law of thermodynamics and the general mass and energy balances. The main attention is focused on the factors affecting heat transfer and discharge flow in the spinning cone evaporator. The system model has been developed to describe the dynamic relationships between the input variables and the output variables, and to provide a method of better understanding the principle and operation of the spinning cone evaporator.

Chapter 4 explains the model implementation that is to implement a simulation of the system model described in Chapter 3 and to verify the model comparing its simulated output to the spinning cone evaporator, with data calculated from the mathematical equations.

Chapter 5 presents the experimental system and experiment methods for this project. The objective of the experiments to be undertaken is to collect data for validating the model of the spinning cone evaporator.

Chapter 6 explains the model validation for both steady state and dynamic comparisons. It gives the predicted results of the system model and experimental results of actual system. A comprehensive discussion of above results is presented.

Chapter 7 gives the conclusions of this research work and recommendations for any future work.

1.4 General background

Milk is one of the most precious natural materials and has been a basic component of human food for a long time. It is one of the oldest foods and at the same time the most important one. However, in many cases the milk cannot be consumed in its raw form. It needs to be prepared in order to be consumed and digested by humans. During treatment, the chemical and physical modifications can reach such levels that the raw material is distinctly different to the finished product. The non-desirable parts are to be removed or an enrichment or reduction of the nutritional content takes place.

Raw milk formation takes place in the milk gland of the cow. Milk production (called milking) is the discharge of milk from the udder either manually or mechanically.

Milking must be done in such a way that the udder remains healthy, and milk is not damaged. Then the milk is thermized or/and chilled, and can be stored for several days on the farm.

The raw milk is collected on the farm, and then is transported to the dairy factory to process the milk, including heat treatment as a precondition for milk processing. It is a basic process in each dairy plant. The type and intensity of the treatment are selected accordingly.

Processing milk into a wide variety of products requires a highly developed technology (Spreer, 1998). Evaporation is, and no doubt will remain, a major technique used for the removal of water in the dairy industry, although there are other concentration techniques (membrane technology and freeze concentration etc.) being developed. In New Zealand, evaporation is the only process used for the concentration of milk (Chen, 1997).

1.4.1 Evaporation technology

Evaporation is a special case requiring heat transfer to boil liquid. This particular heat transfer application is so common and important that it is treated as a separate unit operation. Historically, the first form of evaporation was the use of a direct fire pot or pan. Later the development of the steam heated pots; the enclosed vacuum pans and multiple effect evaporation expanded the use of this operation and added many economies to the process. Today, evaporation is one of the basic unit operations in use throughout industry (Bhatia, 1983). It is, and will continue to be an important technique used for the removal of water and sometimes other liquids in industry.

There are a variety of evaporation methods and a wide range of industrial applications, however the predominant applications of evaporators are the food industry. Evaporation is extensively used in the food industry for the following reasons (Russell, 1997 and Chen, 1997):

1. Pre-concentration of liquor to further processing, for example before spray drying, freeze and crystallization, etc.
2. To reduce transporting or storage volumes of liquid foods, hence prior to reduce packaging, transportation and distribution costs.
3. To reduce 'water activity' in certain foods by increasing the concentration of soluble solids in order to aid preservation.

4. For the utilization and reduction of effluent.

The number of the food processes that use the evaporators are extensive and include the milk powder, fruit juice, wine, salt and sugar processing. In New Zealand, evaporators are common within the dairy industry. These are used to pre-concentrate the milk before spray drying it to produce the milk powder.

Evaporation as a unit operation in industrial processes needs is done to separate materials (volatile liquid). Successful evaporation needs two things (Nisenfeld, 1985):

1. The necessary heat must be supplied to the liquid.
2. The vapour produced must not be allowed to accumulate above the surface of the liquid; it must be removed immediately.

1.4.1.1 Heat transfer principles in a single effect evaporator

An evaporation stage is referred to as an effect. The principles of evaporator heat transfer are most simply explained for a single effect. A simple single effect evaporator is illustrated in Figure 1.2. A flow of cold dilute solution and heating steam are fed to an effect. The effect brings these two flows into contact via a heating surface transferring heat from the hot steam to the cold liquid. If heat losses are ignored then all the energy given up by the steam is transferred to the liquid solution as sensible heat and latent heat of vaporization. The saturated vapour produced as the liquid boils is separated from the liquid phase and drawn out of the effect. Generally the concentrated liquid is the desired product of the evaporation process.

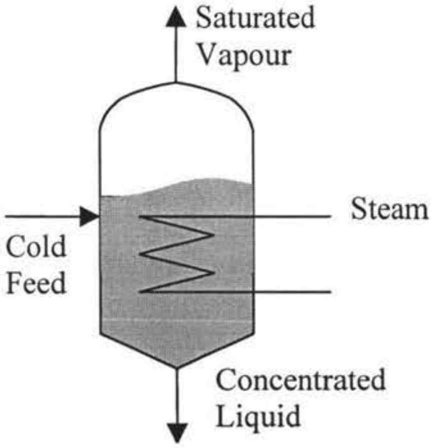


Figure 1.2: Schematic of a single effect evaporator

To analyse heat transfer in this single effect use is made of the basic heat transfer equation (Jebson, 1988).

$$q = U \cdot A \cdot \Delta T \qquad 1.1$$

Where: q is the overall rate of heat transfer, W

U is the overall heat transfer coefficient, W/m^2K

A is the total heat transfer area, m^2

ΔT is the temperature difference between the steam and the boiling liquid, K

A good understanding of this equation is important for the design, selection and operation of evaporators. Each of the four terms in this heat transfer equation 1.1 will now be considered separately.

q, the overall rate of heat transfer, is calculated from an energy balance on the evaporator as follows (heat loss to surroundings is ignored):

$$\begin{aligned} q &= VH_V + LH_L - FH_F \\ &= S\lambda_S \end{aligned} \qquad 1.2$$

Where: V is the mass flowrate of vapour streams, kg/s

L is the mass flowrate of Liquid streams, kg/s

F is the mass flowrate of feed streams, kg/s

S is the steam usage, kg/s

H_V is the vapour enthalpy, kJ/kg

H_L is the liquid enthalpy, kJ/kg

H_F is the feed enthalpy, kJ/kg

λ_S is the latent heat of evaporation/condensation of steam, kJ/kg

A, the total heat transfer area, is usually based on the area where the resistance to heat transfer is greatest. For an evaporator this is almost always the inside of the tubes available for the heat transfer, so A is the internal tube surface. It is calculated as follows:

$$A = N\pi Dl \qquad 1.3$$

Where: N is the number of tubes

D is the tube diameter (usually internal), m l is the tube length, m

ΔT, the temperature difference for heat transfer, is calculated simply as follows:

$$\Delta T = T_s - T_l \qquad 1.4$$

Where: T_s is the temperature of condensing steam, K

T_l is the temperature of boiling liquid, K

U, the overall heat transfer coefficient, is easily the most complex term in equation 1.1. In general, the overall heat transfer coefficient depends on the properties of the solution, the heating medium, and the surface geometry and type, including smoothness, cleanliness, composition, and thickness of metal.

Consider the plane wall in Figure 1.3 (Geankoplis, 1993) with a fluid at temperature T_s on the inside surface and a cold fluid at T_l on the outside surface. U is calculated as follows:

$$\frac{1}{U} = \frac{1}{h_i} + \frac{\delta_w}{k_w} + \frac{1}{h_o} \qquad 1.5$$

Where: h_i is the heat transfer coefficient inside tubes, W/m^2K

h_o is the heat transfer coefficient outside tubes, W/m^2K

k_w is the thermal conductivity of tube wall, W/mK

δ_w is the thickness of tube wall, m

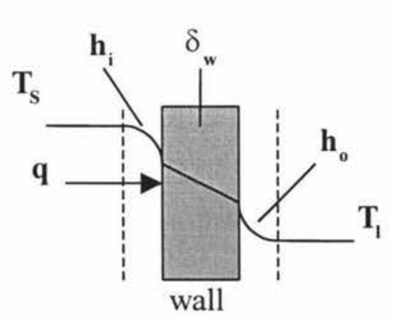


Figure 1.3: Heat transfer through a plane wall

Equation 1.5 is most simply regarded as the summing of three thermal resistances or three resistances to heat transfer from the condensing steam to the evaporating liquid inside the wall.

1.4.1.2 Type of evaporation

Ever since the beginning of the industrial use of evaporation a large number of evaporators of different design have been developed (Kessler, 1981). We will discuss here the most important types of evaporators that are of interest to the dairy industry. Basically an evaporation system involves a number of necessary elements which are together designed and constructed for a specific application. The essential components of a continuous industrial evaporation system consist of (Chen, 1997):

1. A heat exchanger to supply sensible heat and latent heat of evaporation to the liquid, this is usually via saturated steam.
2. A separator in which the vapour is separated from the concentrated liquid phase.
3. A condenser to condense the vapour and remove the condensate from the system.
4. A vacuum device to withdraw non-condensable gases and maintain a constant evaporating temperature.

These elements are usually of stainless steel construction that are easily cleaned and non-corroding.

There is not any single type of evaporator that could be satisfactory for all applications and different kinds of feed. In general, the different designs for the heat exchangers result in different types of evaporators. The main types of evaporators and their applications are briefly described below (Billet, 1989):

The natural circulation evaporator was the first type of evaporator to receive wide acceptability for the industrial concentration of liquids, known as the standard evaporator. The vertical tube bundle with a central downtake is located inside a steam chest enclosed by a cylindrical shell. The circulation of liquid (natural convection) past the heating surface is induced by boiling, which improves the heat transfer.

The forced circulation evaporator was developed from the standard evaporator. It has a vertical heated tube bundle located in an external heater separated from the downtake, which made it possible to pump the liquid upwards through the tube bundle. The fast liquid velocities improve the heat transfer and reduce the degree of fouling.

The agitated thin film evaporator employs a heating surface consisting of a large diameter tube in which the liquid product is spread over the inside wall by a series of rotating blades. These blades produce a thin film of liquid that is heated by steam surrounding the vessel. The action of the blades also suppresses the formation of fouling on the heating surfaces.

The falling film evaporator has vertical heating tubes and the liquid is carefully distributed to the top of the evaporation tubes. Then the liquid forms a thin film and flows down on the inner surfaces of the tubes. As the liquid moves down it boils off and the separation of the vapour and the concentrate takes place at the bottom of the tubes. This evaporator works successfully with a small temperature difference and has a low residence time.

The flash evaporator is a low-pressure chamber without heaters, in which a hot liquid is fed and the flash evaporation of the liquid occurs. This type of evaporator is the most suitable for solutions that are prone to salting or scaling and are very corrosive. It has been successfully used for producing potable water from seawater (Chen, 1997).

The centrifugal evaporator, also known as Centritherm evaporator (Figure 1.4), consists of a stack of hollow, rotating conical elements. The steam is fed into each cone and the liquid product is sprayed over the cones as they rotate. The centrifugal force spreads the liquid thinly over the heating surfaces to provide rapid heat transfer. The concentrate accumulates around the outer edge of the cones and is displaced upwards. Since a spinning cone evaporator is one kind of centrifugal evaporator, a more detailed outline of the uses, operation and characteristics of spinning cone evaporator is presented in following sections.

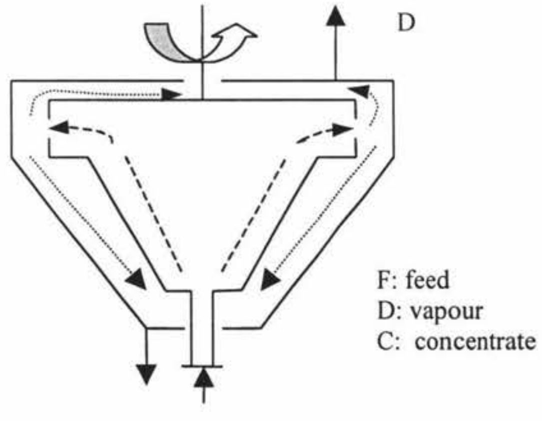


Figure 1.4: Centrifugal evaporator (Billet, 1989)

1.4.1.3 Vapour recompression

For an evaporation system, there are more efficient methods of using steam to heat the effects for reducing steam consumption, namely vapour recompression methods. In vapour recompression a part or the whole of the vapour from an evaporator is compressed with a resulting increase in temperature and is then used to heat the same evaporator again. In other words, the vapour recompression involves compressing the vapour generated in an effect to increase its temperature before returning it to the same effect as the heating steam. There are two methods of performing vapour recompression. Compression can be carried out either mechanically or thermally (Davidson et al., 1996).

Thermal vapour recompression (TVR). Here, the vapour is compressed by a steam-jet ejector to increase the temperature and pressure of the steam supply. This method can only compress part of the vapour from the effect so the remainder of the vapour is passed on to the next effects. The thermal compressor is relatively simple and inexpensive, without moving parts and therefore has a long useful life. A thermo compressor consists of a spray nozzle or propelling nozzle, a suction chamber with fittings for sucking in the vapour, a mixing chamber and a compression chamber.

Mechanical vapour recompression (MVR). The vapour is compressed by a mechanical compressor driven by electric motor, fuel engine or steam turbine. The exhaust vapour from an effect is fed to the compressor and then returned to the system. Only a small amount of make-up steam is required. Evaporators using mechanical recompression have the lowest operating costs but have the highest capital and maintenance costs. Mechanical compression can be effected by axial flow, by rotary piston or by single stage or multiple stage radial flow compressors.

Basic analysis of a fan compressor

In general, fan compressors are used at relatively low pressures, often at high flowrates. The fan operates on the centrifugal principle. Because of the change in density during compressible flow, the integral form of the Bernoulli equation is inadequate. The Bernoulli equation, however, can be written differentially and used to relate the shaft work to differential change in pressure head. In compressors the mechanical, kinetic, and potential energies do not change appreciably, and the velocity and static-head terms can be dropped. Also, the compressor is assumed to be frictionless. With these simplifications, the Bernoulli equation becomes (McCabe et al., 1993)

$$dP_{comp} = \frac{dP}{\rho} \qquad 1.6$$

Where: P_{comp} is work by ideal compressor, J/kg

P is pressure, kPa

ρ is density, kg/m^3

Integration between the suction pressure P_e and the discharge pressure P_{comp} gives the work of compression of an ideal frictionless gas:

$$P_{comp} = \int_{P_e}^{P_{comp}} \frac{dP}{\rho} \qquad 1.7$$

Where: P_{comp} is the discharge pressure

P_e is the suction pressure

Power required for the compression of gas. To use Eq.1.7, the integral must be evaluated, which requires information on the path followed by the fluid in the machine from suction to discharge. For the isentropic (adiabatic) compression of an ideal gas, the relation between P and ρ is given by Eq.1.8:

$$\frac{P}{\rho^{\gamma}} = \frac{P_e}{\rho_e^{\gamma}} \qquad 1.8$$

Where: $\gamma = C_p/C_v$

ρ_e is the inlet density

Substituting ρ from Eq.1.8 into Eq.1.7 and integrating gives (Eck, 1973):

$$P_{comp} = \frac{p_e \gamma}{(\gamma - 1)\rho_e}\left[\left(\frac{p_{comp}}{p_e}\right)^{\frac{\gamma-1}{\gamma}} - 1\right] \qquad 1.9$$

Temperature relation. When the pressure on a compressible fluid is increased adiabatically, the temperature of the fluid also increases. For the isentropic (adiabatic and frictionless) pressure change of an ideal gas, the temperature relation is,

$$\frac{T_{comp}}{T_e} = \left(\frac{p_{comp}}{p_e}\right)^{1-\frac{1}{\gamma}} \qquad 1.10$$

Where: T_{comp} is outlet absolute temperature

T_e is inlet absolute temperature

Substituting Eq.1.10 into Eq.1.9, this equation becomes

$$P_{comp} = \frac{p_e \gamma}{(\gamma - 1)\rho_e}\left[\left(\frac{T_{comp}}{T_e}\right) - 1\right] \qquad 1.11$$

Equation 1.11 shows the importance of the temperature ratio T_{comp}/T_e. It will be used to derive the outlet temperature of a compressor in the chapter 3 of this thesis.

1.4.1.4 Condensing and vacuum generator

The vapours produced in an evaporator are re-used where possible, either to heat other effects or for preheating the feed. Surplus vapours must be condensed and the heat removed from the system. Condensers operate at the end of the evaporation process to maintain the heat balance in the system, control the temperature of the effects and to provide a vacuum for evaporator operation. Most condensers are water-cooled surface condensers or mixing condensers (Russell, 1997).

Most evaporation systems in the food industry are operated under vacuum to lower the boiling temperature and so protect the food products from heat damage. This further improves the energy consumption of the systems.

The vacuum in the system is created by the condensing vapour in the condenser or/and the vacuum pump. Air may flow into the plant via leakage and non-condensable gases may enter. These gases may accumulate and interfere with the performance of the process if not removed.

Although the underlying principle of evaporation is simple, the detailed liquid flow patterns and the evaporation mechanisms on the heating surfaces are not very well understood. Complications in the evaporation process also arise as a result of the variety of products processed that have different properties. The behaviour of the liquid during evaporation must be considered in designing, selecting and operating evaporators. In other words, many parameters that depend on the characteristic properties of the evaporating liquid, govern the choice of equipment and the operating conditions (Chen, 1997).

1.4.2 On-farm evaporation system

On-farm evaporation systems have attracted worldwide interest for many years. When milk can be pre-concentrated at a farm the pre-concentrated milk, clearly, should be cheaper to transport to the processing plants than normal milk. Similarly, all other volume dependent handling and processing costs should be favourably affected by concentration. For example, pumps, milk storage capacities and refrigerated loads would be reduced proportionately, as well as cheese vat capacities or evaporator loads in milk powder plants. The dairy factory effluent would also be reduced.

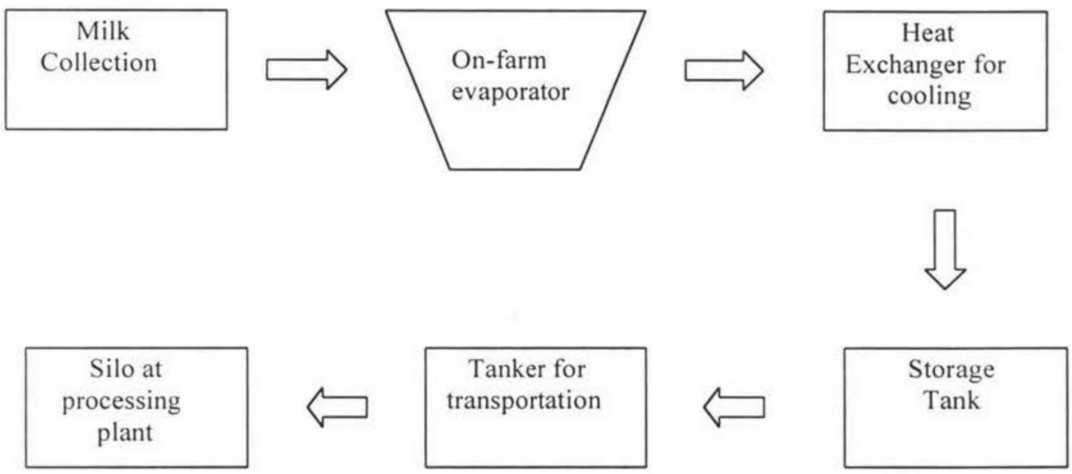


Figure 1.5: On-farm evaporation system

The on-farm evaporation system is shown in Figure 1.5. This system is operated as follows:

- The fresh whole milk is first collected in the buffer tank and then pumped into the on-farm evaporator.
- The concentrated milk from the evaporator is chilled in a plate heat exchanger and then passed to the storage tank. The storage temperature is about 5 °C and the maximum storage time is about three days.

- Finally the concentrated milk is transported by tanker to the dairy factory for further processing and may be stored in the dairy factory for a while (maximum two days) before processing (Chen, 1997).

The trials of the on-farm concentration have been undertaken for application of membrane technology and evaporation technology (Xu and Jebson, 1994). But membrane technology requires very pure water, which is not readily available on many farms (for cleaning). Therefore, evaporation technology is the only practicable alternative for on farm pre-concentration of milk.

The ideal evaporator to be used in the on-farm evaporation system should be highly efficient and compact, and cause minimal damage to milk. Research on both chemical and microbiological qualities of the pre-concentrated milk were completed (Xu, 1996). The results showed that pre-concentrated milk had an acceptable quality to be used in the dairy industry. A spinning cone evaporator, a highly efficient and compact evaporator, is being developed for the on-farm evaporation system in the Institute of Technology and Engineering at Massey University.

1.4.3 Spinning Cone Evaporator

In plain language, the spinning cone evaporator is a compact, cost-effective, newly developed, on-farm method for concentration of the sensitive materials, i.e. milk. A spinning cone evaporator provides an alternative of small to medium capacities in food, chemical, and pharmaceutical industries. The spinning cone evaporator utilizes the centrifugal force generated in a rotating system called 'spinning cone' to enhance the heat transfer performance and reduce the fouling on the evaporator heat transfer surfaces. This reduces the heat contact time in the evaporation process to a minimum, so that it is possible to evaporate quite sensitive materials at relatively high temperatures.

As shown in Figure 1.6 the spinning cone evaporator has an externally heated rotating rotor in the form of truncated cones. The general processes occurring in the spinning cone evaporator can be briefly described as follows:

- The steam for heating is fed into the steam jacket and condenses on the outer surfaces of the rotating cones.
- The preheated liquid to be concentrated is fed into the inner surfaces of the spinning cone. Under the centrifugal field, the liquid is immediately spread on the inner

surfaces in a film, which moves very fast in a radial direction, consequently the liquid film becomes extremely thin. Its heat transfer resistance is small, therefore, vapour is released rapidly.

- Vapour passes out via the vapour outlet to a condenser.
- The condensate from the steam is flung off by the action of the centrifugal force from the outer surfaces of the cone as soon as the condensate is formed, which results in a greater area being exposed for steam to be condensed. Further, the fast moving liquid film also reduces the formation of deposit on the heating surface because the formation of deposit on the heating surface is caused not only by the temperature but also by the movement of liquid on the heating surface. Therefore, the measured overall heat transfer coefficients on the spinning cone surfaces could be as high as 10 kW/m^2K(Chen et al., 1993) compared to 2-3 kW/m^2K for a falling film evaporator (Chen, 1992). Unvaporized liquid is removed using a pump and the centrifugal force of the rotating cone.

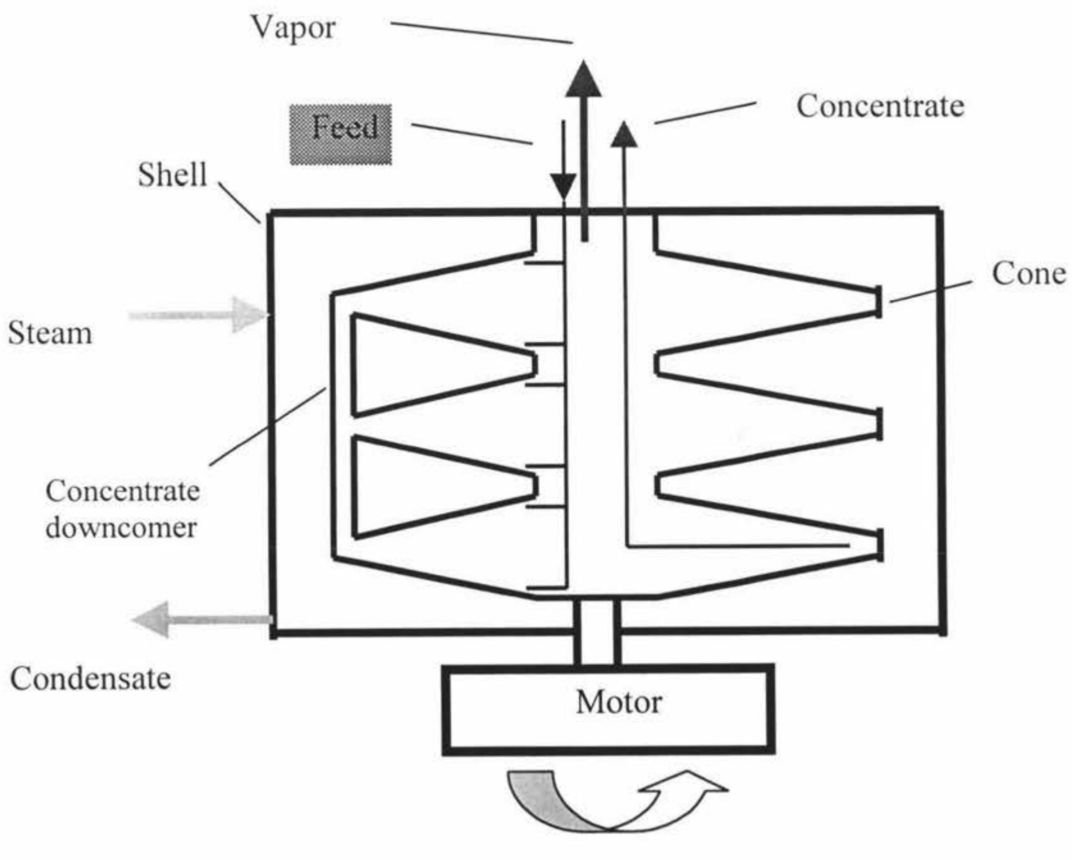


Figure 1.6: Structural diagram of a spinning cone evaporator

This type of evaporator is especially suitable for heat sensitive biological materials. As the liquid film passes over the heating surfaces rapidly, the liquid residence time in the actual evaporation zone may be only a fraction of a second (Billet, 1989).

