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FACTORS AFFECTING HEAT TRANSFER IN THE FALLING FILM EVAPORATOR

A THESIS PRESENTED IN PARTIAL FULFILMENT OF THE
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Abstract

A pilot scale single-tube falling film evaporator made of stainless steel (2 metres length) was used to gain some understanding of the mechanism of evaporation in this type of evaporator.

Results obtained from commercial milk evaporators were used to select operating conditions on the pilot evaporator. The study was carried out using the simple liquids: water and sugar solutions.

It was found that the overall heat transfer coefficient decreased with increase of the overall temperature difference. The rate of decrease of the overall heat transfer coefficient is more rapidly in the range of 3-8 °C than in that of 8-18 °C.

The dependence of the overall heat transfer coefficient on the evaporating temperature was observed in the range of 70 °C to 90 °C.

The longer the heating tube length, the lower the overall heat transfer coefficient on this pilot evaporator.

The relationship of the overall heat transfer coefficient with average Reynolds number and liquid viscosity obtained was:

$$U = 0.939 + 1.58 \times 10^{-3} Re - 3.5 \times 10^{-7} Re^2 + 6.45 \times 10^{-4} \mu$$

$$(R^2 = 97.8\ \%)$$

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Chapter 1

Introduction

Evaporation is a unit operation in which solutions are concentrated by evaporating a solvent. It is one of the oldest means ever to have been adopted for separating liquid mixtures by heat on an industrial scale (Billet, 1989). In the food industry, as in other areas of application, the evaporated portion is usually pure water, although some other volatile components of the solutions are also partially or totally vaporized. Evaporation is, and no doubt will remain, a major technique used for the removal of water in the dairy industry (Fergusson, 1989).

Evaporation is extensively used in food industry for:

(1) reducing storage and shipping volumes for liquid foods, hence to reduce storage, transport and distribution costs;

(2) pre-concentrating foods prior to drying, freezing or sterilisation and hence saving energy in subsequent operations;

(3) increasing the solids content of a food and hence providing storage stability for it by reducing water activity;

(4) providing concentrates that are convenient to use for consumer and for the manufacturer.

In an evaporator, heat must be supplied to the liquid through a heat exchange surface, and the vapour and liquid separated. Evaporation is therefore both a heat exchange operation and a vapour-liquid separation.

Basically an evaporation system consists of:

(1) a heat exchanger to supply sensible heat to raise the liquid to its boiling point and provide the latent heat of vaporisation;

(2) a separator in which the vapour is separated from the concentrated liquid phase;

(3) a condenser to effect condensation of the vapour and its removal from the system;

(4) a vacuum device to withdraw the non-condensables for maintaining the consistent evaporating temperature.

Although generally the underlying principle is quite simple in evaporation, complications arise as a result of the infinite variety of products which have different properties. The behaviour of the liquid during evaporation introduces a number of aspects of which consideration must be given in designing and selecting evaporators. In other words, many parameters that depend on characteristics properties of the liquid govern the choice of equipment and the operating conditions. They generally include: the liquid's viscosity in relation to the degree of concentration required; the tendency to salting, scaling, and fouling; the hold up time for thermally unstable products; and reduction of evaporating cost. For dairy products, the particular properties are creaming, souring, foaming, undesirable changes in viscosity and taste, discolouring, and burn-on at the surface of the heaters.

Due to the heat-sensitive nature of milk and the energy costs rising, there are two aspects of the evaporation process which have received much attention during its development: one is to improve the quality of the finished product following an evaporation process to enable a better price to be obtained from it; another is to reduce the cost of removing water by this means. These can be seen from the history of evaporation technology in dairy industry.

The evaporation of milk has been known for many years, even as early as in the year of 1200 when Marco Polo described the production of a pasta, which was effectively as milk concentrate, in Mongolia (Westergaard, 1983).

The most simple evaporator was an ordinary open pan heated with fire, hot water or steam. The simple evaporator was used to produce storable milk with adding sugar by De Heine in 1810 (Wiegand, 1985).

In 1813, E. C. Howard invented a vacuum pan with a condenser to deal with the vapour produced (Taylor, 1982). This invention by lowering the evaporating temperature improved the quality of products and was a important step on the way to industrial production of concentrated and durable dairy products.

In 1850, Gail Borden was the first person to concentrate and sell milk on an industrial scale. He used a vacuum pan with a heating coil to enlarge the heating surface (Armerding, 1966). At that time sweetened condensed milk was produced in the United States.

In 1852, Robert Von Seelouitz was granted a patent for an evaporator, the so-called Robert evaporator, which had a vertical tube bundle arranged around a wide central circulation tube. A natural circulation, which improved the heat transfer, occurred in it. This kind of evaporator had been improved and used in the dairy industry for quite long time. Its use was a important precondition for high capacities and continuous operation.

In 1901, J. F. Kestner invented the so-called climbing-film evaporator, which was fitted with longer heating tube (about 7 m long and 50 mm diameter) and arranged the chamber for the separation of the liquid from vapour alongside the calandria. The type of evaporator was developed on the basis of the Robert evaporator, but its heat transfer coefficients were higher and it was operated with less recycle than the Robert evaporator. In addition, the amount of recycle was controllable in climbing film evaporator, but not in Robert evaporator.