This type of evaporator can also be used to concentrate viscous solutions, because the liquid can be distributed and moved along the heating surfaces with the assistance of the centrifugal force generated. The viscosity of the final product produced in this evaporation system could be up to 20 Pa.s (Chen, 1997).

A comparison between a falling film evaporator and a spinning cone evaporator has been taken by Billet (1989). Table 1.1 presents the data from this comparison. The spinning cone evaporator offers the following features (Chen, 1997) compared with the falling film evaporators that are used so popularly in the dairy industry or other evaporators:

Table 1.1: A comparison between falling film and spinning cone evaporators

Magnitude	Falling film	Spinning cone
Heat transfer coefficient (W/m2K)	730	2900
Temperature difference (°C)	40	10

1. The spinning cone evaporator has shorter residence times, which is important for heat sensitive products such as enzymes, antibiotics, herbal medicines, protein solutions, fermented liquids. When evaporating or concentrating these heat-sensitive liquids, it is required that the process liquid receives a specific quality of heat at a minimum temperature for an extremely short time period. The spinning cone evaporator satisfies all three of these important criteria.
2. It has a higher overall heat transfer coefficient. Rotation of the heating surface does not only improve the heat transfer on the liquid side by effectively distributing and quickly passing the liquid to be evaporated on the surface, but also benefits the steam condensation on the other side. As soon as the steam condenses on the outer surface of the cone, the condensate forms droplets, which are flung from the rotating surface by centrifugal force. Therefore, no condensate film, which would offer resistance to heat transfer, is formed. The dropwise condensation is considered to be

a major mechanism of condensation, which gives rise to very high heat transfer coefficients. Further, as presented above, the fast moving liquid film also reduces the formation of deposits on the heating surface because the formation of deposit on the heating surface is caused not only by the temperature but also by the movement of liquid on the heating surface.

3. It operates with high concentrated liquids. It was reported that the milk concentrate produced in the spinning cone evaporator (Evapo unit) can be up to 85% dry matter (Anon, 1990).
4. It is also highly suitable to concentrate high viscosity products. In a conventional falling film evaporator, the liquid film flow is assisted by vapour velocity, but at high liquid viscosity this is less effective (Jebson and Iyer, 1991). Hence, the final concentration of the products is limited. In the spinning cone evaporator, however, because the centrifugal force generated in the rotating system could be more than one hundred times greater than the force of gravity, the liquid transport can be enhanced using mechanical assistance, which means considerably higher velocity and hence larger heat transfer coefficients and the possibility of obtaining a higher final concentration.
5. It meets the requirement for cleaning production plants, since the amount of volatile decomposition products, which can enter the atmosphere or the effluent through the vacuum system, is reduced due to extremely short residence times.

Thus spinning cone evaporators will be accepted rapidly and allow a fair amount of scope in providing small to medium capacities in the chemical, pharmaceutical and the dairy industries. They permit applications in which conventional evaporators would fail entirely or, at the most, involve a compromise leading to reduced yield and quality. From the aspect of modern process engineering, they represent an irresistible development and can be considered more as welcome companions than as competitors to the other types with short residence times. In view of the increasing severity of legislation on the environment, they also meet the requirements for 'clean' production plants. This may be a convincing argument in swaying decisions on equipment selection in favour of these evaporators (Billet, 1989).

A schematic diagram for a spinning cone evaporation system is shown in Figure 1.7. It consists of a spinning cone evaporator, a fan compressor and a surface condenser, the feed section and the product transportation section.

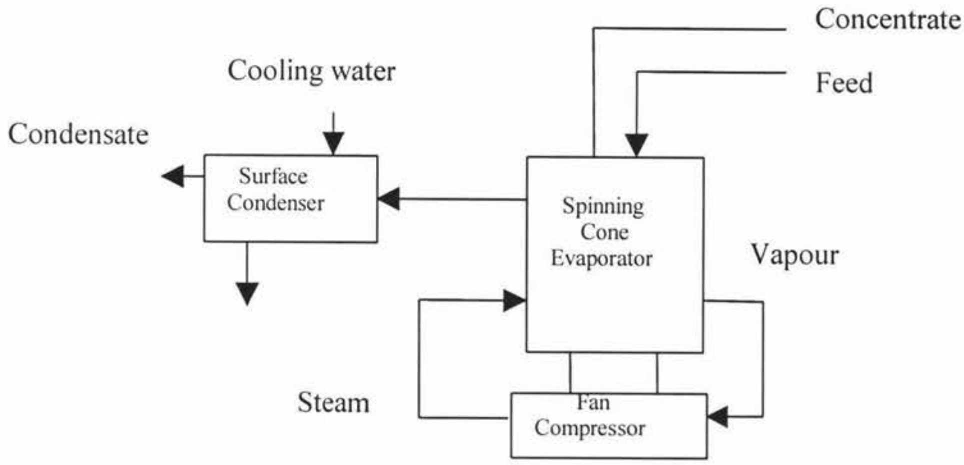


Figure 1.7: Schematic diagram for a spinning cone evaporator system

Figure 1.8 shows the flow chart of the spinning cone evaporator system at Institute of Engineering and Technology of Massey University. The general processes occurring in the evaporator can be briefly described as follows. The liquid to be concentrated is fed into the inner surface of the rotating cone. Under the centrifugal field, the liquid is immediately spread on the inner surface as a film, which moves very fast in a radial direction. Since the film is very thin, its heat transfer resistance is small, consequently vapour is released very rapidly. Vapour passes out via the vapour outlet to a condenser, where the vapour is condensed.

The steam for heating is fed into the steam jacket and condenses on the outer surface of the rotating cone. The condensate from the steam is impelled off by the action of the centrifugal force from the outer surface of the rotating cone as soon as the condensate is formed, which results in more areas being exposed for steam to be condensed.

Vacuum is provided by a vacuum unit. When the water is pumped by the circulation pump through the ejectors, a vacuum is created. The vacuum is adjusted manually during operation, and released at the end of the operation.

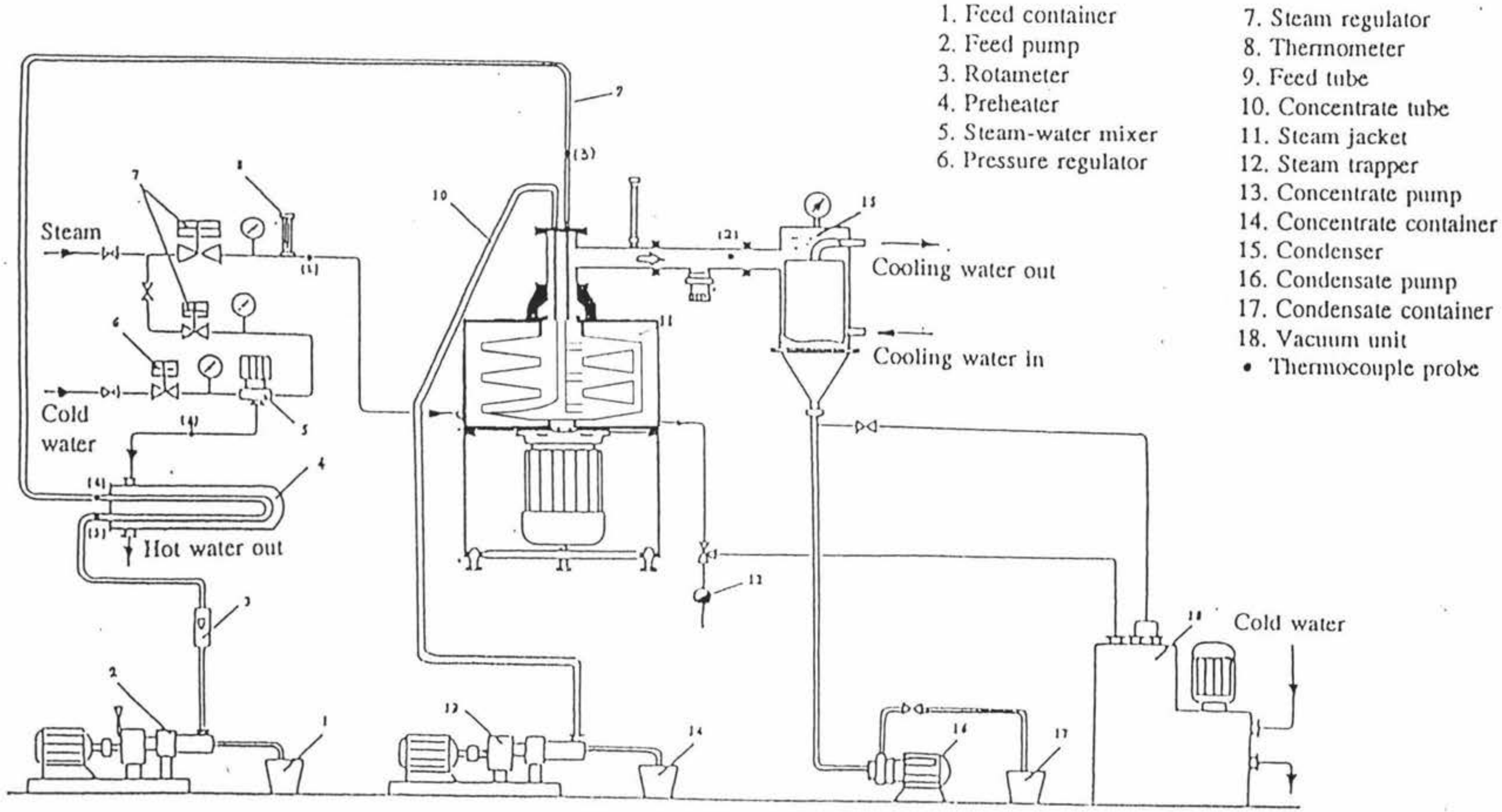


Figure 1.8: Flow chart of the spinning cone evaporator system

(Adapted from Hong, 1997)

1.4.4 Analytical, dynamic modelling

Process modelling and computer simulation have proved to be extremely successful engineering tools for the design and optimisation of physical, chemical, and biological processes. The use of simulation has expanded rapidly during the past three decades because of the availability of high-speed computers and computer workstations (Ramirez, 1997). In developing this new evaporator and its control system, a process simulation should be used to test and refine the performance of the evaporation system. This is an important step in remedying any problems before the system is implemented on the real system. Additionally, a process simulation allows the settings of different variables to be optimised.

A good mathematical model should be general (apply to a wide variety of situations), realistic (based on correct assumptions), precise (its estimates should be finite numbers), accurate (its estimates should be correct) and there should be no trend in the deviations of the model from the experimental data. A good model should be robust and fruitful.

The first step in building a mathematical model is the definition of the system. Factors affecting the system should be identified, and the data should show the effects of the individual factors. The system may be simplified by neglecting the effects of the marginal inputs. A single equation or a set of mathematical equations will usually be derived after using many fundamental principles and techniques. Comparison of the mathematical model, i.e., solution of the equations, with the experimental data is the final stage of modelling. The model is validated if it agrees with the data. If such an agreement should not be obtained, all the steps of modelling, starting with the definition of the system, are repeated until a satisfactory representation is obtained (Fig. 1.9).

The first task in the process would be the development of an analytically-derived dynamic model of the system. The ultimate application of the dynamic model largely determines the method of analysis of the system and the complexity of the final model. Models for the purpose of design or analysis of the process mechanisms often require distributed models containing partial derivative information. On the other hand models for design of the evaporator and its control system do not require as much complexity and lumped-parameter analyses usually suffice.

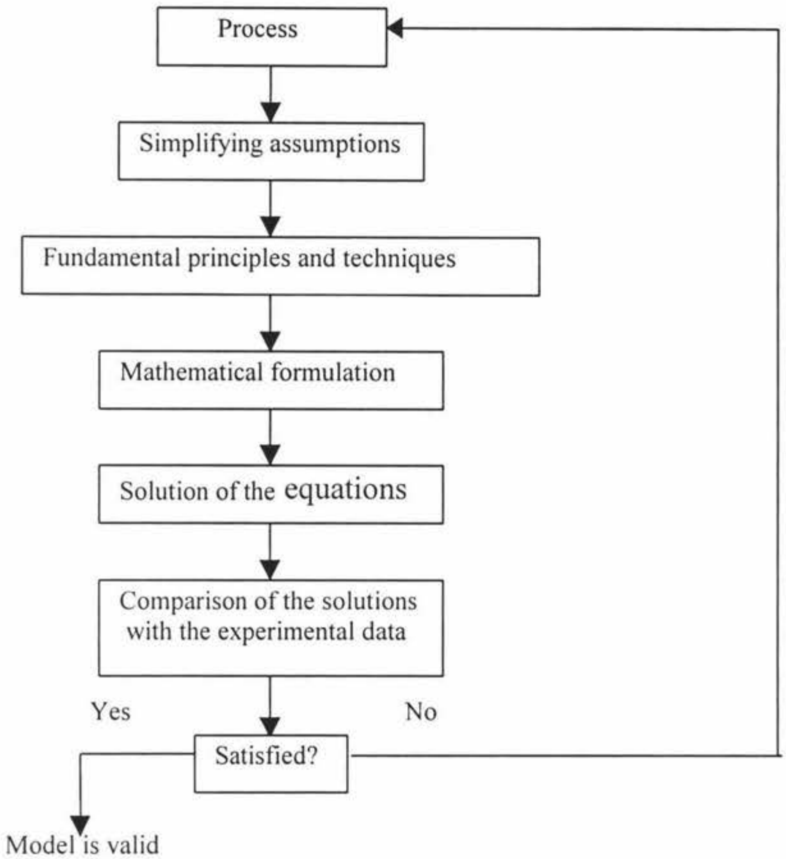


Figure 1.9: Schematic description of modelling (Ozilgen, 1998)

The mathematical simulation of complex operations is one type of tool which can be used to improve resolutely the type of process performance of the existing complex operation, to increase with change in construction size and the quantity of products, to decrease the specific material requirement and to perfect the energy balance. It has become exact and effective in last decade, on the one hand by utilising new scientific results of process engineering related to the internal mechanism of the processes and on the other hand by using computers (Benedek, 1980).

Simulation and modelling generally use technical devices. The modelling consists of creating a "quasi-object" suitable for studying the behaviour of a system, from which the output of the system to a given input may be established. This type of research is termed simulation. Simulation is used to obtain by simple means and with low cost that information which could otherwise only be available by measurement on the original system.

Other reasons to use a process simulation include:

1. Process optimisation. Through investigation of the process simulation one could find the optimal settings that should be applied on the actual plant.
2. Testing operating scenarios. Various operating scenarios could be run without danger of disrupting the actual process. This could benefit the process engineer who wishes to determine the effects of applying different operating strategies, using different raw materials or making physical alterations to the plant.
3. Operator training. Simulation models could provide off-line training to operators and an understanding of how the characteristics and responses of the process can be gained.
4. Process monitoring tool. Generally a model represents a perfect system. Any deviations the plant makes from the model may indicate areas where the process needs attention, for example it may indicate when the vacuum has been changed.

Chapter 2 Literature Review

The literature relevant to the present work may be divided into four principal areas: liquid film flow and heat transfer enhancement by surface rotation, spinning cone evaporator, modelling and simulation techniques. This review will mainly focus on the evaporation system and application dynamic models, modelling and simulating methodologies applied in related fields. Although there have been some studies on the dynamic nature of simple forced circulation evaporators and falling film evaporators (Winchester and Marsh, 1999), there have been very few on the spinning cone evaporator (Zhu and Bakker, 2000).

The dynamic modelling and simulation are the first step to the study and operation of evaporation system. In some cases these can be done by deriving the appropriate equations from first principles and calculating the parameters from physical property data and/or design correlation. In other cases the parameters must be determined by using actual plant data. While individual parts have been modelled for some time, the complexity of an industrial plant consisting of many components was beyond the resources of researchers until the early 1970s. Since then, the increasing availability of digital computers has allowed researchers to develop dynamic mathematical models which are sophisticated enough to model and simulate any evaporation system (Lovatt, 1992).

2.1 Liquid film flow and heat transfer enhancement by the rotating surface

2.1.1 Basic concepts

In many chemical engineering applications, a liquid undergoes rotational motion, for example, in a centrifugal pump, a stirred vessel, a basket of a centrifuge or a cyclone-type separator. The general case of the flow on a rotating surface is hardly elaborated because of the enormous mathematical difficulties associated with the problem. Using the Navier-Stokes equations (Mills, 1995), Schlichting (1979) developed the boundary-layer model for the conservation of mass, momentum and energy in the liquid film on the rotating cone.

The rotation of the heat surfaces has the following advantages for heat and mass transfer (Jachuck et al., 1994):

- The equipment volume could be reduced and the exchanger design and operation may be less constrained because of the high heat transfer coefficient.
- The liquid film process is enhanced due to the rotation.
- There is a self-cleaning action whereby rotation can better handle liquids that contain solids as well as viscous products.
- The heat sensitive liquids can be processed due to less fluid residence time in the heating surfaces.

Orozco and Francisco (1992) developed a boundary layer model of laminar, subcooled, free convection film boiling from a rotating sphere. The conservation equations for the vapour and liquid were simplified, transformed into ordinary differential equations using an integral approach, and solved numerically. The theoretical variation of vapour film thickness with heater temperature and the resulting boiling fluxes were investigated. Moreover, an experimental facility was built for the purpose of verifying the validity of the theoretical model and good agreement was found between the model and the experimental data at low rpm. The instability of the vapour film near the minimum heat flux for a rotating surface flux was also investigated.

2.1.2 Film evaporation

Film condensation takes place on a surface when it is cooled to a temperature below the saturation temperature, corresponding to the pressure of the vapour. On the other hand, if a film of liquid is on a surface that is heated to a temperature above the saturation temperature, evaporation from the surface of the film will occur.

An analytic method for calculating the vaporisation rate of a falling film heated by a gas or saturated steam was described by Gutorova et al. (1984). The method is capable of correcting, with accuracy acceptable in engineering calculations, for the various factors affecting this process.

Conder et al. (1982) presented a mathematical model for the vaporisation of liquid from a laminar film flowing down the inside surface of a smooth tube into a countercurrent laminar flow of gas. The partial differential equations describing temperature and composition distributions are integrated across the tube to give a set of four coupled ordinary differential equations.

The thin film flow, as the result of the application of a forced centrifugal field, can be found in the thin film evaporator with rotating surface and rotating disk (Blass, 1979). Extensive research work has been studied on the film flow on plates and tubes both theoretically and experimentally.

2.1.3 Heat transfer through a thin film

Heat transfer problems almost always involve more than one mode of heat transfer occurring simultaneously. The proportions of each mode are mainly dependent on the heating rate and the liquid flow pattern. The thermophysical properties of the liquid, namely thermal conductivity, viscosity, density, specific heat, latent heat, surface tension and conditions of the heating surface all may also have an effect on the heat transfer process (Bett et al., 1975).

Heat transfer, characterised by the dimensionless Nusselt number Nu, depends on the flow characteristics of the film (Reynolds number Re) and on the properties of the solution (Prandtl number Pr). Evaporation without bubble boiling shows no evidence of dependence on the driving temperature difference. Heat transfer was measured in a pilot-scale falling-film evaporator by Lehnberger et al. (1995), in relation to a wide range of parameters: vapour pressure, dry substance content, juice circulation rate and temperature difference. Both laminar and turbulent films (film Reynolds number 14-2000) were studied. The temperature difference was varied between 1 and 10 K. The measured overall heat transfer coefficient did not depend exclusively on the dynamic viscosity of the juice but also on such parameters as recirculation rate and vapour pressure. Nevertheless, correlation of all experimental data approximately confirmed the GEA-Wiegand equation for dimensioning of falling-film evaporators. The experimental results were compared with published data on heat transfer to laminar falling flows, and equations for predicting the heat transfer in falling-film evaporators were evaluated.

Kessler (1981) stated out that the convection heat transfer in rotating system can be enhanced and this method is characterized by high heat transfer coefficient.

To gain some understanding of the mechanism of liquid film flow and evaporation on the inner surface of the rotating cone, Chen and Jebson (1997) developed a theoretical method of calculating the local liquid film thickness and heat transfer coefficients on a rotating cone using a Nusselt-type assumption. They found reasonably good agreement between the calculated heat transfer coefficients and the experimental results obtained using two evaporators with the heating rotating surface containing different cone angles.

Chen (1997) investigated the overall heat transfer coefficients of the spinning cone evaporator. He showed simple theoretical models for analysing heat transfer and fouling in film evaporators with rotating surfaces (present names spinning cone evaporators). The on-farm evaporation system and experimental method on heat transfer were introduced. The results showed that the liquid film thickness and heat transfer coefficient are affected by the feed flow rate, rotating speed, length, angle of the cone, and liquid viscosity. The film thickness decreases with decrease in feed flow rate and liquid viscosity, increase of cone length and speed. Increase of cone angle results in thinner liquid film and a higher heat transfer coefficient.

A method for calculating the heat transfer coefficient in an evaporating liquid film was introduced by Taubman and Kalishevich (1976). The method makes allowance for the superheating of the liquid by friction between the vapour and film surface.

Schroder et al. (1979) introduced the direct measurement of overall heat transfer coefficients in a multi-effect falling film evaporator pilot plant. The results indicated that a considerable amount of heat is transferred by convection.

It is difficult to accurately determine the temperature in an evaporator during the evaporation process. However, Hoffman et al. (1981) reported a technique that can be used to measure the temperature in the evaporator during the evaporation process.

2.1.4 Pressure gradient and discharge flow rate in a centrifugal field

The centrifugal pressure gradient in the rotating bowl was analysed by Coulson (1999). The results showed the centrifugal pressure gradient increases with the radius of rotation and the functioning of the gravitational force can be ignored.

Studies on the discharge flow rate through the nozzle in a centrifugal field were carried out by Hsu (1981) for a nozzle in tubular centrifuge and in a cone-type disk centrifuge. The result was summarised as follows: the flow rate may be the function of the rotational speed of the cone, the cross-sectional area of the nozzle and the radius of the cone.

2.2 Spinning cone evaporator

Milk, which has been pre-concentrated on farm, clearly should be cheaper to transport to the processing plant than normal milk. Similarly, all the other volume dependent processing and handling costs should be greatly affected by concentration. Robertson (1987) pointed out that on-farm concentration of milk using either membrane processing or mini-evaporators might well be a concept that will be of rapidly increasing importance in the decade ahead.

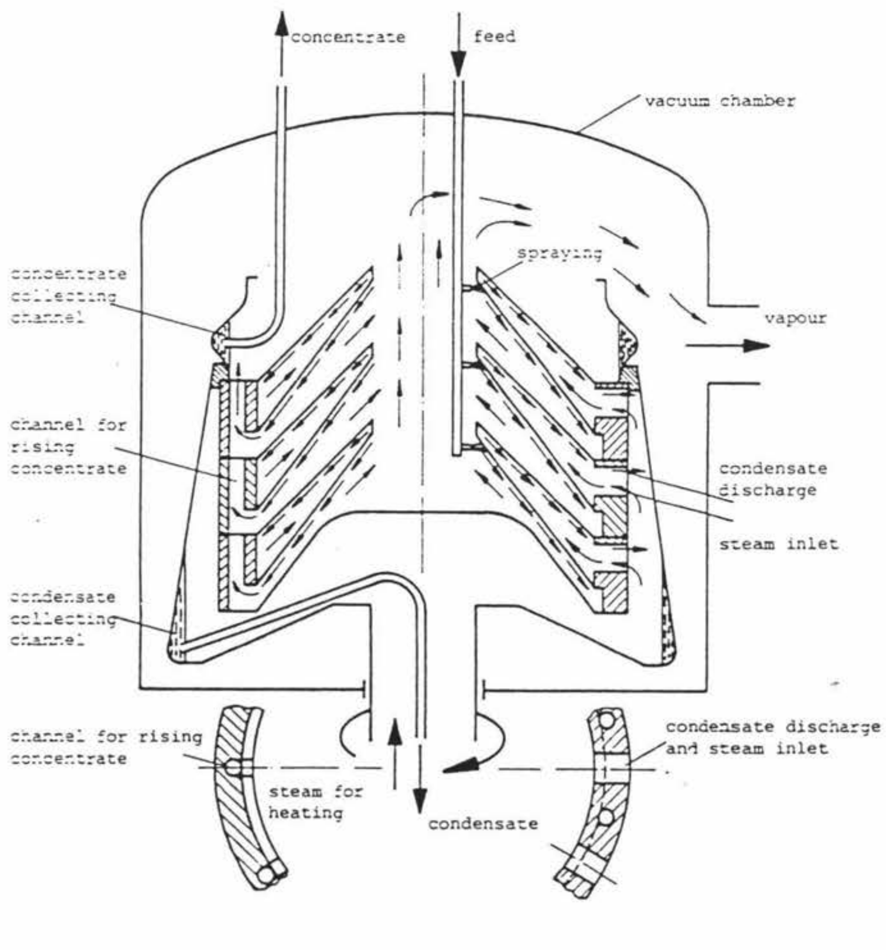


Figure 2.1: Diagrammatic representation of film evaporators with mechanically moved parts (Kessler, 1981)

However, it is known that industrial scale plants are more effective and the smaller the plant is, the lower effectiveness. When an evaporator is used on farm, it has to be a small or medium scale one. How will it compete with the industrial scale plant? Dodeja et al. (1981) presented the constructional details of an agitated thin-film evaporator along with the heat and mass transfer data, and discussed the specific applications of the evaporator in the dairy industry. The centrifugal force was used to enhance the heat transfer performance in evaporator. Therefore its cost effectiveness is very high.

Kessler (1981) introduced that a film evaporator with mechanically moving parts (Figure 2.1) was developed to evaporate highly viscous products. This method was characterised by high heat transfer coefficients (up to 8000 W/m^{2}K), extremely thin films (approx. 0.1 mm) and extremely short product contact times. A disadvantage of rotating equipment is the requirement of high cost machining.

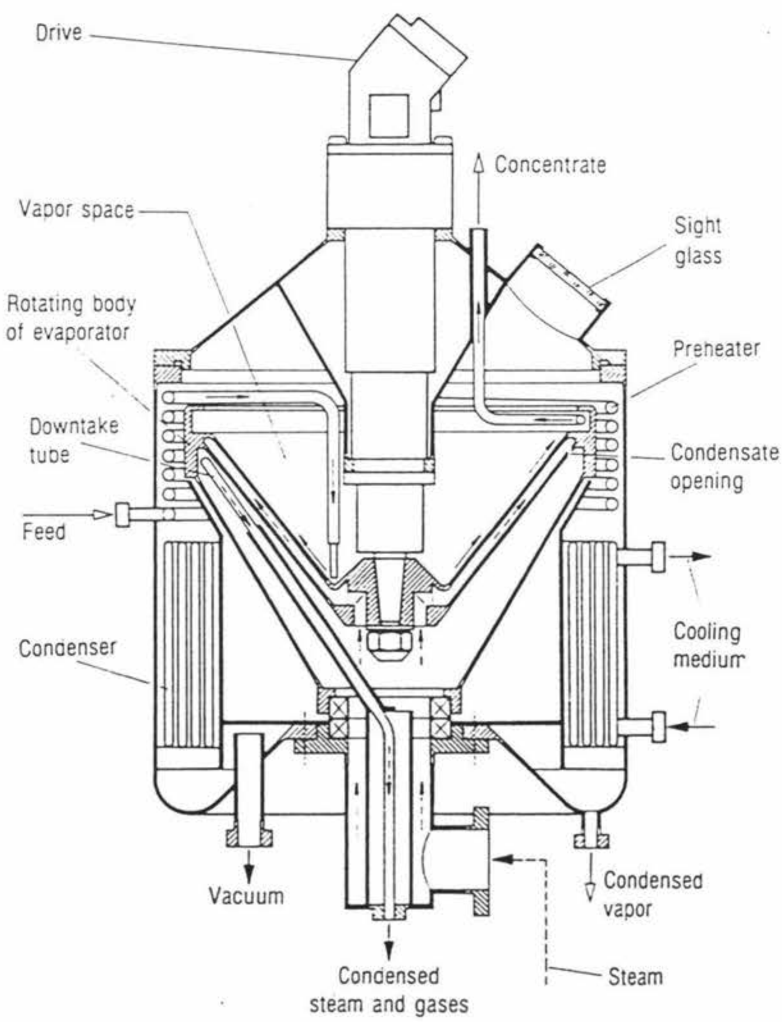


Figure 2.2: A compact evaporation system (Billet, 1989)

A compact evaporation system (Figure 2.2) was discussed by Billet (1989). In units of small to medium capacity, the condensers can be designed as an integral part of the evaporator with the rotating conical heater. There are a number of processes engineering advantages of this compact system. For instance, it can save space, and the path followed by the vapour to the condenser becomes much shorter, with the result that the evaporators can be operated at lower pressures, and that there is less expenditure on insulation and the heat losses are lower. However, the limited area of the rotating heater implies that these evaporators cannot be considered for large capacities.

Wersel (1993) introduced the use of Alfa-Laval plate evaporators. Trials were performed in 1987 by Alfa Laval and tested in 1989/90. Results showed that residence times are only 10-20 s, compared with 200 s for a falling-film evaporator and 600 s for a Robert evaporator; hence, much less inversion and colour formation occurs for a given ratio of surface area to juice through-flow.

Anon (1992) reported that an on-farm concentrator could halve the water content in milk. The experiments were carried out with a small-scale mechanical vapour recompression evaporator. It concentrated the milk at temperatures between 61 °C and 65 °C, at a rate of 200 litres of whole milk per hour. The researchers believed the unit has potential to increase milk solids level to 36%.

Xu (1996) also used a Centritherm evaporator to find out the most suitable evaporating temperature and the optimum concentration of milk in an on-farm evaporation system. The effect of evaporation conditions on the changes in both chemical and microbiological qualities of pre-concentrated milk during the storage on farm, transportation from farm to the dairy factory, and the storage at the dairy factory were investigated. It was found that pre-concentrated milk had an acceptable quality to be used in the dairy factory. From a microbiological point of view, it was found that the number of total bacteria were much less in the pre-concentrated milk than in the raw milk, therefore, the pre-concentrated milk could be stored safety for the proposed period of time at the farm and the dairy factory.

To obtain some understanding of the mechanism of evaporation for high viscosity solutions on a rotating conical surface, Chen and Jebson (1997) used a Centritherm evaporator to concentrate the sugar solution from 30% total solids to 68%. The overall heat transfer coefficients were affected mainly by the speed of the rotation of the heating cone, the temperature difference between the steam and the evaporating liquid, and the viscosity of solutions. A theoretical model of overall heat transfer coefficients as the function of solution viscosity on the rotating cone was developed and the values were reasonably close to the measured overall heat transfer coefficients. Therefore, a spinning cone evaporator was developed at the Massey University, which is a compact, high effective and on-farm method for pre-concentrating milk. It has been used as an important part of an on-farm evaporation system to remove most of the water in the raw milk.

2.3 Evaporator modelling techniques

Because the spinning cone evaporator under development for the dairy industry and operation on farms there is no literature on its modelling. There has been some work done modelling other evaporators.

2.3.1 Basic evaporator model

Evaporation and spray drying in the New Zealand dairy industry were reviewed by Jebson (1988). He listed a large number of different factors that may influence the overall heat transfer coefficient. These include:

Design factors:

- Viscosity
- Vapour momentum (velocity)
- Temperature difference

Operating factors:

- Liquid distribution
- Fouling --- inside & outside tubes
- Calandria angle
- Gas blocking
- Restricted condensate flow

An optimisation of evaporating systems was introduced which manipulated plant-operating parameters to achieve maximum throughput.

In an evaporator, heat is transferred from a heating medium (usually steam) to a solution, usually by conduction through a solid surface (tube wall). As the solution boils, mass and heat are simultaneously transferred into the vapour phase. Consequently, an evaporator model consists of (Price, 1998):

- an overall material balance
- component material balances
- energy balances
- heat transfer equations

2.3.2 Early developed evaporator model

Newell and Fisher (1972) presented a generalised approach to the modelling of multi-effect evaporators, which separates the development of dynamic equations from the specification of evaporator configuration. The result is a modular approach that is effective and convenient to use. A high-order non-linear, dynamic model of a double-effect pilot plant evaporator was derived using this approach and then simplified and linearized to produce a state space model which gave generally good comparisons with experimental open-loop responses.

Rastogi (1974) derived the non-linear dynamic mathematical model from the physical processes existing in a single effect of the evaporator system. The model also included a mechanism to quantify changes in concentration and temperature of black liquor. For small deviations from the operating point, the non-linear model was simplified by additional assumptions, and was then linearized. The overall model for a multiple effect evaporator system was set up by expanding the linear or the non-linear model to cover all the effects in the system. The input and output variables between the effects were then related in accordance with the plant flow diagram. Results from simulation experiments showed that the models exhibit all important response characteristics of the actual evaporator. A domain of heat transfer coefficient parameters was found in which the non-linear model exhibited an oscillatory strong black liquor concentration response similar to that observed in actual multiple effect evaporator systems.

A mathematical model for computer design and analysis of a five-effect evaporator system, commonly used in the sugar industry, is given by Radovic et al. (1977). The model consists of four equations per effect: (1) the enthalpy balance, (2) the heat transfer rate, (3) the phase equilibrium relationship and (4) the mass balance equation. Two modes of operation were employed. The first one calculates the steam consumption, the heat transfer surface, and the distribution of temperature, composition, and mass flow rates, to give a desired exit composition of the solution. The second one can be used to calculate all the necessary process parameters of an existing industrial evaporator system. The model was tested in solving some real problems frequently encountered in the evaporation of the sugar solution.

A mathematical model for simulating transient behaviour of a concentration process in a rotary steam-coil vacuum evaporator was developed by Lima Hon et al. (1979). A heat transfer correlation was determined. The research work showed good agreement between the simulated and experimental results validating the model and methodology. The model can be used to simulate the performance of the evaporator under a wide range of conditions.

Adamopoulos et al. (1980) developed a reduced model of a climbing film evaporator. Using a technique that allows for the distributed effect by empirically defining a parameter in a reduced lumped parameter model that is dependent on operating variables, a satisfactory approximation to observed behaviour could be obtained. This reduced model was used to predict important non-measured variables.

Nielsen et al. (1989) described a real-time simulator for control experiments. The simulator represents the evaporator section of a sugar plant. A model has been developed describing the dynamic relations between inputs, outputs and disturbances. In the model a lumped parameter approach has been used.

2.3.3 Latest development of evaporator model

The mass/energy model of a double-train five-effect evaporator has been used by Mulholland and Love (1994). The system was linearized by integrating the mass model in parallel, forcing it to track measured exit compositions. The modelling required 33 equations and 59 variables, of which 26 could be identified. The interconnection of vapour spaces, for each pair of vessels in the two trains, gave a multiplicity of energy paths through the system. In off-line tests, using long records of plant data, the system showed promise as an observer for concentration control, and for monitoring changes in the heat-transfer coefficients, for example due to scaling.

A dynamic analytical model for the straight through configuration of a three effect falling film evaporator was presented by Rangaraju and Bakker (1994). The approach was to divide the effect into four dynamic sub-systems: distribution plate, evaporation tube, product transport and energy. The model was used in controller design simulations.

Based on same sub-systems as above, a model of a falling film evaporator with mechanical vapour recompression was developed and analysed by Winchester and Marsh (1999) for the purpose of examining the coupling and functional controllability of the three core control loops for effect temperature, product dry mass fraction and product flowrate. A first principle model of appropriate complexity that captured the essential mechanisms was developed and used. The model described the dynamic relationships between the manipulated (condenser coolant flowrate, compressor speed, feed flowrate) and disturbance variables (feed temperature and feed dry mass fraction) and the process variables (evaporating temperature, product dry mass fraction and product flowrate). A single pass evaporator was studied, the applicability of the results to multi-pass system was considered.

Young and Allen (1995) investigated the modelling and control of a 25 kW, 13.6 litres/hour pilot-plant climbing film evaporator concentrating a sodium nitrate solution. The simplest models of the evaporator were global linear lumped-parameter models. Gain-scheduled linear lumped parameter models were identified to compensate for system nonlinearity. Finally, full distributed-parameter models were derived for the evaporator, and their parameters were identified.

Peacock and Starzak (1996) presented a mathematical model of the climbing-film evaporator system used in the South African sugar industry, which was derived and used to develop a simulation tool in Matlab programming language for evaporator designers and operators. The basis of the mathematical model was briefly outlined, and the results of preliminary simulations were reported. Testing of the model against experimental data from the pilot-plant climbing-film evaporator was discussed, and improvement of the model by process identification was described; the effect of scaling can be accounted for if the thermal properties of the scale are known.

Yeates and Morison (1996) made considerable advances in the understanding of falling film heat transfer and evaporation. Models were presented that incorporated wavy films, eddy diffusivity and vapour shear effects in the context of falling film evaporators. A dynamic model of an evaporator tube was developed incorporating two-phase flows, with evaporation, of a real solvent-solute system. They found that there might be considerable changes in the properties (viscosity in particular) over the length of a tube in real evaporator systems.

Russell and Bakker (1997) described the development of a neural network model of a pilot-scale, three-effect falling-film evaporator for use in a model predictive control system. The data used in its development are from a simulation model of the evaporator. The approach taken in the neural network modelling is to divide the full model into a group of sub-networks, each modelling a specific element of the overall system. The results showed that the model can perform satisfactory long-range predictions and hence were well suited for implementation within a model predictive control scheme.

Russell (1997) presented the development techniques of modelling and the model implementation. A dynamic model of a pilot-scale falling film evaporator was provided. The model was implemented using the software package Matlab along with its dynamic simulation environment Simulink. The model did perform well in estimating the dynamic nature of all the variables that were in agreement with what was observed on the plant.

Robertson and Cameron (1997) introduced a method for systematic model development for startup and shutdown simulation that is based on the identification of the essential process structure. Starting from a detailed mathematical process description, both singular and regular structural perturbations are detected. The technique generates detailed insight into the dynamic structure of the models, providing a basis for system re-design and dynamic analysis. The technique is illustrated by the modelling for an evaporator startup.

Telnes and Balchen (1997) presented an automated approach for establishment of dynamic state space models for unit processes. An example demonstrating the modelling of an evaporator is used.

Elhaq et al. (1999) discussed the problem of controlling a multi-stage industrial evaporator in a sugar factory. The control objectives are: syrup should have a sucrose concentration of 72%, the evaporator should produce the required amount of steam, and the steam consumption should be as low as possible. The control problem is solved by developing a physical model that was used as benchmark; then, a parametric identification was performed to obtain a control model; finally, a multivariable GPC controller was used to achieve these objectives.

A model for heat-and-mass transfers in the plant that consists of a falling-film evaporator and condenser is developed by Bourouni et al. (1999), resulting in a set of classical equations. Two experimental pilot studies were used. The influences of different thermal and hydrodynamic parameters on the unit performances were investigated. Experimental results were compared with the model predictions. From this comparison, it can be learnt that the model is well able to predict the trends of the heat-and-mass characteristics of the evaporator.

A performance analysis was carried out for vapour compression, parallel feed, multiple-effect evaporation water desalination system by El-Dessouky et al. (2000). The systems included the mechanical (MVC) and thermal (TVC) vapour compression. The analysis was performed as a function of the brine distribution configuration, the top brine temperature, the temperature of the brine blowdown, and the temperature difference of the compressed vapour condensate. The system models showed good agreement with field data. The balance equations of the condenser, the model of the mechanical vapour compressor and the calculations of the overall heat transfer coefficients are good examples for this project.

Zhu and Bakker (2000) introduced a model of a prototype on-farm evaporation system using a spinning cone evaporator. The analysis used a model structure that uses modules describing; evaporation, vapour recompression and condenser. The system model takes into account the dependence of the parameters that affect the evaporation temperature, product dry mass fraction and product flow rate, which includes the speed of the spinning cone, the speed of the compressor, the flow rate of feed, and the flow rate of cooling water. The feed dry mass fraction and feed temperature are considered as the disturbance variables. The model is implemented using the software package Simulink.

2.4 Simulation

Substantial progress has been made since the early 1960s in the development of general modelling and simulation methodologies that may be applied to a wide range of problems in the physical, social, and management sciences. The literature in this area is extensive and much is not directly useful to the simulation practitioner (Lovatt, 1992). But they are especially beneficial to the author in developing an understanding of the field.

The techniques of modelling and simulation of a heat pump serving as a heating system are investigated by Chen (1992). In modelling, a second-degree polynomial equation in both the evaporating and condensing temperatures is chosen as the form of the equation that characterises the heat transfer rate at the evaporator of the system. A number-processing operation is used to translate the catalogue data of components into nine simultaneous equations that are solved for the nine unknowns by a computer program. In simulation, the equation of the heat transfer rate at the evaporator, along with the energy balance equation for the evaporator, the energy balance equation for the condenser, and the ratio of heat rejected at the condenser to that absorbed at the evaporator constitute a set of four simultaneous equations. These equations, three of them are non-linear, are solved by the Newton-Raphson routine.

A dynamic model for an evaporator system of a sulphite plant at Lielahti, Finland, was developed by Koivo and Huikku (1988). The system consists of seven evaporators. Measurements were made on the plant to verify the model. An interactive computer simulator was constructed in order to study the dynamic behaviour of the system. In addition, the economic and dynamic significance of process changes and control strategies are investigated.

A general, multipurpose simulation and control package was described which assists with the design, analysis and digital simulation of linear, time-invariant dynamic systems. The process model was defined in block diagram form with transfer functions for each component. Experimental and simulated data obtained by applying the package to computer controlled pilot plants at the University of Alberta demonstrated the validity of the methods and the significant improvements possible through the use of modern control techniques (Fisher et al., 1972).

A review by Zeigler at al. (1979) included many papers that cast a light on the problem of applying a methodology to systems modelling and simulation. The similarities and differences between the concepts of design and modelling were identified. Design has a definite objective, a tangible end product, and a definite life span. On the other hand, the objectives of a model are often indefinite, the end product is usually something as intangible as "understanding", and a model invariably has a life cycle which involves repeated cycles of formulation, implementation and validation.

Elzas (1979) constructed a definition of "robust simulation" which covered computational algorithms, assuring model description validity, program integrity, program structure, program readability, and bounds for permissible experimentation. A robust simulation, in Elzas' view, ought to:

- free the user from artificial constraints which would otherwise restrict use of the simulation.
- Ensure that the models are appropriately combined and used.
- Protect against numerical pitfalls and attempts to use the models outside their bounds of applicability.

Oliver et al. (1974) described the computer simulation and its use in evaluating the multi-variable control schemes. Simulation results were presented to show the potential of these design approaches for industrial applications, and experimental data from the actual evaporator were included as a validation of the results obtained from the simulation.

In the chemical process industry, large, realistic non-linear problems are now routinely being solved via computer simulation. The general strategy for the simulation of complex processes follows a fairly well defined path, consisting of the commonsense steps given in the following block diagram (Ramirez, 1997).

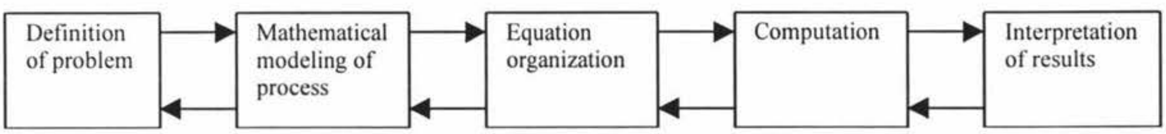


Figure 2.3: General Strategy of Process Simulation (Ramirez, 1997)

Matlab is an interactive program for numerical computation and data visualisation; it is used extensively for developing mathematical models (Palm, 1998). It will be an important development and demonstration tool for this project.

2.5 Summary

Evaporator system model development has been the subject of considerable research, with the result that many of the models discussed in the literature are capable of predicting the thermodynamic behaviour of individual evaporator system, and a suitable tool for use in analysing the application of process control to the evaporator.

Sub-system modelling methodology has generally been used for evaporator model development, in which the process is divided into sub-system for the purpose of analysis. The benefits are in terms of flexibility.

Many features and concepts that have long been utilised in general system simulation had been applied to evaporator system simulation. The Simulink, a toolbox extension of the Matlab program, is selected as a dynamic simulation environment for this project.

Chapter 3 Modelling

3.1 Introduction

Before beginning a task of model development, one first needs to specify the purpose for which the model will be used. The purpose of the model affects the method of analysing the system and the complexity of the model.

The model in this project is primarily to be used as a simulation tool for the development and testing of a pilot-scale spinning cone evaporator and its control system. For this purpose the model is required to mimic the behaviour of the actual system in order to generate accurate estimates of the measurable outputs in response to the various input and disturbance signals. In this respect the model is not required to be extremely complex and a dynamic lumped-parameter analysis is satisfactory. The steady-state behaviour of the system may be considered by the dynamic model developed using the method which assumes the time parameter is maximum ($t=\infty$). A system approach is taken and therefore a distributed analysis of heat transfers, flow regimes and localised variations of steam properties are not considered. The model is based on energy balances, mass balances and/or thermodynamics through each part of the system. The analysis is simplified and hence a number of physical assumptions are required. Once the system's basic characteristics have been described by the model, the model is tuned using a variety of coefficients and constants in order to quantitatively verify the model. The final model may not be the most accurate.

3.2 Sub-system modelling approach

The main differences between the spinning cone evaporator system and the other evaporator systems are on the evaporation part and product transport part because it applies the centrifugal force to enhance the evaporation. Therefore, the development of the spinning cone evaporator system model focuses on these two parts. Its feed, condenser, vacuum and compressor parts are similar to the other evaporator systems. Russell (1997) and Winchester (1999) developed the complete models for these parts.

The simplest models of the condenser and the compressor parts will be used, however the feed and vacuum parts may not be included in this project.

The basis for the approach used for the spinning cone evaporator model development was taken from the work of Russell (1997) and Winchester (1999). They developed the dynamic models for the multi-effect falling film evaporators and used a system approach in which the process was divided into sub-systems for the purpose of analysis. A similar analysis is used for the development of spinning cone evaporator model and the system consists of the four sub-systems (shown in Figure 3.1), they are the evaporation sub-system, compressor sub-system, condenser sub-system and product transport sub-system.

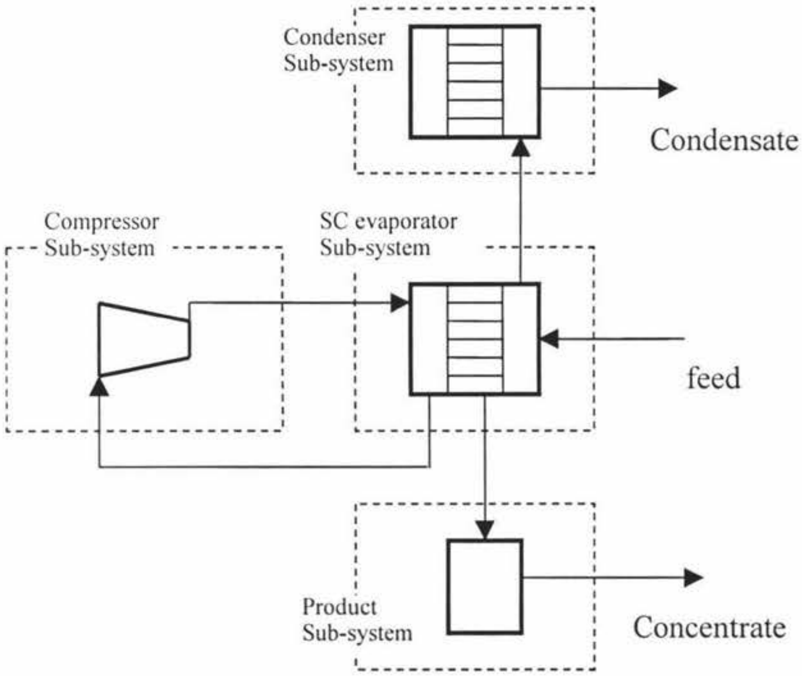


Figure 3.1: Outline of spinning cone evaporator model

3.2.1 System model

The system model is created to describe the dynamic relationships between the input variables (cooling water flowrate, M_c, speed of compressor, N_{comp}, feed flowrate, M_f, feed temperature, T_f and mass composition of feed dry matter, w_f) and the output variables (outlet temperature of cooling water, T_{co}, evaporating temperature, T_e, mass composition of product dry matter, w_p and product flowrate, M_p), shown in Figure 3.2. The sub-system modelling techniques of the evaporator are used to create the dynamic model. Each sub-system is analysed and their equations are derived from the first law of thermodynamics and the general mass and energy balances.

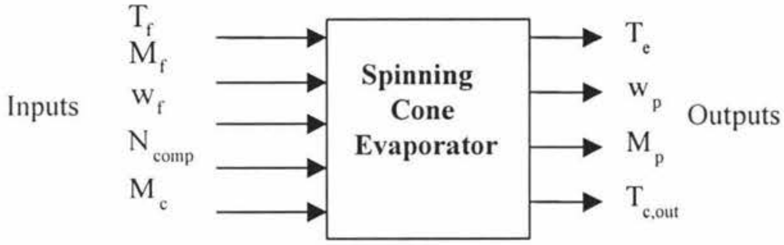


Figure 3.2: Block diagram of spinning cone evaporator model

3.2.2 Evaporation sub-system

This models the mass and heat transfers occurring in the evaporation process which result in a flow of un-evaporated liquid, a flow of vapour, and the heat flows involved in the evaporation of the product. Figure 3.3 shows the boundaries and block diagram of this sub-system. Under the action of the centrifugal force and gravitational fields, the pressure gradient of the liquid to be evaporated may produce on the spinning cone. Firstly, basic equations describing transport phenomena in centrifugation are used to create an equation which expresses the relationship between the pressure gradient and the radius of the cone. Secondly, this equation combines with the Clausius-Clapeyron equation (Hougen et al., 1965) to derive the temperature gradient on the cone. Then the temperature difference between the steam and evaporating sides can be derived, which is the most important term to develop the model of the evaporation sub-system. The overall heat transfer coefficient will be taken from the work of Chen (1997). The fourth procedure is the general evaporator modelling method, which creates the model of the evaporation sub-system using the mass and heat balances. This model gives the dynamic relationships between the inputs (steam temperature, T_s, speed of cone, ω, pressure of evaporating vapour, P_e, feed flowrate, M_f, feed temperature, T_f and mass composition of feed dry matter, w_f) and the outputs (evaporating temperature, T_e, evaporating and flashing vapour flowrate, M_v, mass composition of un-evaporated liquid dry matter, w_p, and un-evaporated liquid flowrate, M_l).