With the development of production, high quality products were required. The problems in using a climbing film evaporator in dairy industry was that the hold up of liquid in evaporator was high (that means the residence time of the milk in the evaporator was rather long), the static pressure nearly prevented the liquid in the bottom parts of the tube from boiling, further worsens the heat transfer in this area considerably. The evaporator cannot be used effectively with small temperature difference.

These disadvantages of the climbing film evaporator were recognized in the earlier 1930's. In 1935, the falling film concept was introduced by D. D. Peebles and P. D. V. Manning and the first patents are from this period (Hallström, 1988).

However, because stainless steel was very expansive at that time, no suitable pumps were available and the problem of how to distribute the liquid evenly onto the heating tubes was not resolved, a commercial falling film evaporator was not manufactured for use in industry until 1953 by Wiegand in Germany. Since then, the falling film evaporator has been developed quickly. In many spheres this type of evaporators have almost entirely displaced the other types of evaporators and this is true in the case of the dairy industry. Today all evaporators used in New Zealand dairy industry are the falling film evaporators (FFE).

During the development of evaporation technology, two methods of reducing energy consumption were created and are still used today. One is multiple effect evaporation and another is vapour recompression.

In England in the year 1825, and in France in the year 1833, it was suggested to use the heat which is contained in the vapour produced in one evaporator for the heating of another evaporator and, if need be, to heat a third evaporator by means of the vapour produced in the second one, i.e. to operate according to the so-called multiple effect principle (Hallström,1988). But, before about 1910, most evaporation plants had only one effect. The first continuous multiple effect evaporators were used in the middle of 18th century in the sugar industry (Hahn, 1985). The multiple effect operation was introduced into dairy plant at the end of the nineteen thirties in European for producing evaporated cream (Wiegand, 1985). The theoretical specific steam consumptions equal to the reciprocal value of the number of effects, it is therefore 100% with only one effect, 50% with two effects, 33.3% with three effects. But the number of effects is limited by the total temperature difference available. In the food industry it is often necessary to avoid boiling temperatures above 70 °C when dealing with sensitive products due to the deposit on the heating surface and the products' quality. On the other hand it is very costly to operate with boiling temperature below 40 °C. Low temperature not only results in a very great increase of the cooling water consumption, but also in a considerable increase of the dimensions of the evaporator, for one reason because of the higher vacuum required and for another because of the higher viscosity of the product with its lower heat transfer.

The vapour can be compressed thermally by high-pressure steam-jet ejectors; or mechanically, by positive-displacement or centrifugal compressors to higher pressure and used as heating medium. The enthalpy difference between the vapour and steam is about 24 kJ/kg, while about 2239 kJ/kg between water and steam. It is obviously a large saving in energy to compress the vapour. Although steam jet compressors were equipped on milk evaporators in European after World War I, and mechanical vapour compressors were first found in the Swiss dairy industry during World War II (Wiegand, 1988), they were scarcely used before the oil crises in 1973 (Hallström, 1988).

After the two energy crises of 1973 and 1978, energy requirement became a serious problem, the number of effects increased greatly. In 1966, the first quadruple effect evaporation plant was produced, in 1974, the first five effect plant, in 1976, the first six effect plant and shortly after this, the first seven effect evaporator was installed (Hahn, 1985). In 1979, the mechanical vapour recompression was used once again in Germany in a triple effect plant for the evaporation of whey (Kessler, 1985). Now the most recent developments are a single effect evaporator equipped with a simple fan which makes it possible to operate with a temperature difference of about 4 °C (Fergusson, 1989).

The reasons for widely using the falling film evaporator in dairy industry are that the falling film evaporator has its peculiar characteristics, which can be described as follow:

(1) The falling film evaporator can be operated at any low temperature difference. This means that the falling film evaporator can be equipped with as many effects as one wanted;

(2) The lower temperature difference is favourable to mechanical vapour recompression and multiple effects operation;

(3) The very short residence time of products is a result of the small amount of liquid hold up and the high flow velocity. Since the product is generally processed in single-pass all liquid particles have an almost equal time of direct contact in the evaporator. Thus undesirable bacteriological, physical and chemical changes in the milk resulting from excessive heat treatment are avoided (Knipschildt, 1984);

(4) Due to the high flow velocity and low temperature difference, the deposit on the tube wall is small, i.e. the operating time before cleaning is required can be extended;

(5) High final concentration are possible in the falling film evaporator since the high vapour velocity propel viscous products without risking plugging, and the low temperature difference used (the minimum order of 2 °C or 3 °C, Gray, 1984) makes it possible to reach high concentration without scaling of the tubes;

(6) The capability of evaporation of the falling film evaporator can be extended to satisfy any requirement due to single construction and the possibility of using longer tubes (15 m or more). Because low operating costs, low steam consumption, and high quality can be achieved more easily in bigger evaporating plants (Wiegand, 1978);

(7) Relatively very small amount of cleaning agents is needed as the liquid volume in the evaporator is low.