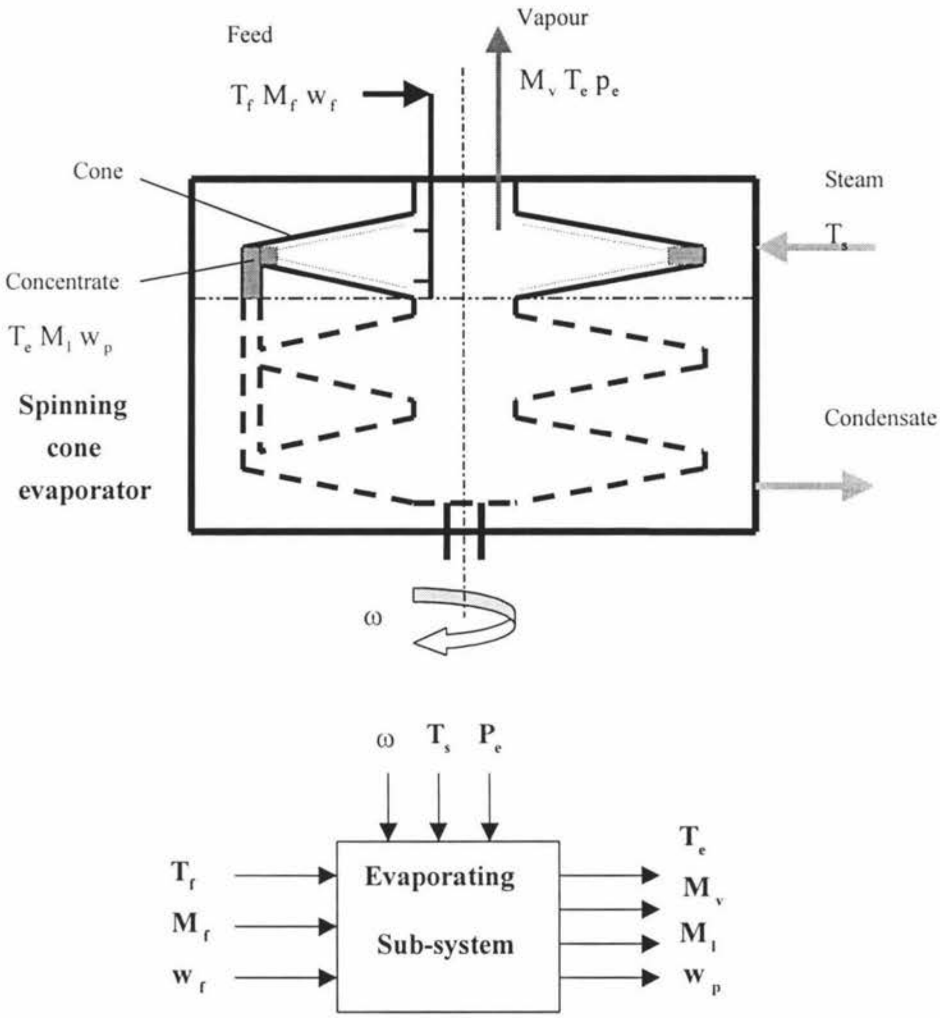


Figure 3.3: Evaporation sub-system

3.2.3 Compressor sub-system

This models the steady-state relationships between the inputs (compressor speed, N_{comp}, evaporating temperature, T_e, vapour flowrate to be compressed, M_v) and the outputs (steam temperature, T_s and steam flowrate, M_{comp}). Figure 3.4 gives the boundaries and block diagram of this sub-system.

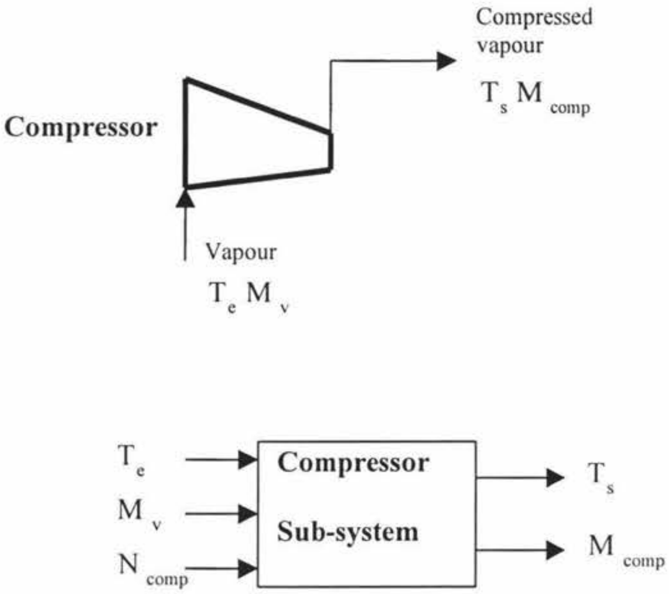


Figure 3.4: Compressor sub-system

3.2.4 Condenser sub-system

This models the rise in cooling water temperature due to the condensate and gives the steady-state relationship between the cooling water flowrate, M_c, steam flowrate, M_{comp} and outlet temperature of the cooling water, $T_{c,out}$. Figure 3.5 gives the boundaries and block diagram of this sub-system.

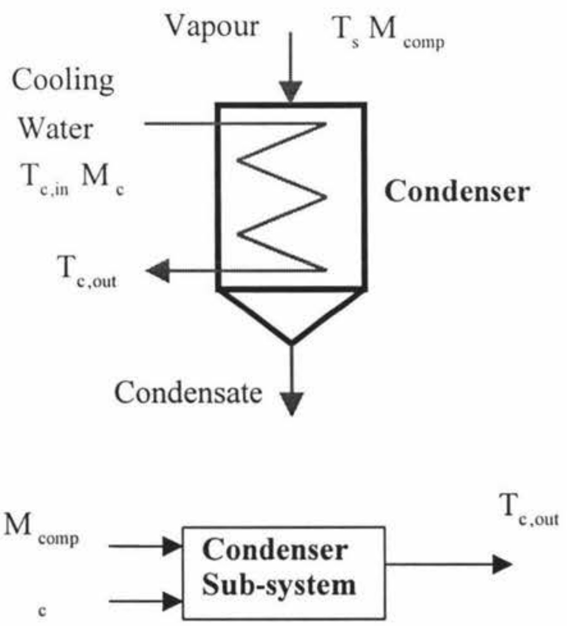


Figure 3.5: Condenser sub-system

3.2.5 Product transport sub-system

Figure 3.6 gives the boundaries and block diagram of this sub-system. This model gives the dynamic relationships between the inputs (speed of cone, ω, pressure of evaporating vapour, P_e, un-evaporated liquid flowrate, M_l) and the outputs (product flowrate, M_p).

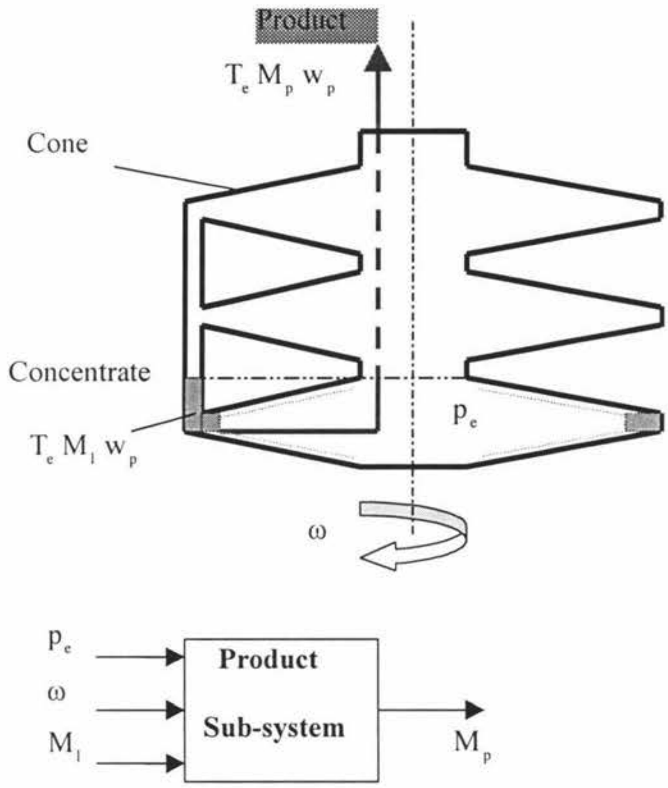


Figure 3.6: Product sub-system

3.3 Evaporation sub-system

Starting on the evaporation sub-system, the heart of this sub-system is the externally heated rotating heat transfer surface in the form of a truncated cone, as shown in Figure 1.3. The feed enters at the inner rotating edge and is evenly distributed by the centrifugal force over the conical surfaces of the evaporator. The centrifugal force caused by the speed of rotation gives rise to an extremely thin, fast-moving liquid film. As the thin film moves outwards, some of the solvent is evaporated until the film reaches the outer edge, where the concentrate is withdrawn.

To derive the flowrate and the dry matter mass composition of un-evaporated liquid, we need to know the evaporated and flashed vapour flowrates. Chen (1997) derived the evaporated vapour flowrate from the inner surface of the rotating cone, and

Winchester (1997) determined the mass flowrate of flashed water vapour by making static mass and enthalpy balances. From their works, these two flowrates vary with the evaporating temperature. So it is important to determine the evaporating temperature.

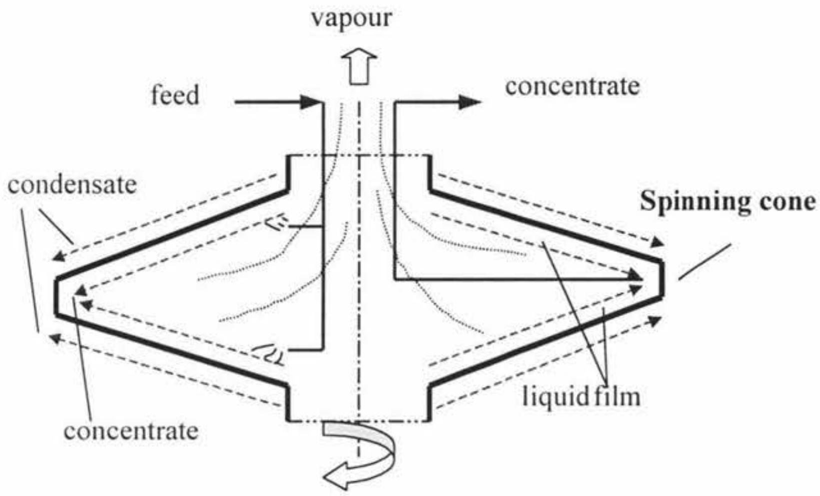


Figure 3.7: Principle schematic of SCE

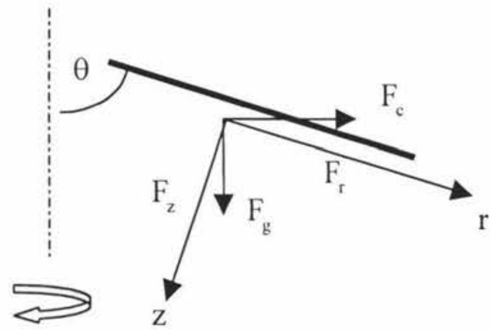


Figure 3.8: Schematic diagram of the cone used in the force analysis

Under the action of the centrifugal force and gravitational fields, the pressure and temperature gradients of the liquid to be evaporated may be produced along the spinning cone. An assumption was made by Chen (1997), which the temperature of the thin film of the liquid is equal to the evaporating temperature. Then we make the liquid film flow and energy analysis as follows.

As shown in the Figure 3.7, a cone rotates about its vertical axis and the liquid is fed onto the inner surface. The liquid may be considered to be rotating about a vertical axis, it will then be subjected to vertical forces due to gravity and centrifugal forces in a horizontal plane. For the z-r coordinates as shown in Figure 3.8, the total force on the liquid and pressure distribution is then obtained by summing the two components. In z direction, the pressure gradient of the liquid attributed to the centrifugal force and force of gravity on a small element of liquid is given by (Coulson et al., 1999):

$$\frac{\partial P}{\partial z} = F_z = F_g \cdot \sin\theta - F_c \cdot \cos\theta \qquad 3.1$$

Where: P is the pressure of liquid

F_Z is the force on a small element of liquid at z coordinate

F_g is the gravity force on a small element of liquid

F_c is the centrifugal force on a small element of liquid

θ is the half angle of the cone

The forces act in the r plane and the resulting pressure gradient may be obtained by taking a force balance on a small element of liquid:

$$\frac{\partial P}{\partial r} = F_r = F_g \cdot \cos\theta + F_c \cdot \sin\theta \qquad 3.2$$

Where: F_r is the force on a small element of liquid at r coordinate

Now:

$$dP = \frac{\partial P}{\partial z} dz + \frac{\partial P}{\partial r} dr \qquad 3.3$$

Substituting for $\partial p/\partial z$ and $\partial p/\partial r$ from equations 3.1 and 3.2:

$$dP = (F_g \cdot \sin\theta - F_c \cdot \cos\theta)dz + (F_g \cdot \cos\theta + F_c \cdot \sin\theta)dr \qquad 3.4$$

The force of gravity on a small element of liquid is present as Eq.3.5:

$$F_g = \rho g \qquad 3.5$$

Where: g is the acceleration due to gravity

ρ is density of fluid

And the centrifugal force on a small element of liquid is present as Eq.3.6:

$$F_c = \rho\omega^2 r \qquad 3.6$$

Where: ω is the angular velocity of rotation

In this project, the angular velocity of rotation (ω) is approximate 150 rad/s, the radius of cone is about 0.1m. So we have $F_c >> F_g$, and θ is nearly 90 degrees, thus F_g and $\cos\theta$ can be ignored. Eq.3.4 simplifies to:

$$dp = \rho\omega^2 r \sin\theta dr \qquad 3.7$$

Here ρ is not a constant since we are discussing the vapour. Assuming vapour and steam are ideal gases, the ideal gas equation can be used:

$$pv = nRT \qquad 3.8$$

Where: n is the molar unit of the gas

v is volume of the gas

R the universal gas constant

T is the absolute temperature

According to the definition of the density of the material and the ideal gas equation, the density of the gas can be calculated as follows:

$$\rho = \frac{m}{v} = \frac{nM_m}{v} = \frac{M_m P}{RT} \qquad 3.9$$

Where: m is the mass of gas

M_m is the molecular weight of gas

Thus, Eq.3.7 changes to:

$$dP = \frac{M_m P}{RT}\omega^2 r \sin\theta dr \qquad 3.10$$

Rearrange Eq.3.10:

$$\frac{dP}{P} = \frac{M_m}{RT}\omega^2 r \sin\theta dr \qquad 3.11$$

From this equation we can predict the pressure at any distance from the axis. The change in pressure is expected to be significant. Assume ω is a constant, the thicknesses of the liquid on both sides of the cone and the cone thickness are very small compared to the length of cone.

For vapour,

$$\frac{dP_v}{P_v} = \frac{M_v \omega^2 \sin\theta}{RT_e} rdr \qquad 3.12$$

where: P_v is the vapour pressure of the effect side

M_v is the vapour molecular weight

T_e is the vapour temperature

For steam (the shell rotates as well in this project),

$$\frac{dP_s}{P_s} = \frac{M_s \omega^2 \sin\theta}{RT_s} rdr \qquad 3.13$$

where: P_s is the steam pressure of the steam side

M_s is the steam molecular weight

T_s is the steam temperature

To find the temperature of saturated vapour/steam from this equation we can use the Clausius-Clapeyron equation (Hougen et al., 1965)

$$\frac{\partial P}{P} = \frac{M_m \lambda \partial T}{RT^2} \qquad 3.14$$

Where: λ is the enthalpy of vaporization

Combining Eq.3.11 and Eq.3.14 gives

$$\frac{dT}{T} = \frac{\omega^2 \sin\theta rdr}{\lambda} \qquad 3.15$$

Integrating both sides of equation 3.15, we obtain:

$$\ln\left(\frac{T}{T_0}\right) = \int_{r0}^{r} \frac{\omega^2 \sin\theta}{\lambda} rdr \qquad 3.16$$

Rearranging Eq.3.16 gives

$$T = T_0 e^{\int_{r0}^{r} \frac{\omega^2 \sin\theta}{\lambda} rdr} \qquad 3.17$$

Substituting Eq.3.17 into Eq.3.12, the vapour temperature at any point is thus:

$$T_e = T_{e0} e^{\int_{r0}^{r} \frac{\omega^2 \sin\vartheta}{\lambda_e} r dr} \qquad 3.18$$

And for steam:

$$T_s = T_{s0} e^{\int_{r0}^{r} \frac{\omega^2 \sin\vartheta}{\lambda_s} r dr} \qquad 3.19$$

From Eq.3.18 and 3.19, the temperature difference driving heat transfer can be got.

$$\Delta T = T_s - T_e \qquad 3.20$$

Now, we consider the overall heat transfer coefficient, which was derived by Chen (1997):

$$U = \frac{1}{\frac{1}{h_s} + \frac{2\delta_w}{k_w}\left(\frac{R_i + r_i}{2\delta_w} + 1\right)\ln\left(1 + \frac{2\delta_w}{R_i + r_i}\right) + \left(1 + \frac{2\delta_w}{R_i + r_i}\right)\frac{1}{h_l^{'}}} \qquad 3.21$$

Where: U is the overall heat transfer coefficient

h_s is the steam side heat transfer coefficient

$h_l^{'}$ is the liquid side heat transfer coefficient

k_w is the thermal conductivity of wall

δ_w is the thickness of the wall

R_i is the inner radius at the bottom of cone

r_i is the inner radius at the top of cone

According to Eq.1.1, we can obtain the overall rate of heat transfer from the steam side to the liquid side using the temperature difference ΔT and the overall heat transfer coefficient h_l. Then, the evaporation rate can be calculated by the overall rate of heat transfer. It is a important term in this sub-system.

The evaporation rate from the inner surfaces of the cones can be written as follows (Chen, 1997):

$$M_{evap}(t) = \frac{UA_e}{\lambda}(T_s - T_e) \qquad 3.22$$

Where: M_{evap} is the evaporation rate

U is the heat transfer coefficient

A_e is the heat transfer area of the cone

The temperature at which evaporation occurs in the spinning cone can be determined from a dynamic energy balance. Because the residence time is ultra short (Kessler, 1981 and Billet, 1989), the evaporation is considered to take place immediately. When considering heat transfer from the steam side to the liquid side, the heat loss can be ignored. By making other usual assumptions (i.e., potential and kinetic terms are negligible, liquid/metal internal energy equivalent to enthalpy, enthalpy independent of pressure and constant heat capacities) the following differential equation is derived (Winchester and Marsh, 1999).

$$M_e \frac{dT_e}{dt} = q_{feed} + q_{heat} - q_v \qquad 3.23$$

Where: q_{feed} is the net enthalpy of the feed

q_{heat} is the heat flow through the evaporator cone surface

q_v is the latent enthalpy of the evaporating vapour

M_e is the thermal mass of evaporator including liquid, vapour, and metal

The heat from the feed is given by

$$q_{feed} = M_f C_{pm}(T_f(t) - T_e(t)) \qquad 3.24$$

Where: T_f is the temperature of the feed

c_{pm} is the specific heat capacity of the milk

M_f is mass flow of liquid to the evaporator

The energy flow through the evaporator is considered as

$$q_{heat} = A_e U(T_s(t) - T_e(t)) \qquad 3.25$$

The heat flow of the evaporating vapour is given by

$$q_v = M_v \lambda \qquad 3.26$$

Where: M_v is the vapour removal rate

In formulations of the mathematical model for the evaporator system considered here, following assumptions are made:

1. The vapour phase is always in equilibrium with the liquid phase
2. No fouling is formed
3. Latent heat, heat transfer coefficients are constants

4. Boiling point elevation is negligible

The mass balance on the liquid is given by the following equation (Lima Hon et al., 1979):

$$\frac{dW}{dt} = \rho_p A_l \frac{dh}{dt} = M_f - M_v - M_l \tag{3.27}$$

Where: W is the liquid mass inside evaporator

ρ_p is the discharge density

A_l is the area of the surface of the non-evaporated liquid

M_l is the mass flowrate of the non-evaporated liquid

h is the thickness of liquid

The mass balance on the vapour is given by the following equation

$$\frac{dM}{dt} = M_{evap} + M_{flash} - M_v \tag{3.28}$$

Where: M is the mass of gas from evaporation

M_{flash} is the flashing rate

If the liquid enters the evaporator at a temperature higher than the saturation temperature, then some of liquid will flash. The mass flow of flashed water vapour can be determined by making static mass and enthalpy balances (Winchester, 1999).

$$M_{flash} = \frac{M_f C_p}{\lambda}\left(T_f - T_e\right) \tag{3.29}$$

For Eq.3.28, we assume there is no the gas mass accumulated because of the small volume of the evaporating space in the spinning cone evaporator. So the steady state can be considered, Eq.3.28 reduces to

$$M_v = M_{evap} + M_{flash} \tag{3.30}$$

The mass flowrate of the non-evaporated liquid is given by the equation (Billet, 1989)

$$M_l = \zeta h \tag{3.31}$$

Where: ζ is the correlation coefficient

For the calculation of the dry matter, we have

$$w_p = \frac{M_f}{M_l} w_f \qquad 3.32$$

Where: w_f is mass composition of dry matter in feed

w_p is mass composition of dry matter in product

3.4 Compressor sub-system

A mechanical compression method is proposed to be used in this project, then a fan compressor is chosen. After evaporation has taken place in a spinning evaporator, the evaporating vapour then passes to a fan compressor to be compressed to 0.024 MPa, which corresponds to a saturation temperature of 64 °C (Jabson, personal communication). Figure 3.9 shows a flow chart of a fan compressor.

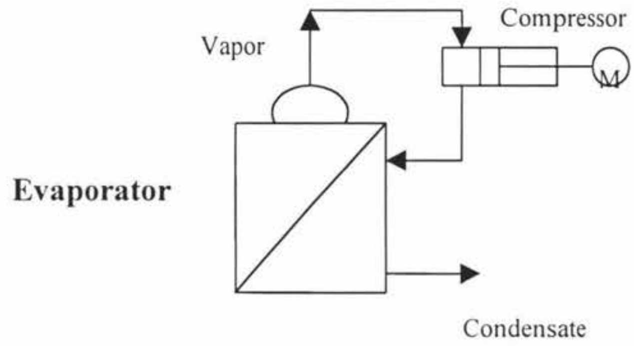


Figure 3.9: Flow chart of a fan compressor

The temperature of vapour compressed (steam) and the steam mass flowrate may be derived using the mass and energy balances. The compressor is assumed to be frictionless and an adiabatic process.

Speed Equation. A model described the relationship between the compressor speed and the flow of compressed vapour was developed by Winchester (1999).

$$P_{comp} = a\ N_{comp}^2\ Q_v + b\ N_{comp}\ Q_v^2 + c\ Q_v^3 \qquad 3.33$$

Where: P_{comp} is power required to compress the vapour

N_{comp} is compressor speed

Q_v is flow of the compressed vapour

a, b and c are the dimensional coefficients

Compressor efficiency. The actual work done will be greater than that calculated, because of frictional loss and leakage in the compressor. The ratio of the theoretical work (or fluid power, P_{comp}) to the actual work (or total power input) is the efficiency and is denoted by η. The maximum efficiency of the fan compressor is about 70% (McCabe et al, 1993).

Power equation. The power required by an adiabatic compressor is readily calculated as follows, in SI units (McCabe et al., 1993):

$$P_{comp} = \frac{0.371 T_e \gamma Q_v}{(\gamma - 1)\eta}\left[\frac{T_{comp}}{T_e} - 1\right] \qquad 3.34$$

Where: $Q_v = (M_{evap} + M_{flash})/\rho$

P_{comp} is power required to compress the vapour, kW

$T_{comp} = T_{s,}$ K

η is compressor efficiency

γ is the ratio of specific heat

In reality the compressor is not an isentropic (adiabatic) process. It contains inefficiencies and this means that equation (3.34) is not an accurate equation for the compressor power supply. However, its general form can be used to model the power supplied by a compressor.

Temperature of the compressed vapour. When a compressor is selected and its power is known, the temperature of the compressed vapours can be derived from Eq.3.34:

$$T_{comp} = T_e + \frac{P_{comp}(\gamma - 1)\eta}{0.371\gamma Q_v} \qquad 3.35$$

Steam flow. Because the water will be sprayed into the vapour to prevent vapour superheating during compression, the steam flow is not equal to the compressed vapour. It can be derived as following:

$$M_{comp} = M_v \qquad 3.36$$

3.5 Condenser sub-system

The evaporator in this project runs at temperatures ranging between 60—80°C and must therefore be operated under a vacuum. To create a vacuum the following have to be removed:

- steam or vapour produced by evaporation
- gases, either dissolved or dispersed in the feed
- air which has leaked into the plant

The vapour constitutes by far the greatest part of the total volume of the gases. Vapour is removed by condensation on water cooled surfaces. Figure 3.10 shows a diagram of a generic condenser. We consider that the condenser uses cooling water inside and condensing steam on the outsides of the tubes. The inputs are the inlet liquid temperature ($T_{c,in}$), liquid flow (M_c) and the condenser shell temperature (T_s). The outputs are the outlet liquid temperature ($T_{c,out}$).

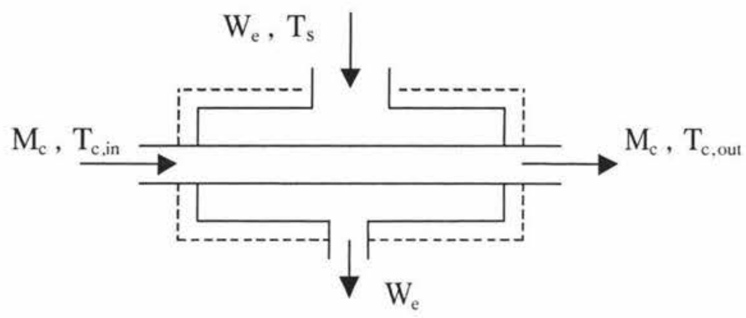


Figure 3.10: Diagram of a generic condenser

The condenser model is developed by making an energy balance over the condenser tubes. In practice some assumptions are made; the inlet temperature of the cooling water is constant, the temperature of the condensate is same as the condensed vapour. For a control volume, the following version of the first law of thermodynamics is produced, which is the form we will use here.

$$\frac{dH}{dt} = \sum MH + Q + W_s - q_{loss} \qquad 3.37$$

Where: H= enthalpy

M= flowrate

Q= total inward net heat flow

W_s= additional work

q_{loss} is the heat loss from the evaporator surfaces

In this project, there is not W_s produced, the differential change of enthalpy in the condenser is given by the following:

$$\frac{dH}{dt} = M_{cond}\left(H_{v,in} - H_{v,out}\right) + M_c\left(H_{c,in} - H_{c,out}\right) - q_{heat} - q_{loss} \quad 3.38$$

Where: subscripts "in" and "out" denote inlet and outlet values, respectively

M_c is the liquid flow

M_{cond}=M_{comp} is the condensed vapour flow

For the condenser, the steady state model is used here. The enthalpy inflow equals the enthalpy outflow:

$$M_c(H_{c,out} - H_{c,in}) = M_{cond}(H_{v,in} - H_{v,out}) - q_{heat} - q_{loss} \quad 3.39$$

If we assume a constant specific heat for the coolant and that the condensate leaves at the saturation temperature, and the whole vapour are condensed, Eq.3.40 becomes

$$M_c c_{pc}\left(T_{c,out} - T_{c,in}\right) = M_{cond}\lambda - q_{heat} - q_{loss} \quad 3.40$$

Where: c_{pc} is the heat capacity of water

$T_{c,in}$ is the inlet liquid temperature

$T_{c,out}$ the outlet liquid temperature

The heat loss to the surroundings is given by

$$q_{loss} = A_{el} h_{el} (T_s(t) - T_a(t)) \quad 3.41$$

Where: h_{el} is loss overall heat transfer coefficient

A_{el} is the surface area of the sub-system

T_a is the ambient temperature

Because the coolant flow rate M_c and inlet temperature $T_{c,in}$ are known, Eq.3.41 relates the coolant outlet temperature $T_{c,out}$ to the amount of vapour condensed M_{cond}.

$$T_{c,out} = T_{c,in} + \frac{1}{M_c c_{pc}}\left(M_{cond}\lambda - q_{heat} - q_{loss}\right) \quad 3.42$$

3.6 Product sub-system

The product transport sub-system removes the concentrate from the spinning cone evaporator. The concentrate is thrown to the periphery of the cone and escapes continuously through the product pipe by the centrifugal force. Thus, an understanding of the product flow rate through the pipe is important in the development of this sub-system.

Figure 3.11 shows a generalized diagram of the product transport sub-system of the spinning cone evaporator. As the thin film formed moves outwards, some of the solvent is evaporated until the film reaches the outer edge, where the concentrate is withdrawn.

From the figure 3.11, the pressure at the point ① consists of three parts: the pressure produced by the centrifugal force, the pressure produced by the rotating liquid and the evaporating pressure. The flow of the concentrate within the product pipe can be assumed to be one-dimensional and in the radius direction only.

To calculate the flowrate out of the evaporator (Q_p), an energy balance between the input point ① and discharge point ② is required. Using Bernoulli's equation one arrives at (Geankoplis, 1993):

$$P_i + \rho_p g \frac{\delta}{2} + \rho_p \frac{v_1^2}{2} = P_o + \rho_p g y_2 + \rho_p \frac{v_2^2}{2} + \Delta P_{fric} + \Delta P_{exit} \qquad 3.43$$

Where: P_i is the pressure of the input point

P_o is the pressure of the discharge point

ΔP_{fric} is the pressure drop due to the friction in pipeline

ΔP_{exit} is the pressure drop across the discharge nozzle

ρ_p is the discharge density

δ is the height of the cone

g is the gravitational acceleration constant

v_1 is the flow velocity at the input point

v_2 is the velocity at the discharge point

y_2 is the height of the discharge point

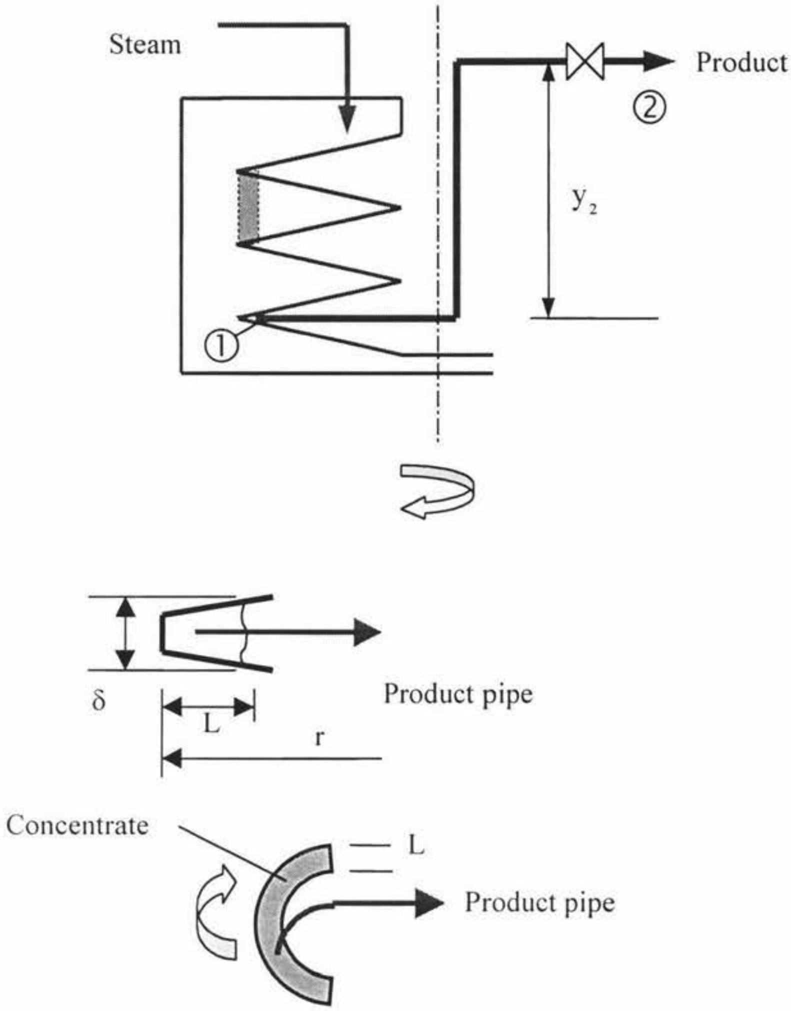


Figure 3.11: Generalized diagram of product transport sub-system

Pressure of the input point. Using a force balance, the pressure at the input point (P_i) may be

$$P_i = P_c + P_{pr} + P_e \tag{3.44}$$

Where: P_c is the pressure produced by the centrifugal force

P_{pr} is the pressure produced by the rotating liquid

P_e is the evaporating pressure

For the pressure produced by the centrifugal force, a force balance on a sector of liquid in the rotating disc gives the pressure gradient at a radius r:

$$\frac{\partial P_c}{\partial t} = \rho_p \omega^2 r \tag{3.45}$$

The centrifugal pressure gradient is a function of radius of rotation r, and increases towards the end of the cone. Integration of equation 3.45 at a given interval L gives the pressure P_c (Hsu, 1981):

$$P_c = \frac{\rho_p \omega^2}{2}\left[(r-L)^2 - r^2\right] \qquad 3.46$$

Where: L is the height of the non-evaporated liquid.

For the pressure produced by the rotating liquid, the gravity and the frictional loss at input point are ignored. P_v is then given by:

$$P_{pr} = \frac{\rho_p v_c^2}{2} \qquad 3.47$$

Where: v_c is the tangential velocity of the liquid.

Substituting $v_c = \omega r$, the equation 3.47 becomes:

$$P_{pr} = \frac{\rho_p \omega^2 r^2}{2} \qquad 3.48$$

Thus from equation 3.44:

$$P_i = \frac{\rho_p \omega^2}{2}\left[(r-L)^2 - r^2\right] + \frac{\rho_p \omega^2 r^2}{2} + P_e \qquad 3.49$$

Pressure drop due to the friction in pipe line. If the frictional losses can be expressed by Fanning's equation, the pressure drop for liquid flowing in a pipe is obtained from the following equation (Coulson et al., 1999):

$$\Delta P_{fric} = 4f \frac{l}{d} \frac{\rho_p v_l^2}{2} \qquad 3.50$$

Where: f is Fanning friction factor,

d is the diameter of pipe,

l is the length of pipe.

Pressure drop across the discharge nozzle. At the pipe exit, the diameter of the pipe suddenly increases. Thus fluid with a relatively high velocity will be injected into relatively slow moving fluid. The loss can be given by (Coulson et al, 1999):

$$\Delta P_{exit} = \frac{\rho_p v_l^2}{2} \qquad 3.51$$

Product flowrate. The velocity in the pipe is given by the volumetric flowrate divided by the cross-sectional area of the pipe at the particular point (Hsu, 1981). It may be written as:

$$v = \frac{Q_p}{A_d} \qquad 3.52$$

Where: Q_p is the product flowrate out of the spinning cone evaporator,

A_d is cross-sectional area of the pipe,

v is the fluid velocity in the pipe.

Assume $v_l = v_2 = v$, ρ_p=constant, P_o=ambient pressure, ignore the height of the cone (δ) term, Bernoulli's equation 3.43 becomes:

$$P_i = P_o + \rho_p g y_2 + \Delta P_{fric} + \Delta P_{exit} \qquad 3.53$$

Substituting from equations 3.49—3.52 into equation 3.53 and re-arranging, Q_p may be

$$Q_p = A_d \sqrt{\frac{2\left(P_e - P_o - \rho_p g y_2\right) + \rho_p \omega^2 (r - L)^2}{\rho_p \left(4\frac{fl}{d} + 1\right)}} \qquad 3.54$$

Height of the non-evaporated liquid. For this sub-system, a differential equation with the height of the non-evaporated liquid, L, as the state variable, is used:

$$\rho_p A_l \frac{dL}{dt} = M_l - M_p \qquad 3.55$$

Where: A_l is the area of the surface of the non-evaporated liquid

M_p is the mass flowrate of the product

From equation 3.55, the state variable, L, therefore is given by:

$$\frac{dL}{dt} = \frac{M_l}{\rho_p A_l} - \frac{Q_p}{A_l} \qquad 3.56$$

Substituting equation 3.55 into equation 3.56, the equation becomes:

$$\frac{dL}{dt} = \frac{M_l}{\rho_p A_l} - \frac{A_d}{A_l}\sqrt{\frac{2(P_e - P_o - \rho_p g y_2) + \rho_p \omega^2 (r-L)^2}{\rho_p\left(4\frac{fl}{d}+1\right)}} \quad 3.57$$

As shown in Figure 3.7, the area of the surface of the non-evaporated liquid can be given as following:

$$A_l = 2\pi r\delta \quad 3.58$$

$$A_d = \pi r_d^2 \quad 3.59$$

Where: r_d is the radius of the pipe.

3.7 Summary of model variables

The spinning cone evaporator model has five input variables, two state variables and four output variables. The following summarizes the variables used in the evaporator model.

Inputs

Five main input variables are:

M_f	Feed flow rate	kgs^{-1}
N_{comp}	Speed of compressor	rpm
M_c	Flow rate of cooling water	kgs^{-1}
w_f	Feed dry mass fraction	$kgkg^{-1}$
T_f	Feed temperature	K

There are five additional inputs that are considered to be constants for the evaporator model:

P_e	Evaporation pressure	kPa
T_a	Ambient temperature	K
P_a	Ambient pressure	kPa
ω	Speed of spinning cone	rpm
$T_{c,in}$	Inlet temperature of cooling water	K

State variables:

Two state variables are:

h	Average thickness of liquid	m
L	Height of the non-evaporated liquid	m

Outputs:

The model has four output variables that are used to compare with measurements from the actual pilot-scale plant:

M_p	Product flow rate	kgs^{-1}
w_p	Product mass fraction	$kgkg^{-1}$
T_e	Evaporating temperature	K
$T_{c,out}$	Outlet temperature of cooling water	K

3.8 Model parameters

The model parameters that have been calculated are discussed below. Further detail of the evaporator geometry and the general constants that were required is given in Appendix 1.

Heat transfer coefficients

The overall heat transfer coefficient has been discussed on Chapter 1 and its representation was given in equation 3.21. The following equations were developed by Chen (1997) and were used to estimate the heat transfer coefficients of the steam side and liquid side, h_s and h'_l.

The liquid side heat transfer coefficient (h'_l) is given as follows:

$$h_l^{'}(r)=\frac{1}{r-r0}\int_{r0}^{r}\frac{k_l dr}{\left[\dfrac{3(F_{in}-W_v)\nu}{2\pi gr\sin\theta\cos\theta\left(\dfrac{\rho}{\rho-\rho_v}\right)+2\pi r^2\omega^2\sin^3\theta}\right]^{\frac{1}{3}}} \qquad 3.60$$

Where: k_l is the thermal conductivity of liquid

ν is the kinematic viscosity

ρ_v is the density of vapour

And the steam side heat transfer coefficient (h_s) is given as follows:

$$h_s = \frac{k_l}{\left(\frac{\nu^2}{g\cos\theta}\right)^{\frac{1}{3}}}\left(\frac{2}{3}\frac{\Pr}{\mathrm{Ja}}\frac{\left(\frac{\nu^2}{g\cos\theta}\right)^{\frac{1}{3}}(\omega\sin\theta)^2}{g\cos\theta}\right)^{\frac{1}{4}} \qquad 3.61$$

Pr is the Prandtl number and defined as:

$$\Pr = \frac{C_p\mu}{k_l} \qquad 3.62$$

Ja is the Jacob number and defined as:

$$\mathrm{Ja} = \frac{C_p(T_s - T_{ws})}{\lambda_s'} \qquad 3.63$$

λ'_s is the effective latent heat and defined as:

$$\lambda_s' = \lambda_s\left(1 + \frac{0.68C_p(T_s - T_{ws})}{\lambda_s}\right) \qquad 3.64$$

T_{ws} is the surface temperature of cone on steam side and defined as:

$$T_{ws} = h_l(T_{wl} - T_e)\left(\frac{\delta_w}{k_w}\right) + T_{wl} \qquad 6.65$$

T_{wl} is the surface temperature of cone on liquid side and defined as:

$$T_{wl} = \frac{(T_s + T_e)}{2} \qquad 6.66$$

Other constants

The fanning friction factor, f (equation 3.50) for stainless steel was estimated using typical steel roughness properties. The value of f is taken as 0.009.

3.9 Conclusion

An understanding of the dynamic behaviour of the evaporator operation is important in this project. The main attention is focused on the factors affecting heat transfer and discharge flow in the spinning cone evaporator. The purpose-designed mathematical model has been developed to describe the dynamics of the evaporator system, and to provide a method of better understanding the principle and operation of the spinning cone evaporator.

Chapter 4 Model implementation

The objectives of the model implementation are to implement a simulation of the system model described in Chapter 3 and to verify the model comparing its simulated outputs to the spinning cone evaporator, with data calculated from the mathematical equations.

There are many simulation packages, which may provide convenient, ready-to-use modules or templates of commonly used components for a specific application. Among the better-known simulation packages, ACSL and ESL are for general systems, whereas EMTP and SPICE2 are mainly for simulating electrical and electronic circuits. Besides these, there are also dynamic simulation packages (HYSYS, Simulink, etc.) which are being used more and more frequently in the chemical process industries for process simulation and control system design (Ong, 1998).

With HYSYS, engineers need only develop a single process model that can be used from conceptual design through on-line applications to improve designs, optimise production and enhance decision-making. This enables the seamless integration of proprietary unit operations, reactions and property packages, and interaction with other applications to create powerful hybrid programs.

Simulink is an interactive tool for modelling, simulating, and analysing dynamic systems. It enables one to build graphical block diagrams, simulate dynamic systems, evaluate system performance, and refine the designs. Simulink integrates seamlessly with MATLAB, providing one with immediate access to an extensive range of analysis and design tools. These benefits make Simulink the tool of choice for control system design, DSP design, communications system design, and other simulation applications (Palm, 1998).

In this project, we applied Simulink to simulate the model of the spinning cone evaporator system and predict the behaviour of the system, because of the benefits presented above. This model will be applied as part of the spinning cone evaporator control system which will be developed using Simulink. With Simulink, systems are drawn on screen as block diagrams. Many elements of block diagrams are available, such as transfer functions, summing junctions, etc., as well as virtual input and output devices such as function generators and oscilloscopes. Simulink, integrated with Matlab and data can be easily transferred between the programs.

Using Simulink blocks, a graphical representation of the flow diagram can be provided, which contains all the transformations of information in the process. The Simulink simulation is very useful in this research study in that it provides as follows:

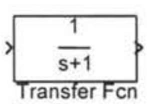
- Observing the actual system.
- Formulating a hypothesis or mathematical model to explain the observation.
- Predicting the behaviour of the system from solutions or properties of the mathematical model.
- Testing the validity of the hypothesis or mathematical model.

4.1 Simulink Basics

There are two major classes of items in Simulink: blocks and lines. Blocks are used to generate, modify, combine, output, and display signals. Lines are used to transfer signals from one block to another.

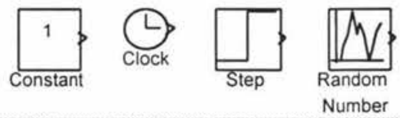
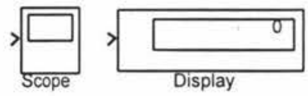
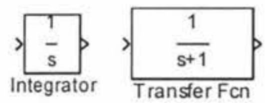
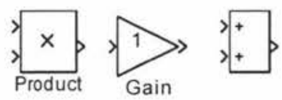
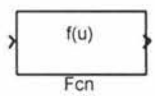
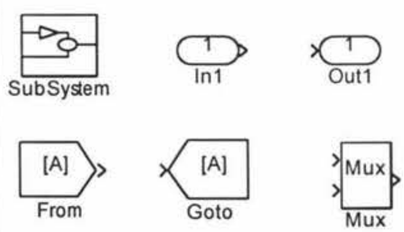
- **Blocks**

Blocks have zero to several input terminals and zero to several output terminals. Unused input terminals are indicated by a small open triangle. Unused output terminals are indicated by a small triangular point. The block shown below has an unused input terminal on the left and an unused output terminal on the right.



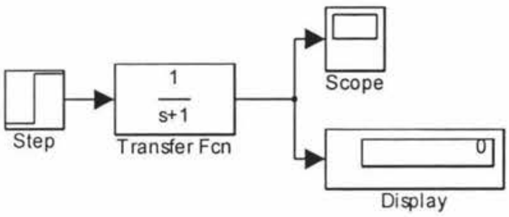
Simulink contains a large number of blocks from which models can be built. The blocks used in the spinning cone evaporator model are briefly introduced as Table 4.1.

Table 4.1: General blocks

Block	Function	Examples
Sources	generate various signals	
Sinks	output or display signals	
Continuous	integrator and transfer function	
Math	linear, continuous-time system elements and connections (summing junctions, gains, etc.)	
Functions	Fcn, Matlab Fcn and S-function, etc.	
Connections	Multiplex, Demultiplex, System Macros, etc	

- **Lines**

Lines transmit signals in the direction indicated by the arrow. Lines must always transmit signals from the output terminal of one block to the input terminal of another block. One exception to this is a line can tap off another line, splitting the signal to each



of two destination blocks, as shown below:

Lines can never inject a signal into another line; lines must be combined through the use of a block such as a summing junction.

A signal can be either a scalar signal or a vector signal. For Single-Input, Single-Output systems, scalar signals are generally used. For Multi-Input, Multi-Output systems, vector signals are often used, consisting of two or more scalar signals. The lines used to transmit scalar and vector signals are identical. The type of signal carried by a line is determined by the blocks on either end of the line.

4.2 Model implementation

The spinning cone evaporator model was implemented using Simulink. Figure 4.1 shows the structure of the simulation implemented in Simulink. The blocks at the top represent the overall model and global constants used in it. The second level contains the individual sub-systems and the bottom level elements within each sub-system.

Prior to setting up the system model in Simulink, we have already developed a mathematical model of the spinning cone evaporator system needed to simulate. This mathematical model of the dynamic system consists of a mix of integral and algebraic equations (see Eq.3.18—3.62). We consider that the inputs are independent variables and the output variables are dependent in the mathematical model. Rewriting the integral equations with the dependent-state variable expressed as some integral of a combination of independent variable and dependent variables, including itself, the

Simulink model can be obtained. Construction of the Simulink model can then follow the rearranged mathematical model.

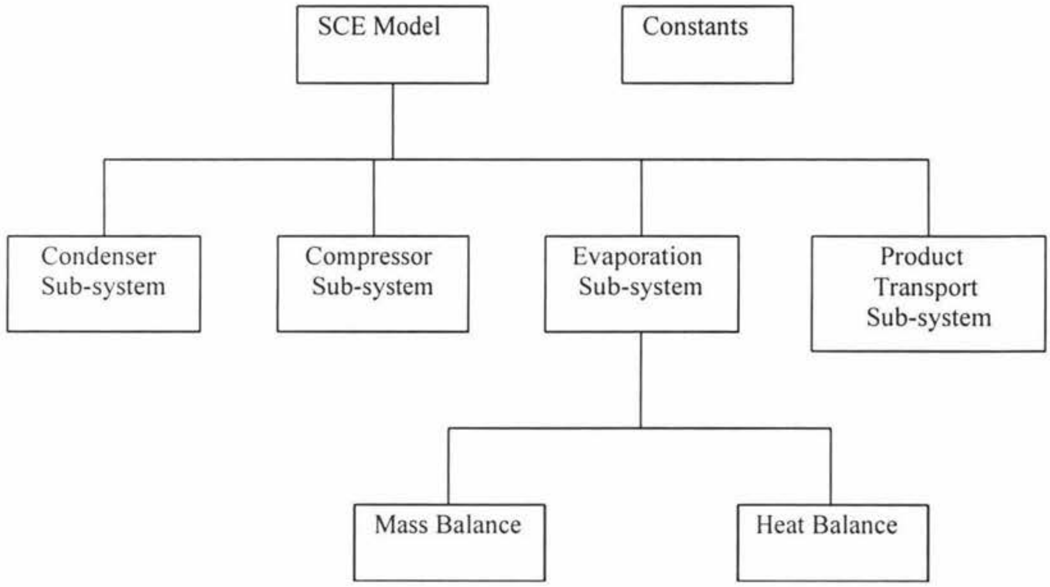


Figure 4.1 : Structure diagram of the evaporator model

- **Inputs**

We use five Simulink subsystem blocks to implement the five model inputs (cooling water flowrate, M_c, speed of compressor, N_{comp}, feed flowrate, M_f, feed temperature, T_f and mass composition of feed dry matter, w_f).

Two examples of the input simulations are that we implement the feed temperature and the feed flowrate. The input variable T_f is considered as a random variable and is of the form:

$$T_f = T_{f0}\left[1 + \text{rand} \cdot \sin(\pi \cdot \text{rand} \cdot t)\right] \tag{4.1}$$

Where: T_f is the feed temperature

T_{f0} is the initial value of the feed temperature

rand is a random number

The feed flowrate is implemented by a step and is of the form:

$$M_f = M_{f0} \cdot u(t) \qquad 4.2$$

Where: u(t) is a step function

M_f is the feed flowrate

M_{f0} is the initial value of the feed flowrate

Figure 4.2 and 4.3 show the contents of the Simulink diagrams for simulating a random feed temperature and a step feed flowrate. They can be changed by setting T_{f0} and rand, M_{f0} and u(t).

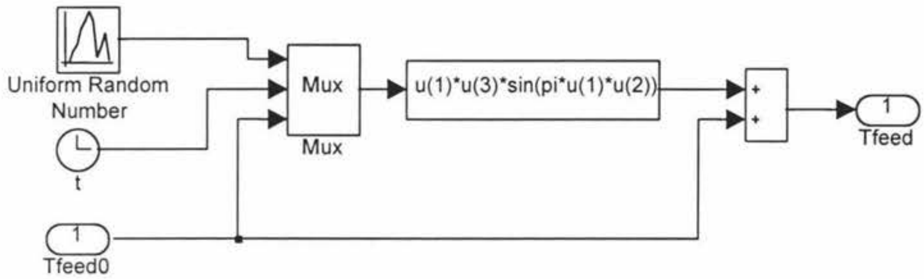


Figure 4.2: Simulink simulation of a random feed temperature

Figure 4.3: Simulink diagram of a step feed flowrate

The other input variables are similar to the feed temperature and given in Appendix 2.

- **Evaporation sub-system**

For evaporation sub-system, we implemented the Simulink model and use it to study the dynamic behaviour of the sub-system. This Simulink model was divided into two parts one calculating the energy balance and another the mass balance for the evaporation process.