It can therefore be seen that the falling film evaporator fully satisfies the demands of the modern dairy industry of today, i.e. the possibility of large capacities, economical running, reliable operation, careful treatment of the products, continuous operation, and the processing of as much as possible in single pass.

The general processes occurring in the falling film evaporator can be briefly described as follows: The liquid to be concentrated is preheated and pumped into the top of the falling film evaporator where it flows though the distributor. The liquid then flows down on the inside of heating tubes as a thin boiling film. The heat flux which is released by condensing steam in the steam jacket outside the heating tube transfers to heating tube walls and conducts through walls, then the heat issues from the inside of tube walls to make a portion of the falling film evaporate. The downward movement of falling film caused by gravity is assisted by the vapour produced which likewise flows downward. The vapour is separated from concentrate in a separator placed at the base of the heating tube, and then is used as the heating medium in the following effect, or is recompressed, or goes to a condenser.

It has been calculated that in New Zealand 1% of total energy consumed is expended in evaporation process. In the manufacture of milk power, 50% of the energy consumed is required at the evaporation stage (Jebson, 1989).

So it is necessary to understand the mechanism of evaporation in falling film evaporator to let the industrial people operate evaporators at their maximum capacity and efficiency.

Despite its industrial importance, however, there are relatively few papers in the literature in which precise performance detail are available, especially for milk evaporation. Much of the information obtained from actual plant operation by evaporator manufacturers or process plant owners has never been published, and much additional information is classed as trade secret and never released.

During last few years, a project has been carried out in the Food Technology Department of Massey University with the help of dairy diploma students to investigate the performance of the falling film evaporator operated in dairy plants in New Zealand. Some valuable information was gained.

However, there is a lack of data for three effect twelve passes evaporator, a test of this type evaporator was carried out at powder plant II of Kiwi Dairy Company. At same time, all data of the whole milk evaporators reported be dairy diploma students were summarised and computed. After that, a pilot scale single-tube falling film evaporator was set up at the pilot plant to study heat transfer in it. The results obtained from commercial milk evaporator was used to select variables in the further experimental work.

Chapter 2

Literature Review

Very little published information is available concerning the influence of liquid flow rate, Reynolds number, overall temperature difference, and absolute temperature and pressure of evaporation on the heat transfer characteristics of falling film evaporators, especially using in dairy industry.

The phenomena occurring inside an evaporator tube are complex (Munro, 1989). This review therefore mainly focused on published reports about flow dynamics and heat transfer in falling film as individual phenomena and the factors influencing overall heat transfer coefficients in the falling film evaporators.

2.1 Dimensionless numbers for film flow

Film flow is a special case of two-phase flow (Fulford, 1964). From the view of film flow patterns, the flow pattern of the falling film flow in vertical gas-liquid systems is annular (Nicklin and Davidson, 1962).

As used in other liquid flow and heat transfer researches, the dimensional analysis method is also used in film flow and its heat transfer. There are five dimensionless group numbers derived from theoretical analysis. They are Reynolds number (Re), Prandtl number (Pr), Froude number (Fr), Weber number (We), and Modified Nusselt number (Nu').

(1) The Re, which is customary to describe the flow type of liquid, is defined for film flow in tube as follows:

$$Re = \frac{4\Gamma}{\mu} \qquad (1)$$

Where: Γ is called irrigation density, which is defined as mass flow rate per unit of perimetric length (kg/m.s).

μ is liquid viscosity (Pa.s).

(2) The Pr, which describes the physical properties of liquid, is expressed as follows:

$$Pr = \frac{C_p \mu}{\lambda} \tag{2}$$

Where: C_p is specific heat of liquid (J/kg.K).

λ is thermal conductivity of liquid (W/m.K).

(3) The Fr, which gives an indication of the important of gravity effects and can also be used to predict the nature of the film flow, is defined as follows:

$$Fr = \frac{\mu^2}{gd} \tag{3}$$

Where: g is gravitational acceleration (m/s^2).

d is the inside diameter of tube (m).

(4) The We, which is important where surface tension effects come into play, is expressed as follows:

$$We = (\frac{\rho u^2 \delta}{\sigma})^{\frac{1}{2}} \tag{4}$$

Where: ρ is liquid density (kg/m^2).

u is velocity of film flow (m/s).

δ is film thickness (m).

σ is surface tension of the liquid (N/m).

(5) The Nu', which includes the surface heat transfer coefficient of falling film, is defined as follows:

$$Nu' = \frac{\alpha_L}{\lambda}(\frac{u^2}{\rho^2 g})^{\frac{1}{3}} \tag{5}$$

Where: α_L is surface heat transfer coefficient on the side of the liquid (falling film)(W/m^{2}K).

2.2 Falling film flow dynamics

2.2.1 Definition of flow regimes

It is well known in fluid flow studies that below a certain critical value of the Reynolds number the flow will be mainly laminar in nature, while above this value, turbulence plays an important part. The same is true of film flow, though it must be remembered that in thin film a large part of the total film thickness continues to be occupied by the relatively non turbulent " laminar sublayer " even at large flow rates (Fulford, 1956).

Strictly speaking, the flow regime of a film cannot be defined uniquely as laminar or turbulent because of the presence of the free surface in film flow. Under suitable conditions, it is possible to have smooth laminar flow, wavy laminar or turbulent flow, where the wavy flows may be subdivided into gravity or capillary types.