In the mathematical description, the evaporating temperature can be rearranged from Eq.3.23.

$$T_e = \int \left(\frac{q_{feed} + q_{heat} - q_v}{M_e} \right) dt \qquad 4.3$$

Where: the mathematical equations of q_{feed}, q_{heat} and q_v are given in Eqs.3.24~3.26.

The equation 3.27 (mass balance) was rearranged to

$$h = \frac{1}{\rho_p A_l} \int (M_f - M_l - M_v) dt \qquad 4.4$$

Where: $Mv=M_{flash}+M_{evap}$, mathematical equation of M_{evap} was given in Eq.3.22, M_{flash} in Eq.3.29 and M_l in Eq.3.31.

Figure 4.4 shows an overall diagram of the Simulink system for the evaporation sub-system. The details inside the main blocks of overall diagram are given in Fig4.5 and 4.6.

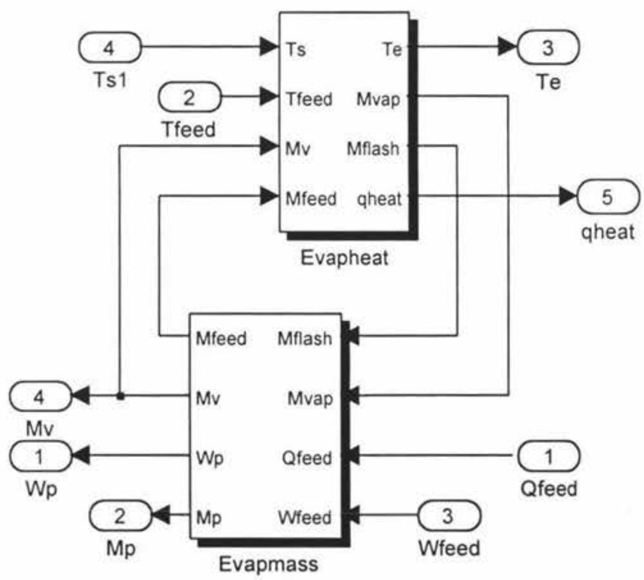


Figure 4.4: Overall diagram of Simulink system of evaporation sub-system

Figure 4.5 shows the contents of Simulink model for simulating the energy balance of the evaporation sub-system. In this simulation, the evaporating temperature, flash rate and evaporating rate are obtained.

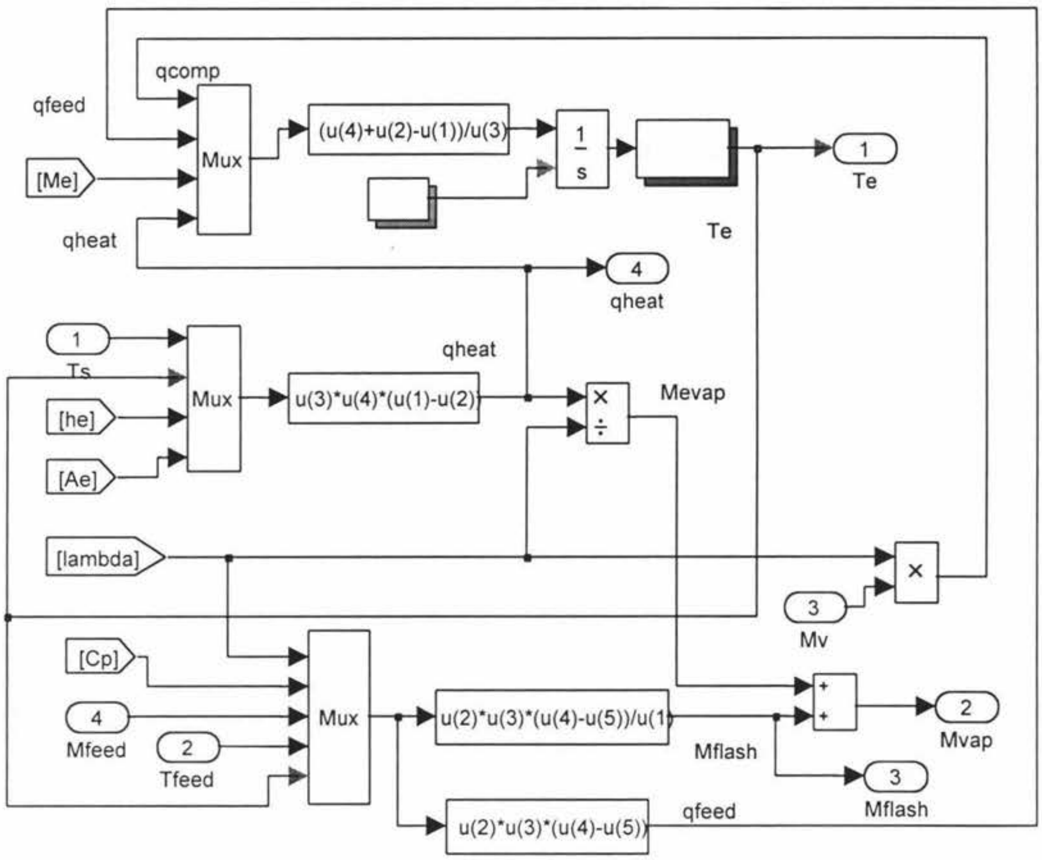


Figure 4.5: Simulink system of energy balance part

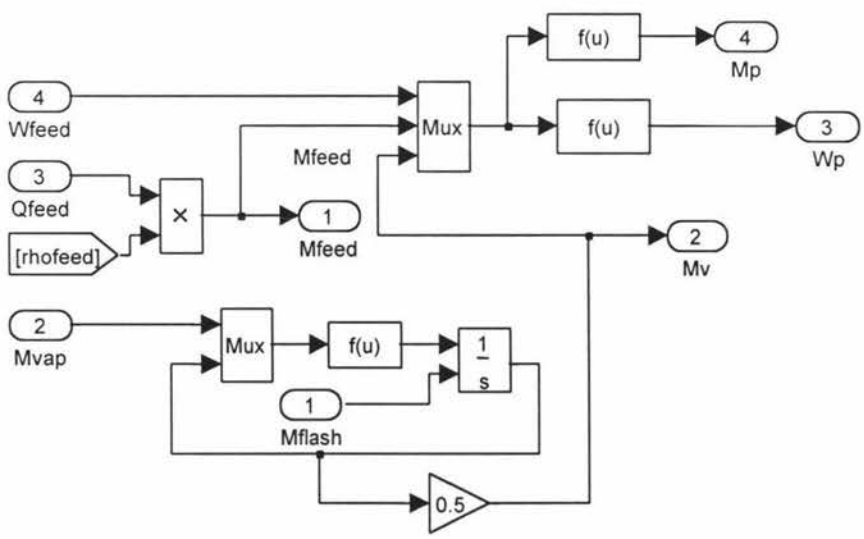


Figure 4.6: Simulink system of mass balance part

Figure 4.6 shows the contents of Simulink system for calculating the mass balance of the evaporation sub-system. In this simulation, the vapour rate, un-evaporated liquid flowrate and mass composition of dry matter in product can be calculated.

- **Compressor sub-system**

The objective of this section is to implement a simulation of the compressor sub-system. The implementation of this system is simpler than the evaporation sub-system because it does not contain any differential equation, rather it contains only a compressor map. The Simulink system of the sub-system (shown in Figure 4.7) provides the calculations of the temperature of compressed vapour T_s (mathematical description as given in Eq.3.35), the power required to compress the vapour P_{comp} (given in Eq.3.33) and the flow rate of the condensed vapour M_{cond} (given in Eq.3.36).

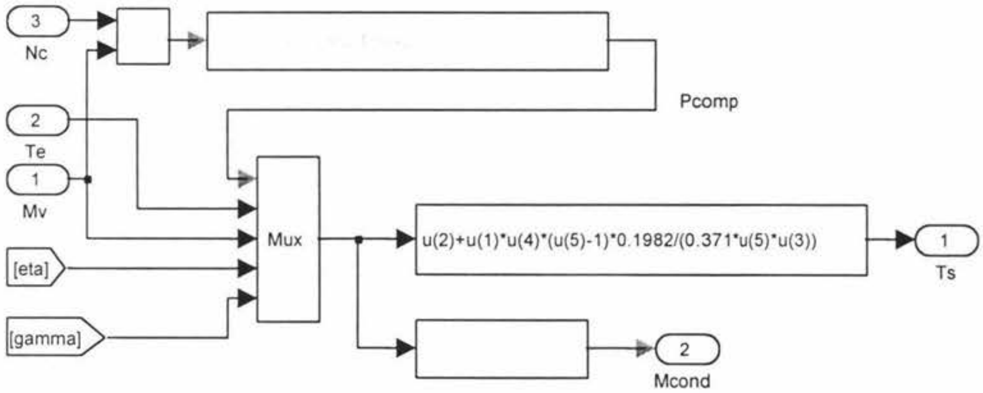


Figure 4.7: Simulink system of compressor sub-system

- **Condenser sub-system**

In this section, we will implement a simulation of the condenser sub-system. A steady-state mathematical description of this sub-system is given in Eq.3.42. $T_{c,in}$ is assumed as a constant.

Figure 4.8 shows the contents of Simulink model for calculating the coolant outlet temperature, $T_{c,out}$.

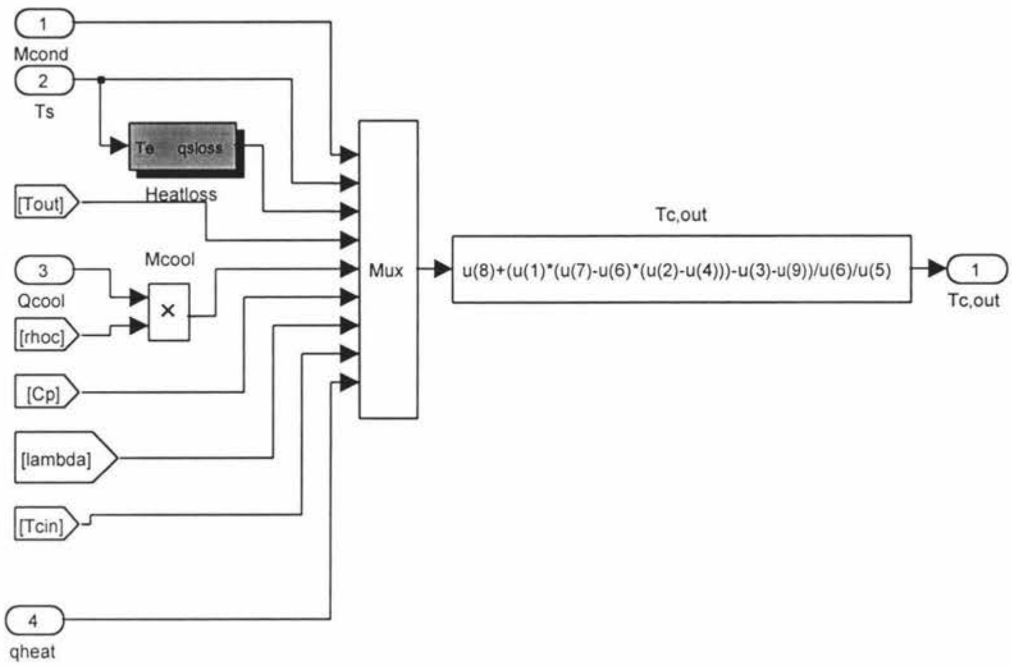


Figure 4.8: Simulink system of condenser sub-system

- **Product sub-system**

In this section, we implemented a simulation of the product sub-system. The mathematical model of a state variable L (given in Eq.3.57) was rearranged as follows:

$$L = \int \left[\frac{M_l}{\rho_p A_l} - \frac{A_d}{A_l} \sqrt{\frac{2(P_e - P_o - \rho_p g y_2) + \rho_p \omega^2 (r - L)^2}{\rho_p \left(4\frac{fl}{d} + 1\right)}} \right] dt \qquad 4.5$$

Where: M_l is given in Eq.3.31.

Finally, the product flowrate was given as follows:

$$Q_p = A_d \sqrt{\frac{2(P_e - P_o - \rho_p g y_2) + \rho_p \omega^2 (r - L)^2}{\rho_p \left(4\frac{fl}{d} + 1\right)}} \qquad 4.6$$

Figure 4.9 shows the contents of Simulink model for the product sub-system. In this simulation, the last output variable--product flowrate M_p was calculated.

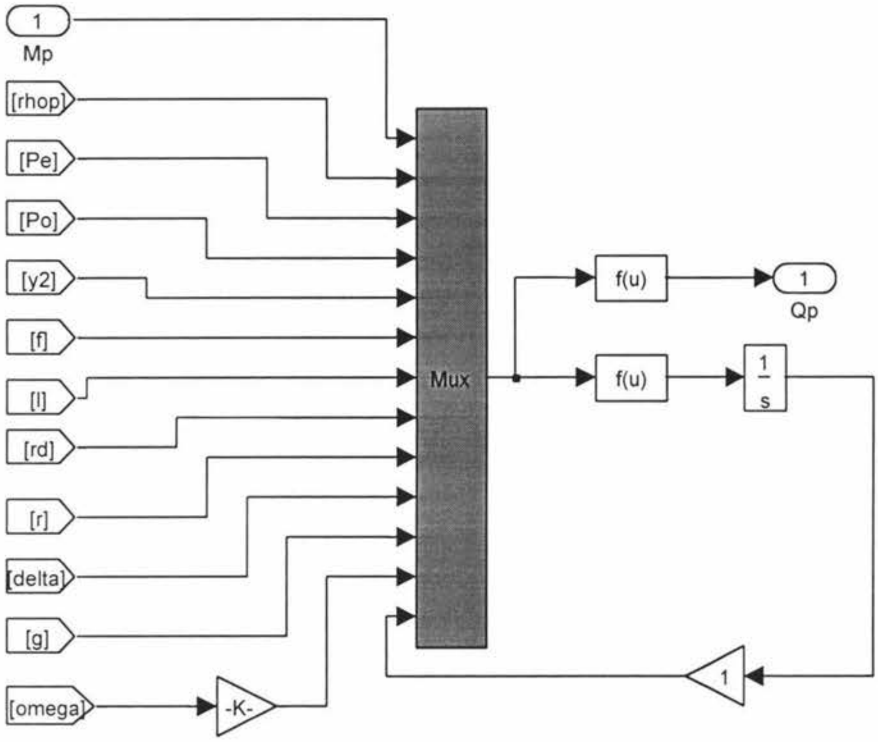


Figure 4.9: Simulink system of product sub-system

- **Spinning cone evaporator system**

Figure 4.10 shows the overall diagram of simulation of the spinning cone evaporator system. The details inside the main blocks have been given above. The constants and evaporator parameters are shown in Appendix 2.

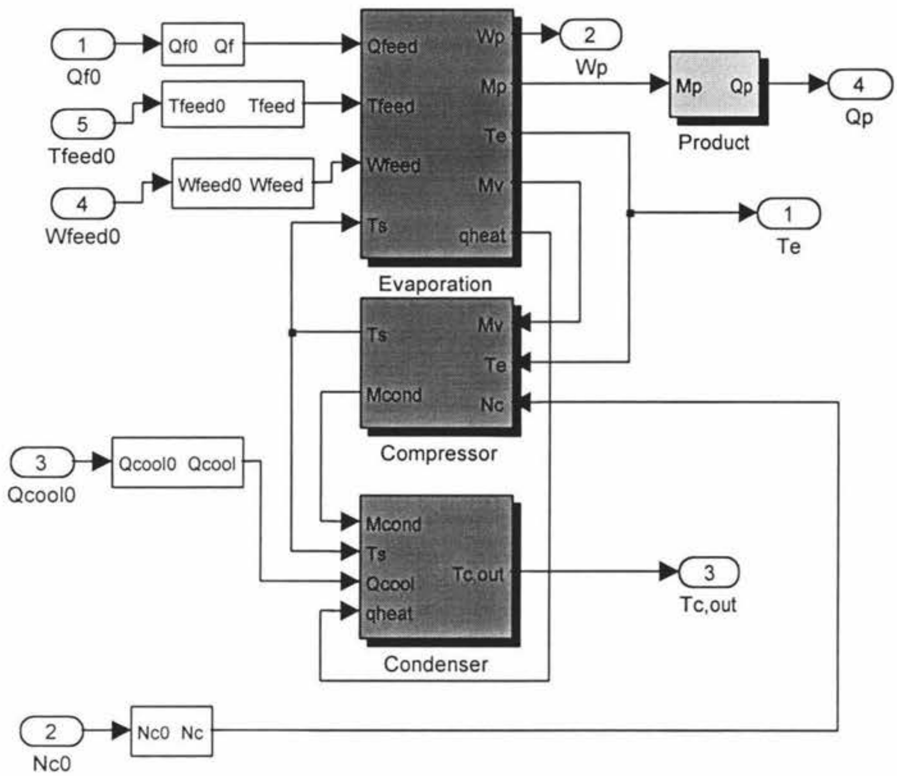


Figure 4.10: Simulink system of spinning cone evaporator

4.3 Model verification

Model verification is an essential activity, which is provided with a simulation computer program that it performs as intended (Law et al., 1991). We have already created and implemented the evaporator system model, but any subsequent analysis is meaningless unless the simulation output of the model can be trusted. The simulation model was rigorously verified as follows.

- Subsections of the simulation model were individually implemented and tested.
- A series of informal structured walkthroughs were used to confirm simulation logic (sequence).
- A person with extensive Simulink programming experience helped with debugging and walkthroughs.
- Simulation output was tested for reasonableness. For every sub-system model, the simulation model outputs were compared to the initial values of the mathematical equations. Results were in agreement with man-calculated data.

Chapter 5 Experimental Method

The objective of the experiments to be undertaken is to collect data for validating the model of the spinning cone evaporator. The data collected include the evaporation temperature, evaporation pressure, feed temperature, feed flowrate, rotating speed of the cone, condensate flowrate, concentrate flowrate, temperature of cooling water, flowrate of cooling water, ambient temperature and steam temperature. The validation includes both steady state and dynamic comparisons.

5.1 Materials

The materials used in this project are as follows:

Water: The water is used to be the feed liquid because its physical properties are well known. Also it is the cooling water for the condenser.

Milk powder: The milk powder is used to make up the milk solutions.

Chemical used for cleaning: A 1.5 % caustic soda solution may be used to clean the heating surface of the cones.

Steam: The steam is used to heat the liquid to be evaporated and pre-heat the feed.

5.2 Equipment

The equipment used in the experiment is described in the following sections:

5.2.1 Spinning cone evaporator system

The project is supported by a pilot-scale spinning cone evaporator system at the College of Science. The spinning cone evaporator system consists of a spinning cone evaporator, a condenser, a vacuum unit, a feed pump, a concentrate pump and a condensate pump. The diagrammatic sketch of the spinning cone evaporator system is shown in Figure 5.1. There is no compressor in this system.

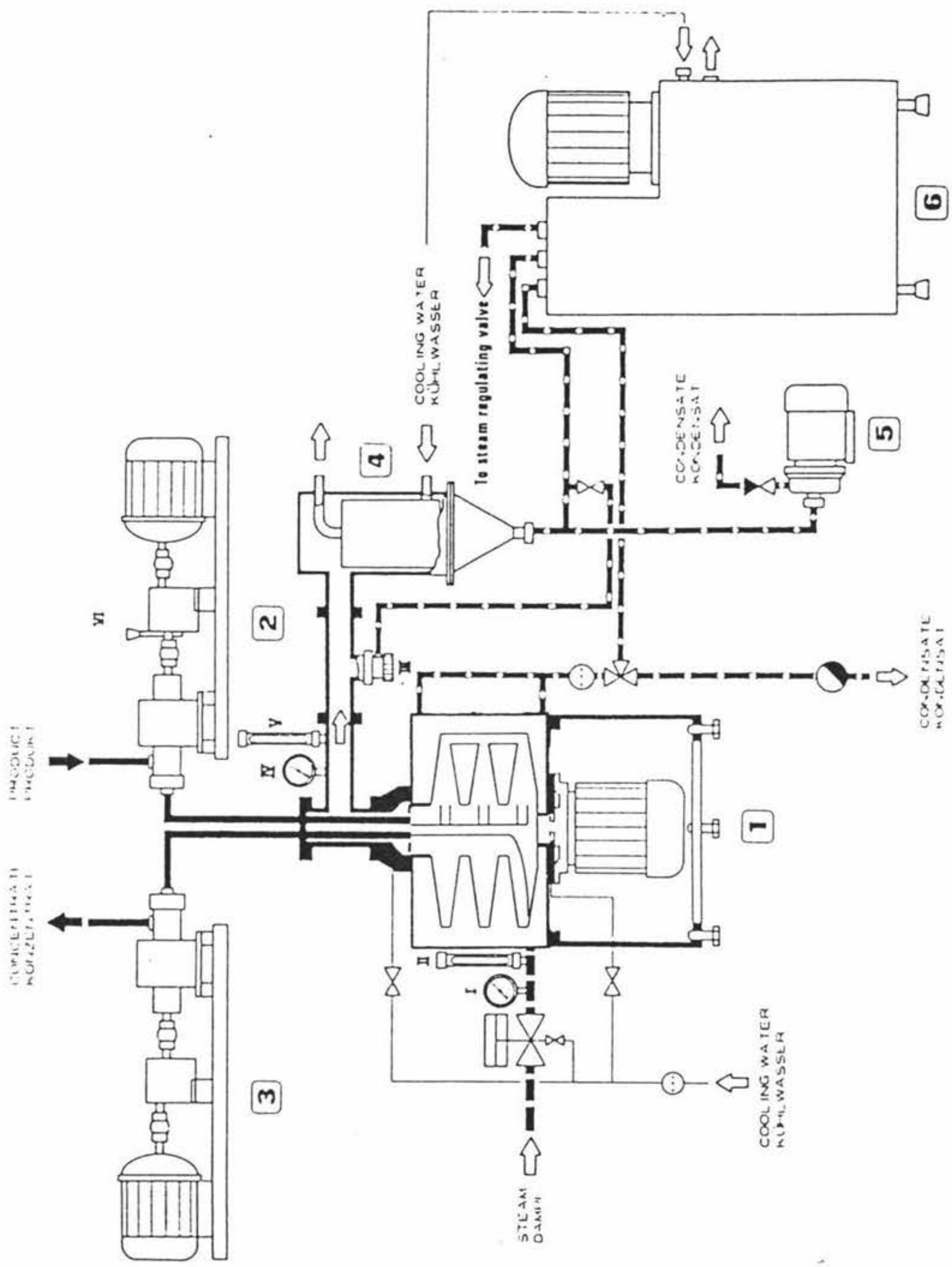


Figure 5.1: Diagrammatic sketch of spinning cone evaporator system

The liquid to be evaporated is pumped through a heat exchanger where it is heated to evaporating temperature, and then into the feed tube. A rotameter is used to measure the feed rate. The temperature of the liquid feed is measured by a thermocouple. After passing through the feed tube, the liquid is fed on to the inside surface of the cone under the centrifugal force caused by the speed of rotation. The unevaporated liquid is pumped out and the vapour passed through the condenser from which the condensate is pumped out. The vapour temperature is measured by a thermocouple. Steam pressure is controlled by means of a steam regulator valve, and measured by a Bourden gauge. Steam temperature is measured by a glass thermometer and a thermocouple. The vacuum is provided by an ejector. The vacuum can be adjusted manually, and measured by a vacuum gauge.

Spinning cone evaporator

The spinning cone evaporator, made up of stainless steel, is the heart of the system. It has a rotor with six 80° cones that is driven by an adjustable speed motor.

The spinning cones have three conical sections, two in the first pass and one in the second. A concentrate tube, a feed tube are connected to the inner surface of the evaporator, and a vapour outlet tube is connected to the top of the evaporator.

Condenser

The condenser (Figure 5.1, ④) is a water-cooled surface condenser with spiral tubes to remove the vapour generated in the spinning cone evaporator. It is cleared by a condensate pump.

Vacuum unit

This unit (Figure 5.1, ⑥) consists of a container for water, a circulation pump and three ejectors with built-in check valves. The minimum pressure is -95 kPa.

Feed pump

The feed pump (Figure 5.1, ②) is a Nemo pump that is positive acting rotary pump, type 1 BD 15.

Concentrate pump

The concentrate pump (Figure 5.1, ③) is same as the feed pump.

Condensate pump

The condensate pump (Figure 5.1, ⑤) is a centrifugal pump, type GM.

5.2.2 Steam regulating valve

The steam regulating valve (type SRV-5) together with a temperature regulator (type VR-5) is intended for regulating the quantity of steam and controlling the temperature of heating steam. Both are manually operated. If the temperature obtained does not correspond with that on the regulator scale, the setting can be adjusted by means of scale adjusting device.

5.2.3 Data logger

A data logger (CR10) is used to record the experimental data (on-line measured parameters: the temperatures of the feed, steam and vapour and the speeds of the feed pump and spinning cone). This is a fully programmable data logger in a small, rugged, sealed module. A personal computer is connected to the data logger's serial I/O port to communicate with the data logger.

5.2.4 Instruments

The temperatures of the feed, steam and vapour are measured by the thermocouples. A vacuum gauge is used to monitor the vapour pressure. Two steam pressure gauges are used to monitor steam line pressures. The pressure of steam is controlled by means of a steam regulator. The inlet feed flow needs to be measured, as does the rotating speed of the spinning cone. Because variable speed drives are used for the motors, the speed can be measured and controlled very easily. Feed flow can then be calculated by calibration.

A video recorder is used to record the weight of product on a scale in order to get a dynamic flowrate of product.