In order to simplify the problem, three different flow regimes have been identified (Fulford, 1956, Hallström, 1985):

(1) Smooth laminar;
(2) Wavy laminar;
(3) Turbulent.

The transition form smooth to wavy laminar can occur for Reynolds number in the 0 to 30 range (Benjamin, 1957). Based on experiments using liquids with wide differences in physical properties falling in vertical columns, it appears that wave formation will develop when Fr > 1 (Jackson, 1955). Hallström (1985) reported that for water the smooth laminar regime takes place for Reynolds number below 20-30, and the wavy laminar regime starts at Re = 30-50, if the vapour shear is negligible.

Turbulent motion first appears at Re = 250-500 (Fulford,1962), Re = 200-600 (Blass,1979), and Re = 1080 (Dukler & Bergelin, 1952); fully turbulent flow exists at Re = 1000-3000 (Hallström,1985).

Schwartzberg (1988) summarised several researchers' work (Yanniotis, 1983, Chavarria, 1983, et al.) and reported that at Re < 500 heat transfer coefficient of the evaporating fluid increased linearity as Reynolds number increased, at 500 < Re < 1600, heat transfer coefficient remained fairly constant or decreased slightly, at Re > 1600, heat transfer coefficient increased slightly as Reynolds number increased.

Moresi (1985) comments that although there is much scatter in the publish experimental data, most investigators seem to support a lower limit for the transition Reynolds number between 1000-1600, with a less well-marked upper value of about 3200.

Dukler (1960) has pointed out that the transition from laminar to turbulent flow in a thin film cannot be expected to be sharp, since due to the thinness of films, a large portion of the total film thickness is occupied by the laminar sublayer even at flow rates above the critical Reynolds number. The transition is likely to be a gradual process.

The correlations developed by Chun and Seban (1971) working with water and Wilke (1962) working with water and ethylene glycol show that the transition point (critical Reynolds number, Re_{CR}) from laminar to turbulent can be detected by the change in heat transfer rate.

$$Re_{CR} = 5800\ Pr^{-1.06} \tag{6}$$

$$Re_{CR} = 2460\ Pr^{-0.65} \tag{7}$$

For equation (6) and (7), Pr is 1.77-5.7 (Chun and Seban) and 5.4-210 (Wilke) respectively.

Chun, Seban and Wilke nevertheless suggested that a Weber number of the order of unity may be used as a transition criteria for falling films.

2.2.2 Falling film flow characteristics.

Knowledge of falling film characteristics is necessary for heat and mass-transfer theoretical analysis (Fulford, 1964). The pioneering work in this field was that of Nusselt (1916) on the theoretical determination of velocity fields, average velocity and film thickness for smooth laminar films falling at steady state on flat surfaces with zero free surface interfacial shear. Since then, few papers have appeared in the literature, and most work has been done only in laminar flow regime, because beyond this regime the film flow is usually so complex that no satisfactory general theory has yet been possible (Blass, 1979). Dukler and Bergelin (1952) deduced velocity profile and film thicknesses for turbulent film flow based on the universal dimensionless velocity profile equations of Nikuradse (1933). Blass (1979) provided the falling film thickness as a function of the Reynolds number graphically at Re = 10-5000, as well as the equations to calculate the mean film thickness and the mean film velocity.

Film surface instabilities are dampened by surface tension, as established in the work of Benjamin (1957), who also states that total stability cannot be obtained by merely increasing surface tension and that vertically falling films are unstable at all flow rates. Brauer (1956) has reported experimental values of the critical Reynolds number of film containing small qualities of surface-active materials in solution. In this case, the value of critical Reynolds numbers appeared to depend on the surface tension of the solution. This effect is probably due to the layer of surface-active material present at the interface.

Blass (1979) discussed the Marangoni effect, which is the surface motion of the liquid as a results of local surface tension differences. If there is a somewhat thin section of the film with a smaller surface tension than the neighbouring thick sections of the film, then, on the basis of the tendency toward a minimum free surface energy, the liquid in the regime close to the interface is drawn from the thin section of the film into the thicker zone. As a result, the film can be thinned so much that it ruptures and flows separate rivulets. This effect would be more likely to occur in highly concentrated solutions with some lower surface tension materials.

Two extensive reviews of the film flow characteristics was given by Fulford in 1964 and Blass in 1979.

2.2.3 Effect of adjacent gas stream on film flow.

In general the various treatments predict that, as the velocity of the gas stream is increased, the mean film thickness for a given liquid flow rate decreases, in the case of downward concurrent flow of the two phases, due to the acceleration of the surface by the gas stream drag (Fulford, 1964).

The effect of gas stream motion on the liquid film flow was first examined by Semenov (1944), he proposed interrelationship of liquid throughput, film thickness and tangential force which can only be applicable for a film with a thickness not great than 150 microns and at moderate gas rates. Brauer (1960) also carried out a detailed analysis of the smooth film and gas streams inside vertical tubes.

For the wavy laminar flow, a relation published by Kapitsa (1948) predicts mean film thicknesses based on irrigation density and mean gas velocity,when the gas stream does not seriously affect wavelength. Wavy film flow shown to be more stable than smooth film, and about 7% thinner than smooth film at same flow rate.

For the turbulent film regime flow, Dukler (1960) provides numerical results relating the effects of a cocurrent downward flowing gas steam on film thickness and liquid velocity. Film heat transfer under these circumstance is also considered.

Zhivaikin (1962) concludes that film thickness is little affected by cocurrent downward flow for gas velocities up to 4 m/s, but that the film thickness decreases at higher gas velocities. He also presents a correlation to predict film thickness, between gas velocity 4 m/s and the velocity at which spray formation commences.

Any pressure drop due to gas stream friction losses and the energy to create gas velocity will cause an increase in evaporating temperature. Consequently, since the falling film evaporators operate at low temperature difference, such pressure drop could significantly reduce the total available heat-transfer driving temperature difference.

Zhivaikin and Volgin (1964) shown experimental results which indicate that pressure drop per unit length is not always constant in the direction of gas flow. They explained that the reason is due to changes in the shape of the gas stream velocity profile and to changes in energy on accelerating the liquid film near the inlet. A particularly interesting feature of gas film is the friction pressure drop in a wetted-wall tube is considerably larger than in the dry tube.

Sinek and Young (1960) published an expression to calculate the friction pressure drop in falling film evaporators and they conclude that for most cases of falling film evaporators, the vapour rate per tube is not high enough for friction pressure drop to deviate significantly from the dry-tube value. On the basis of his own measurements and all studies evaluated by Feind, Hempel (1967) developed a very simple empirical pressure drop relationship.

2.3 Heat transfer in falling films

2.3.1 Heat transfer mechanisms and analysis

Heat transfer mechanisms in evaporating films are not well understood (Kroll and McCutchan 1968). In general, the heat transfer mechanism is dependent on the liquid flow pattern and the heating rate.

Billet (1989) stated that the flow pattern in the evaporator decidedly affects heat transfer on the liquid side of the heater, and that the amount of vapour in the liquid is a crucial factor. The relationship of the heat transfer coefficient is subject to various physical laws that depend on whether boiling is nucleate or convective. Any expressions to define the heat transfer mechanism in two-phase flow of this nature must be very complicated.

Four possible mechanisms are discussed in literature (Dengler & Addoms, 1956; Sinek & Young, 1962; Chun & Seban, 1971; and Angeletti & Moresi 1983). These include:

(1) Vapour-liquid interface evaporation, where heat transfer is mainly governed by liquid flow conditions and thermophysical properties.

(2) Low rate nucleate evaporating, where bubbles form and break but do not affect heat transfer greatly. Observations shows that in this case bubbles move with the film without reaching the surface.

(3) High rate nucleate evaporating, where the presence of the bubbles significantly affects liquid flow pattern and where the bubbles follow one another in channels across the film.

(4) Vapour film evaporating, where vapour is evolved from bubbles produced at the tube surface.

The low rate nucleate evaporating mechanism is regarded by Sinek and Young (1972) as the most probable one for tube film evaporators. They also propose a model that accounts for vapour-liquid interface temperature rise due to the presence of the bubbles.

Kroll & McCutchan (1968) pointed out that there can be no doubt the interface evaporation plays the major role, if not the sole role, in the heat transfer.

O'Connor & Russell (1978) stated that depending on local conditions, several heat-transfer mechanisms may coexist in a filling film evaporator.

Chun and Seban (1971) observed nucleation only at the bottom of an evaporator. They also indicated that their experimental results showed that a superheat of 3.7 °C was required for nucleation in water at atmospheric pressure, and that the necessary superheat increased as pressure decreased. Bell (1982) fount out that the ordinary engineering surfaces require surface superheats on the order of at least 3 to 5 °C to insure nucleation of a stable boiling phase.

According to Blass (1979), for water and aqueous solutions, transition from free surface evaporation to nucleate boiling in films occurs at heat fluxes in the rang of 40 to 60 kW/m^2, and at temperature differences in the range of 6 to 7.5 °C.

Billet (1989) found that if the heating rate is fairly high, e.g. if $q > 30$ kW/m^2, surface evaporation no longer applies, and vapour bubbles will be formed at scores and other nucleation sites on the heating surface. The critical degree of overheating required before vapour bubbles can just be formed at the nucleation sites is up to 7 °C. The attendant mixing of the liquid entails a significant improvement in heat transfer.

But this contrasts with Steiner and Ozawa's (1982) statements that bubble formation reduced heat transfer benefits induced by high irrigation densities and that turbulence promoters were not likely to improve heat transfer significantly.

Dengler and Addoms (1956) pointed out that in all cases an increase in liquid velocity past a surface was shown to raise the temperature difference required to initiate nucleate boiling at that surface. Stated another way, at a constant temperature difference the heat flux contributed by nucleate boiling decreases with increasing velocity until the boiling ceases entirely. They thought this effect of liquid velocity on nucleation, incidentally, can be observed by anyone stirring a pot of boiling water. O'Connor and Russell (1978) also stated that the greater velocity tends to prevent bubble formation at the wall.