Other instruments used include a weigh scale, a refractometer and a stopwatch.

5.2.5 Compressor

In the final evaporator design, a fan compressor will be used to recompress the evaporated vapour. The recompressed vapour will be used as the steam to heat the evaporator again. There was no compressor in this experimental setup so the fan compressor portion of the model could not be validated.

5.3 Variables and their ranges

The variables selected for the experiment are as follows:

- Evaporating pressure
- Feed temperature
- Feed flowrate
- Steam temperature
- Condensate flowrate
- Product flowrate
- Flowrate of cooling water
- Dry matter compositions of feed and product
- Inlet and outlet temperatures of cooling water
- Cone speed

Seven parameters were measured using manual methods:

- Condensate flow rate
- Concentrate flow rate
- Flow rate of cooling water
- Dry matter compositions of feed and product
- Inlet and outlet temperatures of cooling water

Six parameters are selected as the input variables:

- Feed temperature
- Feed flow rate
- Evaporating pressure
- Steam temperature
- Flow rate of cooling water

- Dry matter compositions of feed

Three parameters are considered as the output variables:

- Outlet temperatures of cooling water
- Product flowrate
- Dry matter compositions of product

Two parameters are considered as constants:

- Inlet temperature of cooling water
- Ambient temperature

In order to collect accurate data from the plant and operate the plant under good conditions, the pilot-scale spinning cone evaporator needed to be assembled carefully, leakage of the vacuum system needed to be checked for, all the instruments and regulators used needed to be calibrated.

The ranges of the selected variables are listed in Table 5.1.

Table 5.1: Ranges of Selected variables

variables	Range	
Feed flow rate	8—26	g/s
Feed temperature	75—90	°C
Feed dry matter composition	11—15	%
Cone speed	0—210	rad/s
Evaporating temperature	40—70	°C
Evaporation pressure	-74— -62	kPa
Steam temperature	80—90	°C

5.4 Experimental procedure

In order to find the effect of individual parameters on the model, only one parameter was varied during each run, while other parameters were kept constant.

The experimental procedure was as follows:

The evaporator was started with water, and the steam temperature, evaporating temperature, feed flowrate, feed temperature, rotating speed of the cone and flowrate of cooling water were set at the selected values.

After steady state was reached, the feed was then switched to the milk solution. The necessary adjustments were made to maintain the variables at desired values.

The steady state flowrates of vapour condensate and concentrate were measured using a bucket and stopwatch method. Flow rates were calculated by using the values collected during a three minute period at two minute intervals during each test run. That means that flows of condensate and concentrate were diverted to-and-from an empty bucket and the time was recorded. After three minutes, the amount of condensate and concentrate in the buckets were measured. For each run, two sets of measurements were made. If the error between the each measurement was larger than about 5%, further measurements were made.

The dry matter fractions of the feed and product were measured by a refractometer.

The dynamic flowrate of concentrate (product) was measured by using a video recorder. Flow rates were calculated by using the weight values of the concentrate in buckets, which were read off at two second intervals from the video recorder at a later time.

In order to synchronise the datalogger time and video recorder time with real time, the time was recorded when the step change was made in every input variable.

During the experiment, all other data were recorded on a log sheet.

After a test was completed, the feed was switched to water to flush out the remaining solution in the system, and then cleaning was done.

Initially, the evaporating temperature and the feed temperature were very difficult to control, and the vacuum of the evaporator was difficult to keep constant. After the necessary adjustments were done, the steady state was reached. The concentrate and the vapour condensate were fed back to the feed tank after their weights were measured. A mass balance of the feed, concentrate and condensate was carried out at the beginning of each run to check proper for vapour condensation and accurate measurement.

The raw experimental data are given in Appendix 3.

5.5 Data processing

The overall heat transfer coefficient (U_{exp}) was computed using the vapour condensate flow rate and overall temperature difference (i.e. the temperature difference between the steam temperature and evaporating temperature) measurements. U_{exp} was calculated from the following formula:

$$U_{exp} = \frac{W_{exp}\lambda}{A\,\Delta T} \qquad 5.1$$

Where: W_{exp} = vapour condensate flow rate

λ = latent heat of vapour condensation

A = heat surface area

ΔT = temperature difference

Chapter 6 Model Validation

The aim of the model validation is to determine whether the model is an adequate representation of the physical process. In this chapter the model validation is presented for both steady state and dynamic comparisons.

6.1 Validation process

One of the purposes of generating model behaviour is to compare it with the behaviour of the real system. This critical comparison is called the model validation. In other words, the model validation is about model testing with experiments; model validation includes not just only comparison of modelling results with data from selected experiments, but also evaluation of procedures for the construction of conceptual models and calculational models as well as methodologies for studying data and parameter correlation (Pescatore, 1994).

The spinning cone evaporator system model was validated against data collected from the prototype spinning cone evaporator system at Massey University. This evaporator consists of a single-shell evaporator with three conical sections (as shown in Figure 3.11), two in the first pass and one in the second. The evaporator includes a condenser for the vapours and a product transport section (as shown in Figure 1.8 and 5.1). Steam is used to provide heating on the shell side so there is no compressor. As the experimental rig did not have a compressor, the compressor equations could not be validated.

The validation process is as follows:

1. Agree on the validation process before the study begins:
   Before this model was developed for the spinning cone evaporator system, a model validation process was agreed upon, which involved applying the same set of inputs to the given prototype spinning cone evaporator system and to the simulation model and comparing the results.
2. Identify the assumptions underlying the model where possible:

It is impossible to obtain a perfect model without error. As discussed at the beginning of Chapter 3 the model is primarily required for use as a simulation tool for the development and testing of a pilot-scale spinning cone evaporator. The model should correctly represent the dynamic behaviour of the evaporator system. Therefore a small amount of error within the model may be acceptable as long as the model estimates follow the dynamics of the system and remain within the vicinity of the actual values.

The evaporator model includes four sub-models. These sub-models were derived with many assumptions and many parameters are assumed to be fixed. For example, the assumptions include that potential and kinetic terms are negligible; liquid/metal internal energy is equivalent to enthalpy, enthalpy is independent of pressure and heat capacity is constant. The parameters considered fixed (for a run) are the ambient temperature, evaporation pressure, speed of spinning cone and inlet temperature of cooling water.

3. Perform face validity on the conceptual model. That means the conceptual model has been thoroughly discussed for all the elements and procedures (flowcharts) with knowledgeable people.
4. Explore the model behaviour. That means without looking at data, take the model and see how it behaves for typical or extreme cases. The results of the model have been evaluated and found they look reasonable and are internally consistent.
5. A predictive study and comparison with data have been carried out in two parts: steady state validation and dynamic validation. That is very helpful to build confidence in the model.
6. Plan on periodic review and possible revalidation as more information will be collected after the dry matter compositions of the feed and product can be measured on-line.

6.2 Results

To investigate the validity of the spinning cone evaporator model, simulation results were compared to experimental data. Several sets of operating conditions were used for the model simulation and experiment as shown in Table 6.1.

Table 6.1: Operating conditions for model simulation and experiment

		run 1	run 2	run 3
Cone speed	rpm	1500	1700	2000
Evaporation pressure	kPa	-78	-72	-65
Ambient temperature	°C	25	25	25
Inlet temperature of cooling water	°C	18.25	18.25	18.25

In this dynamic spinning cone evaporator model, the input variables need to be selected over a suitable range. Table 6.2 shows the selected ranges of inputs variables used in the model validation.

Table 6.2: Selected ranges of input variables

Input variables	Range	
Feed flow rate	8—26	g/s
Feed temperature	75—90	°C
Feed dry matter composition	11—15	%
Steam temperature	80—90	°C

The inputs that can be changed on the experiment are the feed flowrate, feed temperatures, feed dry matter composition, steam temperatures, and cooling water flowrate. The measurable outputs are the evaporation temperature, outlet temperature of cooling water, product dry matter composition and product flowrate. The steady state experimental data was initially used to set the model parameters. The some model parameters were further modified by comparing the model data with the experimental data when the operating condition was changed. A repeated procedure of running the simulation, comparing the model results with the experimental data and modifying the model parameters was carried out until an adequate representation and suitable accuracy were achieved.

For the steady state validation a series of experiments were carried out on the evaporator at different operating conditions. Six runs were performed on water and six on reconstituted whole milk. For each of these, the evaporation temperatures, outlet temperature of cooling water, product flowrate and dry matter fraction were measured. These parameters were then calculated using the dynamic model at the same operating conditions (shown in Appendix 4). The combined results of this can be seen in Table 6.3. Note that the mean error for the dry matter fraction was calculated for the six milk runs only.

Table 6.3: Steady state validation results

VARIABLE	MEAN ERROR (%)	MAXIMUM ERROR (%)
Evaporation Temperature	0.315	0.57
Product flow	3.99	9.06
Outlet temperature of cooling water	0.58	0.83
Dry matter fraction	4.58	9.66

The experimentation showed that there was no effect on the evaporation temperature in the case of feed flow changes, feed temperature changes, steam temperature changes and cone speed changes. The model confirmed this.

In order to evaluate the dynamic prediction accuracy of the model, the experiments were carried out on the evaporator for the step changes of three input parameters and performed on water. Three input parameters are the feed flow, feed temperature, steam temperature. The step changes were made by the manual acting. The product flowrates were measured in the case of feed flow step change, feed temperature step change, steam temperature step change. And the product flowrates were then calculated using the system model in the case of feed flow step change, feed temperature step change, steam temperature step change. The results can be seen in Fig 6.1—6.3.

The model predicted there were the consistent changes in the case of feed flowrate changes, feed temperature changes and steam temperature changes. The responses of the concentrate flowrate to a step change in feed flowrate are shown in figure 6.1 (predicted and experimental).

The responses of the concentrate flowrate to a step change in feed temperature are shown in figures 6.2 (predicted and experimental).

The responses of the concentrate flowrate to a step change in steam temperature are shown in figures 6.3 (predicted and experimental).

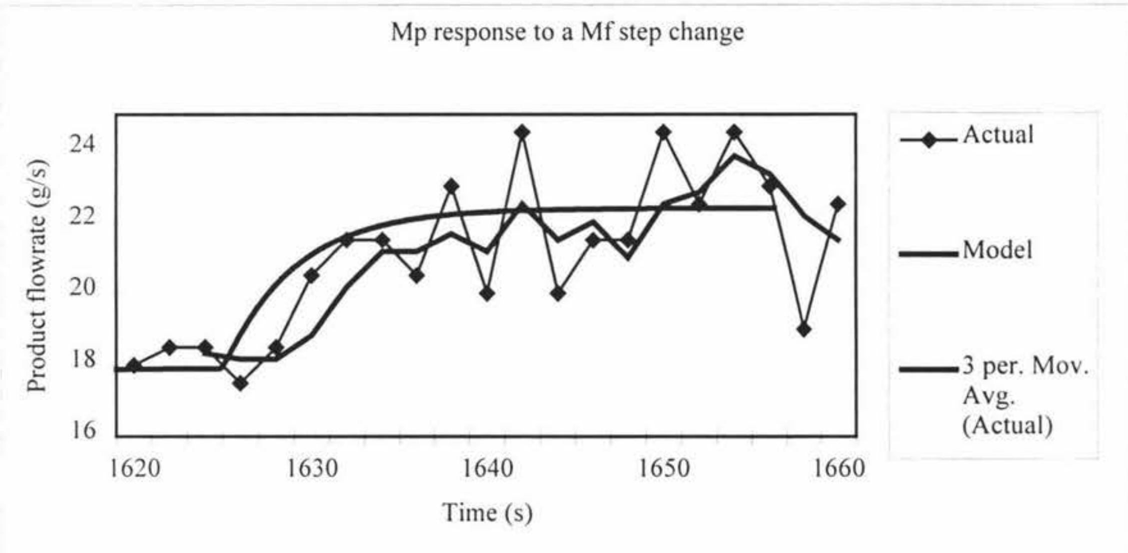


Figure 6.1 : Product flowrate to a step change in feed flowrate

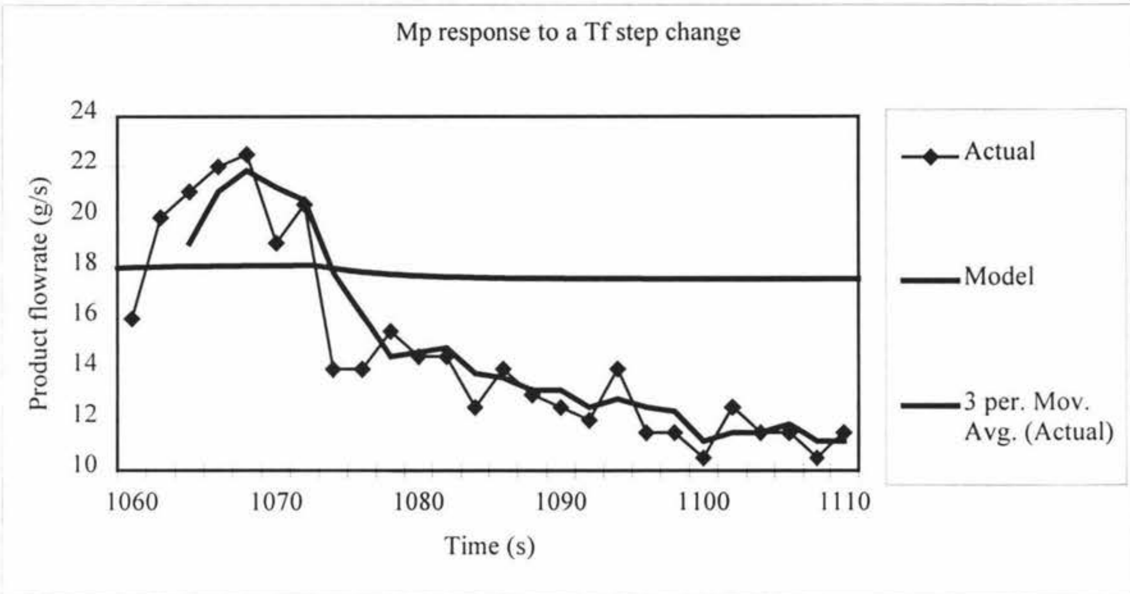


Figure 6.2 : Product flowrate to a step change in feed temperature

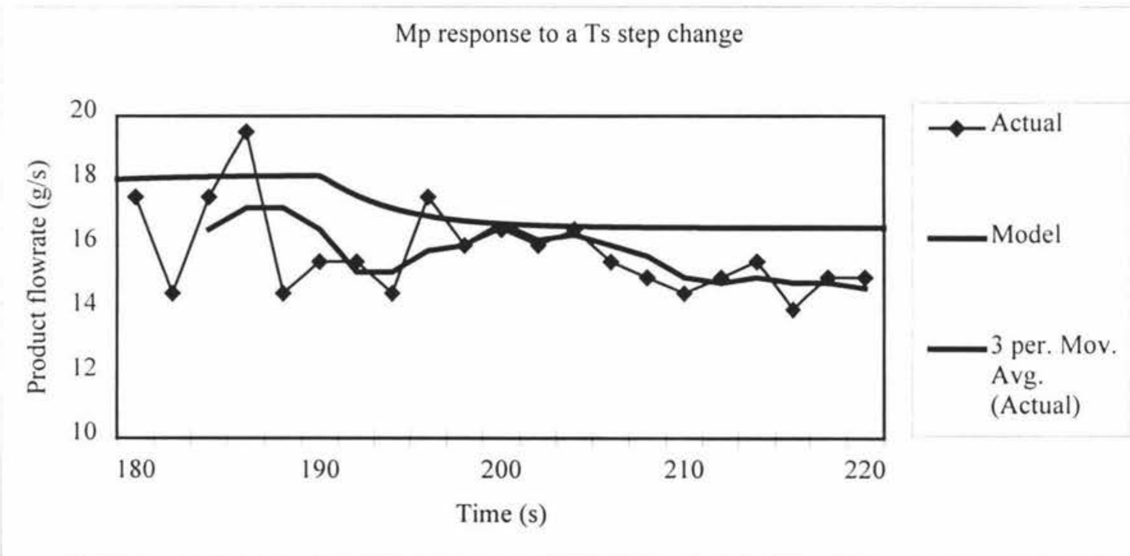


Figure 6.3 : Product flowrate to a step change in steam temperature

6.3 Discussion

From the results of the steady state comparisons in Table 6.3 it can be seen that the errors for the evaporation temperature and outlet temperature of cooling water are very small. While the errors for the product flowrate and product dry matter fraction were larger they were also satisfactorily small. It therefore seems that the model is able to predict the steady state values with acceptable accuracy. From this we can state with confidence that the model has captured the essential characteristics of the experimental evaporator.

As can be seen by the dynamical data in Figure 6.1 the responses of the actual evaporator to a step change in feed flowrate was close to that predicted by the model. The model predicts a response with a time constant of about 8s, and the experimental evaporator appears to have a time constant of about 8s. The magnitudes of product flowrate have been captured in the model. It can be seen that the results predicted for a step change in feed flowrate are in good agreement with experimental data.

From the dynamical data in Figure 6.2 the response of the product flowrate to a step change in feed temperature is slower than that predicted by the model. The model predicts a response with a time constant of about 8s, while the experimental evaporator appears to have a time constant of about 20s. The reason of this difference may be that the step change in feed temperature was mad by a change to the steam-water mixer that controls the feed temperature. There would be considerable thermal mass in the heat exchanger that would slow the response to this step change. The magnitudes of product flowrate have not been captured in the model. It shows that the predicted step change in feed temperature is much smaller than shown by the experimental data. One possible explanation for this discrepancy may be that there would be more flash, and carry over of droplets of feed from the bubbles into the vapour and hence into the condensate when the feed temperature was increased. This would reduce the concentrate flow and account for the much greater apparent drop in the product flowrate. This portion of the model will need to be revisited before the model can be used for simulation.

From the Figure 6.3 the response of the product flowrate to a step change in steam temperature is slower than that predicted by the model. The model predicts a response with a time constant of about 8s, while the experimental evaporator appears to have a time constant of about 10s. The magnitudes of product flowrate change are very close in the model and experiment. The absolute errors for the product flowrate were 10 % but they were also satisfactorily small. It can be also seen that the results predicted for a step change in feed flowrate are in reasonable agreement with experimental data. At times we have had buildup of steam condensate in the bottom of the evaporator casing.

If this occurred the thermal mass of the evaporator would increase, and this may account for the slight difference in time constant.

For the responses of evaporation temperature, the experiment results show that there were no changes on the evaporation temperature in the case of feed flow changes, feed temperature changes, and steam temperature changes. The results predicted in model are in good agreement with experiment.

In the spinning cone evaporator model, the product flowrate, evaporation temperature can be predicted at a given time, and the outlet temperature of cooling water, product dry matter composition can also be presented at a steady state. It can be seen that the results predicted using this spinning cone evaporator model, which accounts for varying concentrate flowrate, evaporation temperature in time, are in good agreement with experimental data.

This model provides a valuable tool to predict performance in a spinning cone evaporator and to modify the design parameters.

Chapter 7 Conclusions

This thesis presents a model of a spinning cone evaporator based on sub-system modelling approaches. The issues we aimed to address include:

1. Developing the dynamic spinning cone evaporator model, which reflects variation of the output variables in time.
2. Implementing an integrated model using Simulink to describe the dynamic behaviour of a spinning cone evaporator.
3. Validating the model to determine whether the model is an adequate representation of the physical process.

Several conclusions can be drawn as a result of this thesis. They are:

1. A dynamic analytical model of a spinning cone evaporator has been derived and presented for a system involving a mechanical compressor, which describes the dynamic relationships between the input variables (cooling water flowrate, M_c, speed of compressor, N_{comp}, pressure of evaporating vapour, P_e, feed flowrate, M_f, feed temperature, T_f and mass composition of feed dry matter, w_f) and the output variables (outlet temperature of cooling water, T_{co}, evaporating temperature, T_e, mass composition of product dry matter, w_p and product flowrate, M_p).
2. The model has been implemented in Simulink, which simulates the model of the spinning cone evaporator system and predicts the behaviour of the system. Simulation output is tested for reasonableness.
3. The model has been validated against an experimental evaporator without a compressor. Validation was carried for both steady state and dynamic conditions using both pure water and milk feeds.
4. The steady state experiments validated the model to an acceptable level of accuracy.
5. The dynamic results of the model have been evaluated and are consistent with the results of the experiments except for the product flowrate response to a step change in feed temperature.

6. The evaporation temperature was not affected by the input variables: feed temperature, feed flowrate, steam temperature.
7. The accuracy of the model in predicting the behaviour (steady state and dynamic) of the compressor sub-system can not be determined until validation with an evaporator which includes these items.
8. Further validation experiments are required for the dynamic response of the condensate flowrate and total dry mass fraction.

The recommendations for any further work are:

1. To complete validation of the model, e.g. in predicting the outlet temperature of cooling water and total dry mass fraction in time, the compressor sub-system needs to be validated after a compressor is added for testing.

2. To investigate the effect of the mass flowrate of flashed vapour to a step change in feed temperature in order to see if this can explain the discrepancy in the dynamic prediction of product flowrate.

NOMENCLATURE

a	fitting coefficient for compressor map ($kgm^{-1}s^{-1}rpm^{-2}$)
A	total heat transfer area (m^2)
A_d	cross-sectional area of the pipe (m^2)
A_e	heat transfer area of the cone (m^2)
A_{el}	surface area for losses (m^2)
A_l	area of the surface of the non-evaporated liquid (m^2)
b	fitting coefficient for compressor map ($kgm^{-4}rpm^{-1}$)
c	fitting coefficient for compressor map ($kgm^{-7}s^{1}$)
c_p	heat capacity of milk ($Jkg^{-1}K^{-1}$)
c_{pm}	heat capacity of the milk ($Jkg^{-1}K^{-1}$)
c_{pc}	heat capacity of the cooling water ($Jkg^{-1}K^{-1}$)
d	diameter of pipe (m)
D	tube diameter (usually internal) (m)
f	Fanning friction factor
F	mass flowrate of feed streams (kg/s)
F_c	centrifugal force on a small element of liquid (N/m^3)
F_g	gravity force on a small element of liquid (N/m^3)
F_r	force on a small element of liquid at r coordinate (N/m^3)
F_z	force on a small element of liquid at z coordinate (N/m^3)
g	acceleration due to gravity (m/s^2)
h	thickness of liquid (m)
h_{el}	overall heat transfer coefficient for losses heat flow ($Wm^{-2}\,K^{-1}$)
h'_l	liquid side heat transfer coefficient ($Wm^{-2}\,K^{-1}$)
h_s	steam side heat transfer coefficient ($Wm^{-2}\,K^{-1}$)
H	enthalpy (Jkg^{-1})
H_F	feed enthalpy (kJ/kg)
H_L	liquid enthalpy (kJ/kg)
H_V	vapour enthalpy (kJ/kg)
Ja	Jacob number

k_l	thermal conductivity of liquid ($Wm^{-1} K^{-1}$)
k_w	thermal conductivity of wall ($Wm^{-1} K^{-1}$)
l	length of pipe (m)
L	height of the non-evaporated liquid (m)
M	mass of vapour from evaporation (kg)
M_c	flow of the cooling water (kgs^{-1})
M_{cond}	mass flow through the condenser (kgs^{-1})
M_e	thermal mass (JK^{-1})
M_{evap}	evaporation rate (kgs^{-1})
M_f	feed flow of milk (kgs^{-1})
M_{flash}	flash rate (kgs^{-1})
M_l	mass flowrate of the non-evaporated liquid (kgs^{-1})
M_m	molecular weight of gas (kg/kmol)
M_{ms}	steam molecular weight (kg/kmol)
M_{mv}	vapour molecular weight (kg/kmol)
M_p	flow of milk from evaporator (kgs^{-1})
M_v	vapour rate (kgs^{-1})
M_v'	vapour removal rate (kgs^{-1})
M_{water}	flow rate of the sprayed water (kgs^{-1})
m	the mass of gas (kg)
n	cone number
N	number of tubes
N_{comp}	compressor speed (rpm)
P	pressure (kPa)
P_c	pressure produced by the centrifugal force (kPa)
P_e	evaporating pressure (kPa)
P_I	pressure of the input point (kPa)
P_o	pressure of the discharge point (kPa)
P_{pr}	pressure produced by the rotating liquid (kPa)
Pr	Prandtl number
P_{comp}	power supply to the compressor (W)
P_s	steam pressure of the steam side (kPa)
P_v	vapour pressure of the effect side (kPa)
q	overall rate of heat transfer, W

q_{feed}	net enthalpy from the feed (W)
q_{heat}	heat flow through the evaporator cone surface (W)
q_{loss}	heat loss from the evaporator surfaces (W)
q_v	latent enthalpy of the evaporating vapour (W)
Q	total inward net heat flow (W)
Q_p	product flowrate (m^3/s)
Q_v	flowrate of the compressed vapour (m^3/s)
r	radius of the cone (m)
r_d	radius of the pipe (m)
r_I	inner radius at the top of cone (m)
R	universal gas constant (J/kmol K)
R_I	inner radius at the bottom of cone (m)
S	steam usage (kg/s)
T_a	ambient temperature (K)
$T_{c,in}$	inlet temperature of cooling water (K)
$T_{c,out}$	outlet temperature of cooling water (K)
T_e	evaporating temperature (K)
T_f	feed temperature (K)
T_s	steam temperature (K)
T_{wl}	surface temperature of cone on liquid side (K)
T_{ws}	surface temperature of cone on steam side (K)
U	overall heat transfer coefficient ($Wm^{-2} K^{-1}$)
V	mass flowrate of vapour streams (kg/s)
v	fluid velocity in the pipe (m/s)
v_1	flow velocity at the input point (m/s)
v_2	velocity at the discharge point (m/s)
v_c	tangential velocity of the liquid (m/s)
w_f	feed dry mass fraction ($kgkg^{-1}$)
w_p	product dry mass fraction ($kgkg^{-1}$)
W	liquid mass inside evaporator (kg)
W_{exp}	vapour condensate flow rate (kgs^{-1})
W_s	additional work (W)
y_2	height of the discharge point (m)
λ	latent heat of vaporisation (Jkg^{-1})