2.3.2 Heat transfer coefficients

Information on heat transfer coefficients in evaporators is necessary to monitor the evaporation process closely (Jebson, 1988).

2.3.2.1 Surface coefficients of heat transfer for boiling liquid

The local film of heat transfer coefficients for boiling film is a indicator which reflects the heat transfer in the liquid film more accurately. So it is used in the theoretical research of heat transfer in falling film.

Referring to the Nusselt film theory, Moresi (1985) conducted that for laminar flow, the surface heat transfer coefficient is approximately equal to the ratio of the liquid thermal conductivity to average film thickness (δ), under the conditions of minimum evaporation taking place at surface of liquid film. That is:

$$\alpha_L = \frac{\lambda}{\delta} \tag{8}$$

If the falling film flow is of constant properties and rheological behaviour of the power-law type, and is steady, uniform, the irrigation density (Γ) is fund to be related to be film thickness as follows (Moresi 1980):

$$\Gamma = \frac{\gamma}{2\gamma+1}\,\rho\,(\frac{\rho g}{K})^{\frac{1}{\gamma}}\,\delta^{\frac{(2\gamma+1)}{\gamma}} \tag{9}$$

Where: K is the consistency index;

γ is the flow behaviour index.

For a Newtonian fluid, γ is equal to 1, while K coincides with the fluid dynamic viscosity (μ). In these conditions, the equation reduces to the following expression:

$$\Gamma = \frac{\rho^2 g \delta^3}{3\mu} \tag{10}$$

So the film thickness can be yielded by rearranging the equation (10), i.e.

$$\delta = (\frac{3\mu\Gamma}{\rho^2 g})^{\frac{1}{3}} \tag{11}$$

By combining equation (8) and (11) and introducing the Modified Nusselt number and Reynolds number, it is possible to derive:

$$\alpha_L = \frac{\lambda}{(\frac{3\mu\Gamma}{\rho^2 g})^{\frac{1}{3}}} = \frac{\lambda}{(\frac{3}{4})^{\frac{1}{2}}(\frac{\mu^2}{\rho^2 g})^{\frac{1}{3}}(\frac{4\Gamma}{\mu})^{\frac{1}{3}}}$$

$$\frac{\alpha_L}{K}(\frac{\mu^2}{\rho^2 g})^{\frac{1}{3}} = (\frac{4}{3})^{\frac{1}{3}}\frac{1}{Re^{\frac{1}{3}}}$$

$$Nu' = 1.1\ Re^{-\frac{1}{3}} \tag{12}$$

This equation is a basic equation and can be used to predict local film heat transfer coefficient only under true laminar film flow.

Based on this basic equation, numerous Nusselt type correlations were developed to predict the local film heat transfer coefficients at many different flow conditions. The general equation form is expressed as follows (Schwartzberg, 1989):

$$Nu' = c\ Re^{n}\ Pr^{m} \tag{13}$$

Where c, n and m are constant.

Wilke (1962) gave the correlations for non boiling falling film at heat transfer rate $q < 18.6\ kW/m^2$, temperature difference < 20 °C, and Pr = 5-210. Chun and Seban (1971) presented the correlations for boiling film at Pr = 1.17-5.7. All constants are listed in table 1.

The available equations for heat transfer rates in turbulent falling film in the literature are reviewed by Moresi (1985).

Peeples et al. (1962) presented more than 10 Nusselt type equations to describe the heat transfer characteristics of different dairy products. Some constants are listed in table 2.

Table 1. Values of constants for falling film heat transfer correlations

	Re≤400	400≤Re≤800	Re>800	Re≤1600	Re>1600
c	0.0614	0.00112	0.0066	0.606	0.0038
n	0.533	1.2	0.933	- 0.22	0.4
m	0.344	0.344	0.344	0	0.65
ref.	Wilke			Chun & Seban	

Table 2. Values of constants for dairy products equations

	Regular skim milk	Reconstituted skim milk (8.8 % SNF)	Regular milk	Whole milk (10 % fat, 12 % SNF)
c	0.46	0.41	0.17	0.27
n	0.535	0.538	0.63	0.577
m	0.4	0.4	0.4	0.4

A correlation developed by Dammann (1973) covers a wide rage of conditions and is claimed to give estimates with less than 20% error for the case of surface boiling falling film and within 12% for non-boiling films.

Whitt (1966) analyzed the performance of three sizes of falling film evaporator and compared published data. He found that the surface heat transfer coefficients for evaporation of falling films are some 7 times greater than those for heating alone and at very low flow rates the heat transfer coefficients of falling film are approximately proportional to the specific liquid flow rates.

2.3.2.2 Overall heat transfer coefficients

The overall coefficient of heat transfer is can be theoretically derived from the surface coefficient on both sides of the heat transfer surface and the thermal resistance of tube wall according to following equation if the tube curvature being ignored:

$$U = \frac{1}{\frac{1}{\alpha_V} + \frac{\delta_W}{\lambda_W} + \frac{1}{\alpha_L}} \tag{14}$$

Where: U is the overall heat transfer coefficient ($W/m^2.K$);

δ_W is the thickness of tube wall (m);

λ_W is the thermal conductivity of tube wall (W/m.K);

α_V is surface heat transfer coefficient on the side of the vapour condensation ($W/m^2.K$).

The overall heat transfer coefficient can be regarded as an expression for the efficiency of heat transport from the medium through the wall to the boiling liquid. In other words, it indicates the amount of heat that is given off from unit area of heater surface in unit time and per unit of total temperature difference and that is imparted to the liquid through the wall. It can be used as an assessment criteria for assessing entire evaporation process.

Its reciprocal represents the total resistance to heat transmission. If there are scaling or fouling on both sides of heating tube, the equation is given as:

$$\frac{1}{U} = \frac{1}{\alpha_V} + R_V + \frac{\delta_W}{\lambda_W} + \frac{1}{\alpha_L} + R_L \tag{15}$$

Where: R_V is outside heat transfer resistance of heating tube ($m^2.K/W$);
R_L is inside heat transfer resistance of heating tube ($m^2.K/W$).

In practice, however, the overall coefficient of heat transfer frequently cannot be derived mathematically from the surface coefficient. This is either because the physical properties required for the calculation are inadequately known or because the equations for the surface coefficient are not entirely valid under the intended operating conditions. Another reason is that the equations apply only under the assumption that the thermal resistance is not increased, i.e. that heat transfer is not impaired by scaling.

In cases of this nature, an empirical method must be resorted to for predicting the overall coefficient of heat transfer. If the amount of heat transferred (Q), the steam temperature (T_S), and the evaporating temperature (T_E) are measured, the area (A) of the effective heat transfer surface is known, the mean overall heat transfer coefficient is given by

$$U = \frac{Q}{A\ (T_S - T_E)} \tag{16}$$

Kroll & McCutchan (1968) considered inlet flow rate, evaporating temperature, temperature difference, tube length and diameter, to be principal perimeters affecting overall heat transfer coefficient. Their experiment showed a decrease in heat transfer coefficient as length increase for lengths varying from 0.85 to 4 m, when using 3/4 -in O.D. tubes and water as evaporant. They observed no significant change in overall heat transfer coefficient in the temperature difference range of 6-11 °C and a corresponding heat flux varying between 20-50 kW/m^2. The maximum vapour velocity in their work was 46 m/s and the irrigation densities ranged from 0.075 to 0.718 kg/m.s. They concluded that the overall heat transfer coefficients should be used in comparing the performance of various evaporators rather than the surface heat transfer coefficient.

Sinek and Young (1962) reported no significant changes in heat transfer coefficient when heat fluxes vary in the range of 35 to 125 kW/m^2. When the tube diameter changes from 1 to 2-in at the same Reynolds number and evaporating temperature, the overall heat transfer coefficient increase 70-80%. This may be because of the increased pressure gradient caused by smaller diameter. They also found that the definite increase in heat transfer coefficient with increase evaporating temperature, for a 55 °C evaporating temperature increase at constant feed flow rate, the overall heat transfer coefficient increase 3 to 4 times in a 1 -in tube, and almost 2 times in a 2 -in tube.

Blass (1979) pointed out that most of the experiments do not take into account the effect of the vapour flow on heat transfer. The results are therefore applicable to falling film evaporators only at low heat fluxes at heating surface or very large across-sectional areas for the vapour flow. So the calculation of heat transfer coefficients in falling film evaporators is not very reliable.

2.4 Steam condensation and non-condensibles

The commonest medium for heating evaporators is steam. It condenses on the walls and thus releases its condensation enthalpy. There are two ways of condensation, which are well described by the terms dropwise and film type.

In film condensation, which is more common than dropwise condensation, the liquid condensate forms a film, or continuous layer, of liquid that flows over the surface of the tube under the action of gravity.

In dropwise condensation, the condensate begins to form at microscopic nucleation sites and then grows coalescing with their neighbours to form visible fine drops. The fine drops, in tern, coalesce into rivulets, which flow down to the tube under the action of gravity, sweep away condensate, and clear the surface for more droplets. During dropwise condensation, large areas of the cold tube are bare and are directly exposed to the vapour. Because of the absence of the liquid film, the resistance to heat flow at these bare areas is very low and the heat transfer coefficient correspondingly high. The surface heat transfer coefficients in dropwise condensation are higher than those encountered in purely film condensation by a factor of about 6-20 (Billet, 1989).

Much of the experimental work on the dropwise condensation of steam was summarized by McCabe and Smith (1965). They stated that dropwise condensation is obtainable only when the cooling surface is contaminated, the quantity of contaminant or promote required to cause dropwise condensation is minute and it is more easily maintained on a smooth contaminated surface than on a rough contaminated surface.

It was pointed out that the longer the tubes, the lower the condensing surface heat transfer coefficient, because the heat transfer coefficient in condensation is inversely proportional to the 4th root of the tube length (Kessler, 1986). It can be seen from following equation for stationary saturated steam condensed on vertical surface in film type. The condensate coefficient of heat transfer (α_V) can be derived by the Nusselt theory (Billet, 1989).

$$\alpha_V = 0.94\ f_k\ (\frac{h_V}{\Delta T_V\ L})^{\frac{1}{4}} \tag{17}$$

Where: f_k is a constant of physical properties of the condensate film;

h_V is condensation enthalpy of the steam (J/kg);

ΔT_V is temperature difference cross the condensate film (K);

L is length of tube (m).