λ_s	latent heat of evaporation/condensation of steam (Jkg^{-1})
λ'_s	effective latent heat (Jkg^{-1})
γ	ratio of specific heat
ρ	density (kgm^{-2})
ρ_p	discharge density (kgm^{-2})
ρ_v	vapour density (kgm^{-2})
η	compressor efficiency (%)
ν	kinematic viscosity (m^2s^{-1})
ω	angular velocity of rotation (rad/s)
θ	half angle of the cone (rad)
δ	height of the cone (m)
δ_w	thickness of wall (m)
ζ	correlation coefficient
ΔP_{fric}	pressure drop due to the friction in pipe line (kPa)
ΔP_{exit}	pressure drop across the discharge nozzle (kPa)
ΔT	temperature difference (K)

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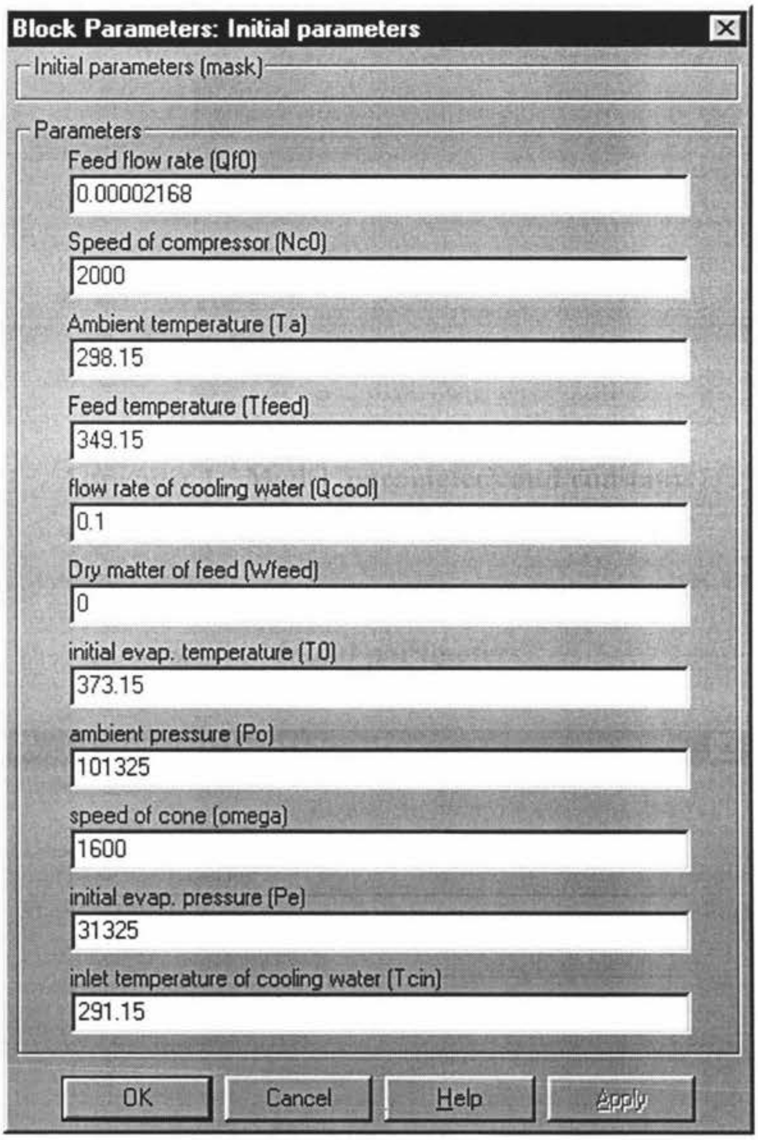
APPENDICES

Appendix 1: Model parameters and constants

1. Initial parameters



Block Parameters: Initial parameters

Initial parameters (mask)

Parameters

Feed flow rate (Qf0)
0.00002168

Speed of compressor (Nc0)
2000

Ambient temperature (Ta)
298.15

Feed temperature (Tfeed)
349.15

flow rate of cooling water (Qcool)
0.1

Dry matter of feed (Wfeed)
0

initial evap. temperature (T0)
373.15

ambient pressure (Po)
101325

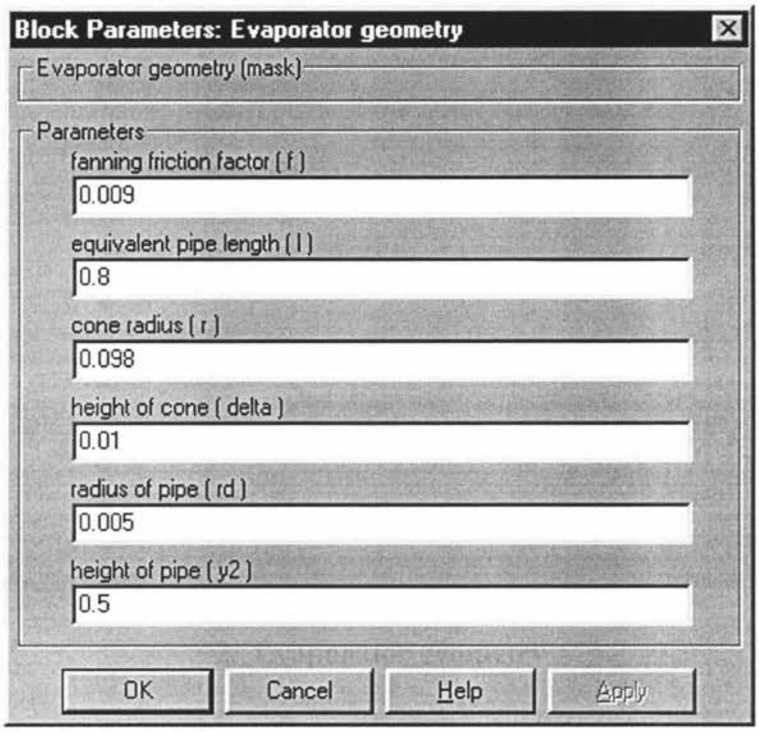
speed of cone (omega)
1600

initial evap. pressure (Pe)
31325

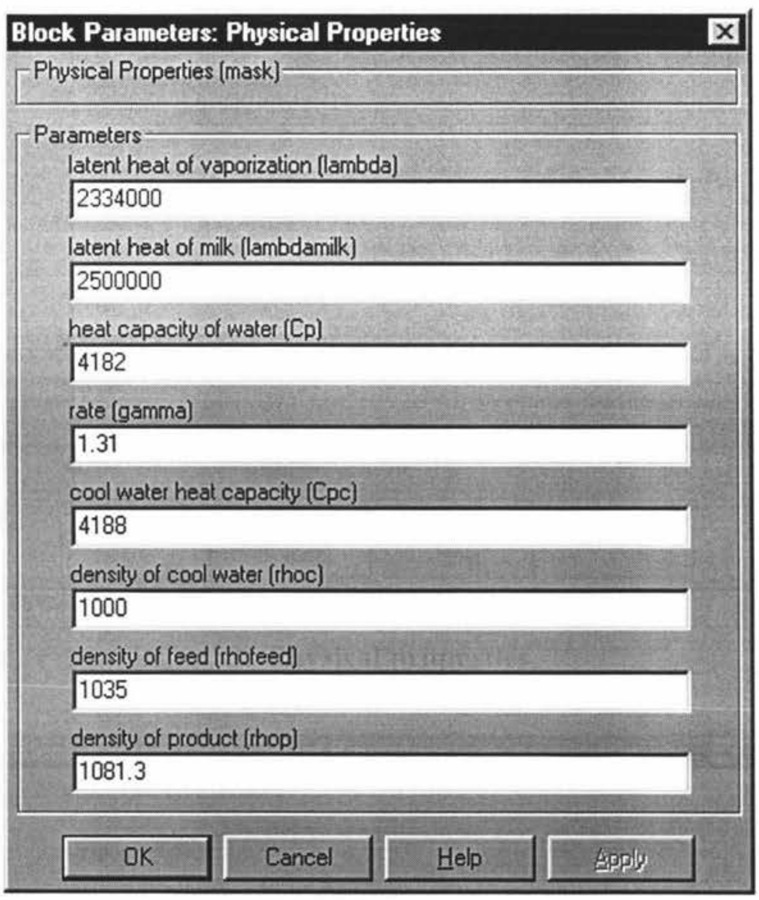
inlet temperature of cooling water (Tcin)
291.15

OK | Cancel | Help | Apply

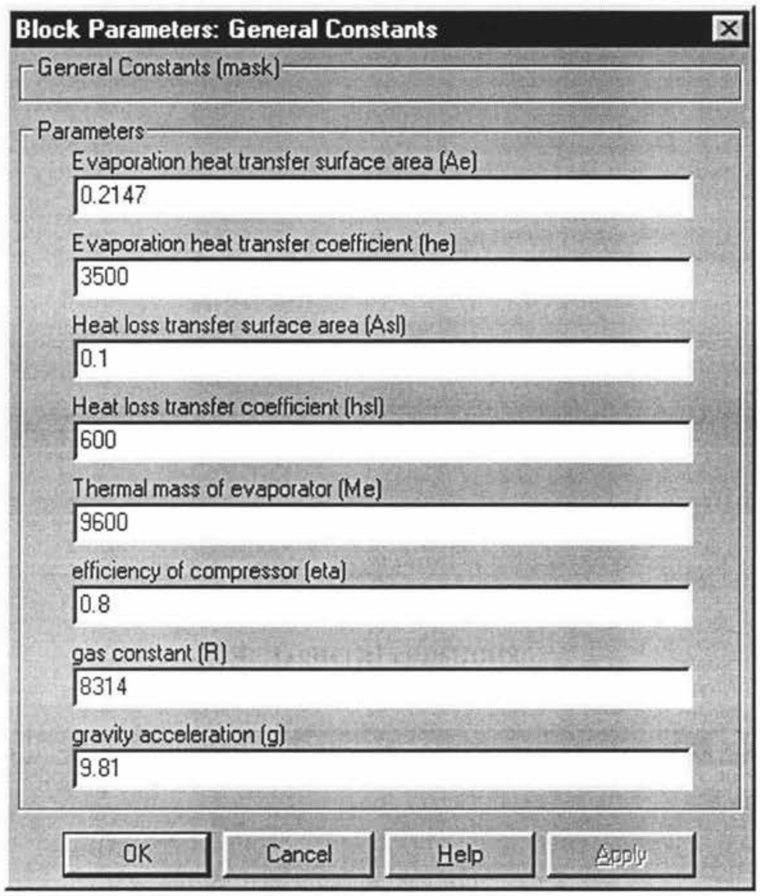
2. Evaporator geometry



3. Physical properties

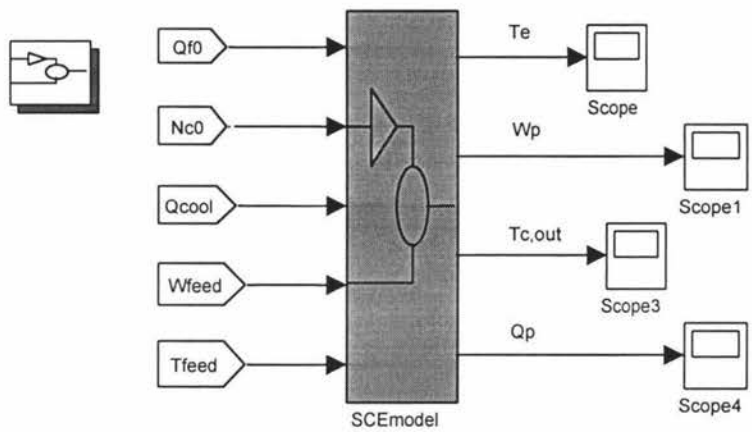


4. General constants

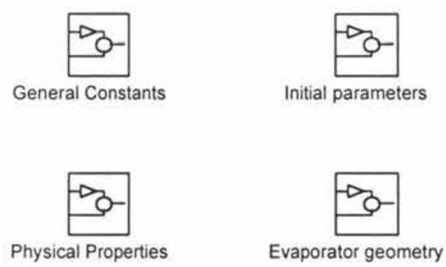


Appendix 2: Simulink systems of spinning cone evaporator

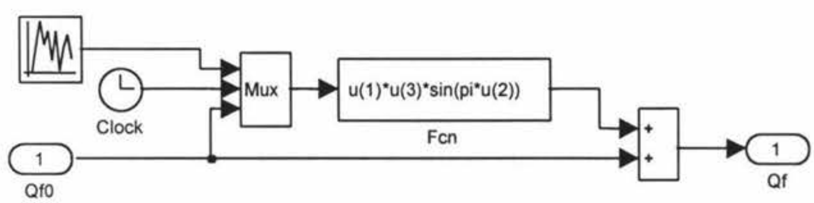
1. Overall model and constants



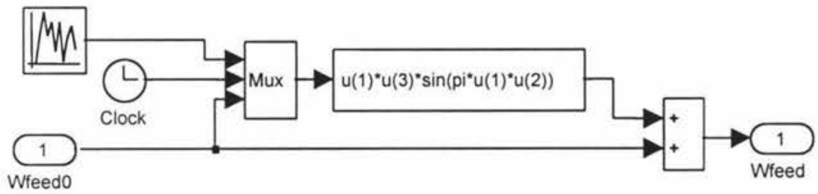
2. Initial parameters and constants of model



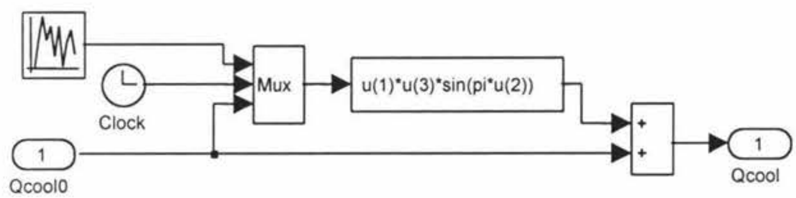
3. feed flowrate



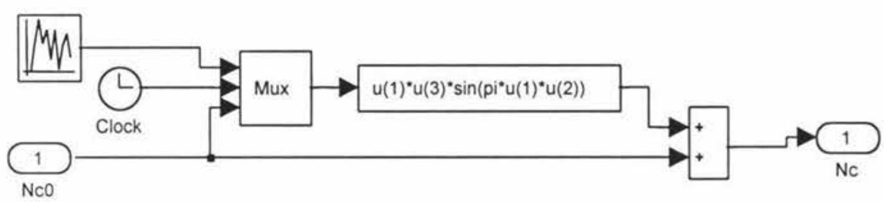
4. Dry matter composition of feed



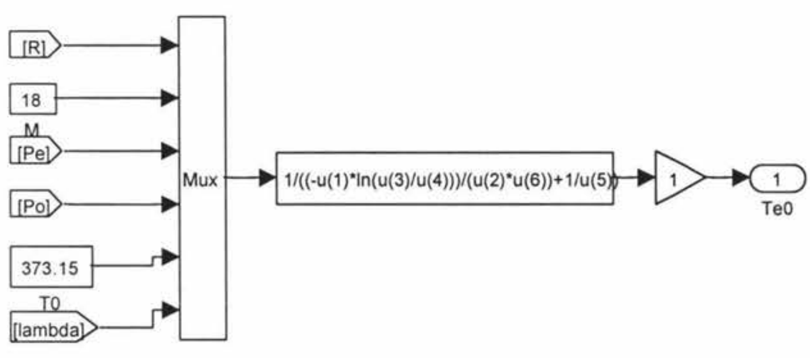
5. Flowrate of cooling water



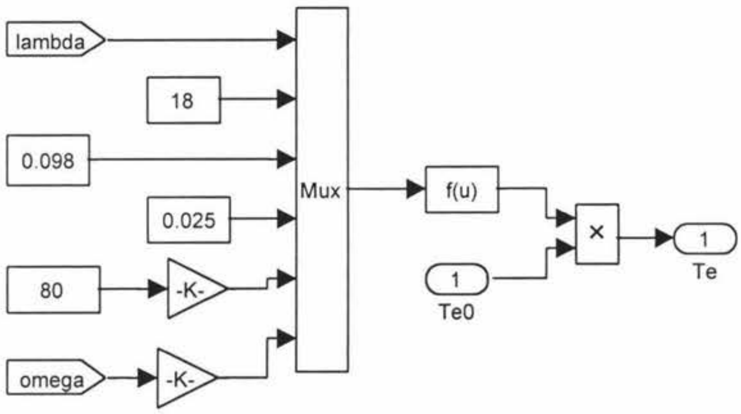
6. Speed of compressor



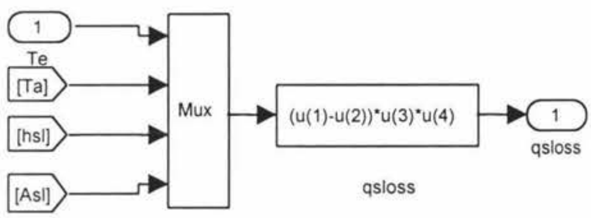
7. Initial evaporation temperature



8. Evaporation temperature



9. Heat loss

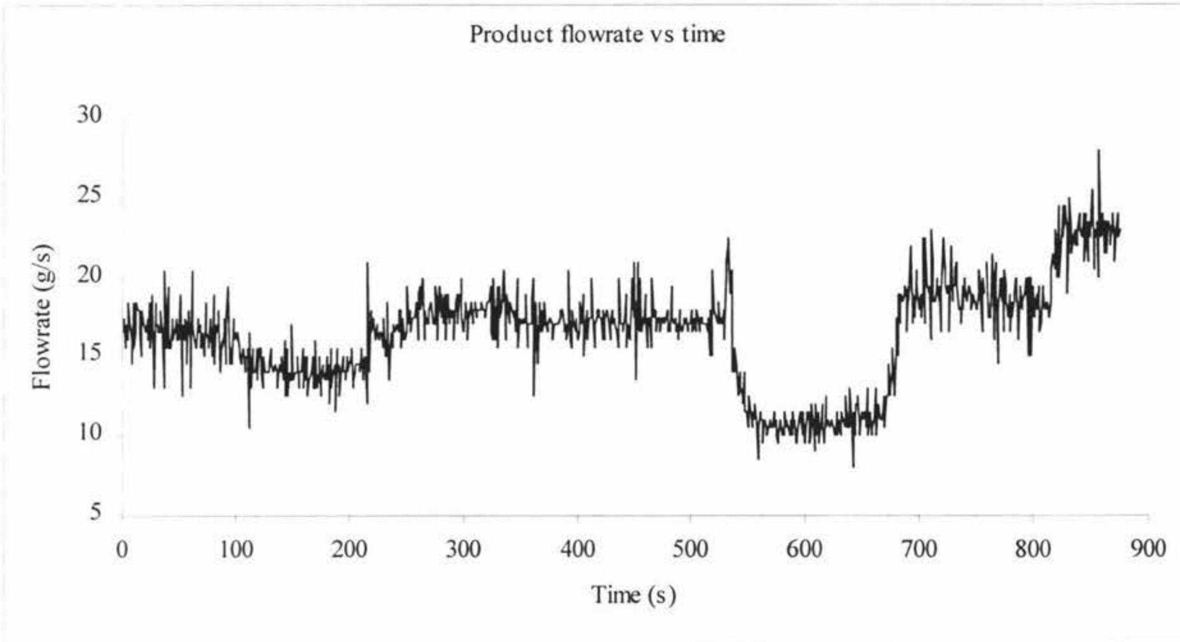


Appendix 3: Experimental data

1. Experimental data record sheet

Trial	T_f (°C)	T_s (°C)	T_e (°C)	T_{cout} (°C)	M_{conc} (Kg/s)	M_{cond} (Kg/s)	M_f (Kg/s)	P_e kPa	SP (rpm)	w_f (%)	w_p (%)
Water											
1	76	92	72	21.5	0.0118	0.0031	0.0149	-70	2000		
2	88	86	70	21.5	0.00695	0.0026	0.0094	-74	1600		
3	87	86	70	21.5	0.0058	0.0023	0.00815	-74	1600		
4	87	86	70	21.5	0.00495	0.0023	0.0072	-74	1600		
5	76	86	70	21.5	0.00485	0.0023	0.0072	-74	1600		
6	92	86	70	21.5	0.0048	0.0025	0.0072	-74	1600		
Milk											
7	80	87	70	21.5	0.0069	0.0038	0.0107	-72	1900	0.133	0.185
8	80	88	69	21.5	0.0070	0.0039	0.0109	-72	2000	0.135	0.178
9	78	90	69	21.5	0.0067	0.0035	0.0105	-72	1500	0.131	0.188
10	78	86	76	21.5	0.0084	0.0016	0.0100	-62	1500	0.119	0.145
11	78	87	76	21.5	0.0058	0.0020	0.0078	-62	1700	0.119	0.136
12	77	87	76	21.5	0.0069	0.0016	0.0085	-62	1500	0.119	0.144

2. Dynamic experimental data



Appendix 4: Steady state results of model

1. Trial 1

Material: Water

Output results

Output variable	Evaporation temperature T_e °K	Product flow rate $Q_p \times 10^{-5} m^3/s$	Cooling water outlet temperature T_{cout} °K
Experiment	345.15	1.18	294.65
Model	344.62	1.202	293.26
Deviation	0.15 %	1.86 %	0.47 %

2. Trial 2

Material: Water

Output results

Output variable	Evaporation temperature T_e °K	Product flow rate $Q_p \times 10^{-5} m^3/s$	Cooling water outlet temperature T_{cout} °K
Experiment	343.15	0.695	294.65
Model	341.21	0.690	293.50
Deviation	0.57 %	0.72 %	0.39 %

3. Trial 3

Material: Water

Output results

Output variable	Evaporation temperature T_e °K	Product flow rate $Q_p \times 10^{-5} m^3/s$	Cooling water outlet temperature T_{cout} °K
Experiment	343.15	0.480	294.65
Model	341.21	0.460	293.45
Deviation	0.57 %	4.17 %	0.41 %

4. Trial 4

Material: Water

Output results

Output variable	Evaporation temperature T_e °K	Product flow rate $Q_p \times 10^{-5} m^3/s$	Cooling water outlet temperature T_{cout} °K
Experiment	343.15	0.580	294.65
Model	341.20	0.561	293.38
Deviation	0.57 %	3.28 %	0.43 %

5. Trial 5

Material: Water

Output results

Output variable	Evaporation temperature T_e °K	Product flow rate $Q_p \times 10^{-5} m^3/s$	Cooling water outlet temperature T_{cout} °K
Experiment	343.15	0.495	294.65
Model	341.20	0.462	293.30
Deviation	0.57 %	6.67 %	0.45 %

6. Trial 6

Material: Water

Output results

Output variable	Evaporation temperature T_e °K	Product flow rate $Q_p \times 10^{-5} m^3/s$	Cooling water outlet temperature T_{cout} °K
Experiment	343.15	0.485	294.65
Model	341.19	0.468	292.95
Deviation	0.57 %	3.51 %	0.58 %

7. Trial 7

Material: Milk

Output results

Output variable	T_e °K	$Q_p \times 10^{-5} m^3/s$	T_{cout} °K	w_p
Experiment	343.15	0.64	294.65	0.185
Model	342.91	0.698	293.15	0.172
Deviation	0.07 %	9.06 %	0.51 %	7.03 %

8. Trial 8

Material: Milk

Output results

Output variable	T_e °K	$Q_p \times 10^{-5} m^3/s$	T_{cout} °K	w_p
Experiment	342.15	0.65	294.65	0.178
Model	342.93	0.701	292.20	0.178
Deviation	0.23 %	7.85 %	0.83 %	0 %

9. Trial 9

Material: Milk

Output results

Output variable	T_e °K	$Q_p \times 10^{-5} m^3/s$	T_{cout} °K	w_p
Experiment	342.15	0.62	294.65	0.188
Model	343.00	0.632	293.20	0.184
Deviation	0.25 %	1.94 %	0.49 %	2.13 %

10. Trial 10

Material: Milk

Output results

Output variable	T_e °K	$Q_p \times 10^{-5} m^3/s$	T_{cout} °K	w_p
Experiment	349.15	0.78	294.65	0.145
Model	349.39	0.764	292.33	0.131
Deviation	0.07 %	2.05 %	0.79 %	9.66 %

11. Trial 11

Material: Milk

Output results

Output variable	T_e °K	$Q_p \times 10^{-5} m^3/s$	T_{cout} °K	w_p
Experiment	349.15	0.54	294.65	0.136
Model	349.43	0.547	292.38	0.144
Deviation	0.08 %	1.30 %	0.77 %	5.88 %

12. Trial 12

Material: Milk

Output results

Output variable	T_e °K	$Q_p \times 10^{-5} m^3/s$	T_{cout} °K	w_p
Experiment	349.15	0.64	294.65	0.144
Model	349.42	0.605	292.35	0.140
Deviation	0.08 %	5.47 %	0.78 %	2.78 %