The most serious problem in steam or vapour condensation is the presence of non-condensibles. Small amounts of non-condensing gas usually are contained or occluded in liquid food being concentrated. Air also leeks into vacuum evaporators through joints and connections. It mixes with the steam or vapour flowing towards the condensing surface, and unless a means of steadily removing it is provided, it will accumulate at the condensing surface eventually to impede the process of condensation, and impair the heat transfer coefficient. The reasons are both that the non-condensibles form a layer on the condensing surface (air blanketing effect) and that non-condensibles lower the condensation temperature at a given total pressure and thus contributes further to a deterioration in the heat transfer efficiency.

Only traces of air caused a steep decrease. Any concentration of non-condensibles greater than 0.01% can have a drastic effect on the condensing heat transfer coefficient (Mincowycz & Sparrow, 1966, Sparrow, et al, 1967). Condensation rates are reduced 10%, when flowing steam contains 2% air, the reduction is 70% in stagnant zones (Collier, 1981).

In order to maintain good heat transfer performance, non-condensibles removal is usually accomplished by vacuum pumps or ejectors connected to the condensing vapour chamber. However, this also removes some vapour and reduces thermal efficiency. Schwartzberg (1988) pointed out that the mix of gases and vapour venting from vacuum device often contain 97% to 98% vapour, which is wasted. Steam losses might be reduced if we know how to control levels of non-condensibles.

It has been pointed out by Jebson (1988) that there is some fouling on the vapour side of the tubes in older milk evaporators due to the very small droplets of milk travelling with the vapours, thus, the condensing heat transfer coefficients are decreased.

2.5 Food (liquid) properties that affect heat transfer

The choice of type of evaporator depends primarily on the characteristics of the liquid to be concentrated. Due to the heat sensitive property of milk products, the falling film evaporators were selected and used widely in the dairy industry.

2.5.1 Viscosity

Food solution viscosities increase markedly during evaporation. The increase of viscosity of solution is not linear as the concentration increase. At high concentrations, small additional changes in the concentrations will lead to rapid change in the viscosity. This could result in reduced flow rates, decreased turbulence, and severe fouling. For skim milk concentrate, there is a sharp increase in viscosity above 35% total solids (Lewis, 1986).

Schwartzberg (1988) reported that the viscosity of weak liquor in the first stage of a sugar evaporator may be 0.32 cPs, that of discharged concentrate, 4.2 cPs. In orange juice evaporators, feed stage viscosities are roughly 0.5 cPs, those in the final stage, 50 cPs. Tomato juice becomes strongly non-Newtonian during evaporation; and high temperatures are needed to keep it reasonably fluid.

2.5.2 Density

Density, which is less temperature dependent but is concentration dependent, typically increases roughly 26% (Schwartzberg, 1989).

Kessler (1981) presented relationships for the density of whole milk over the temperature range 0-150 °C.

$$\rho = 1033.7 - 0.2308\,T - 2.46\,T^{2} \tag{18}$$

Schwartzberg (1989) comments that in falling film evaporators, viscosity and density affect film thickness and product residence time.

2.5.3 Thermal conductivity

The thermal conductivity of most foods is strongly influenced by the moisture content (Lewis, 1987). During evaporation, thermal conductivity of foods generally decreases 35% (Schwartzberg, 1989).

Woodams and Nowrey (1968) presented the literature values of thermal conductivity of various milk products.

For concentrated whole milk, the thermal conductivity can be calculated from following equation (More & Prasad, 1988):

$$\lambda = (0.59 + 0.0012\, T)(1 - 0.0078\, m_s) \tag{19}$$

Where: T is temperature (40-90 °C);

m_s is the total solids percentage (37-72 %).

For the sugar solution, it's thermal conductivity changes induced by concentration and temperature can be seen in table 3 (Woodams & Nowrey, 1968).

Table 3. Concentration and temperature induced thermal conductivity changes in sugar solution

	Concentration (%)	λ (W/m.K)
AT 50 °C	10.1	0.601
	20	0.571
	30	0.538
	30.9	0.502
	Temperature (° C)	λ (W/m.K)
For 30% sugar solution	1.5	0.478
	20.0	0.505
	50.0	0.538
	80.0	0.504

2.5.4 Specific heat

The different components in food have different specific heat values, so it should be possible to estimate the specific heat of a food from the knowledge of its composition.

Water has the greatest influence on the specific heat. During evaporation, specific heat of milk decreases 38% (Schwartzberg, 1989). Bertsch (1983) described how the specific heat of milk changes over the temperature range 50-140 °C. The specific heat of milk concentration has been described by Fernángez-Martín (1972) as:

$$C_p = (m_w + (0.328 + 0.0027T) m_s) 4.18 \tag{20}$$

Where: m_w is water percentage.

This equation was used over the temperature range 40-80 °C and total solids range 8-30%.

2.5.5 Boiling point elevations

For any aqueous solution, the boiling point is not exactly the same as for pure water due to the dissolved substance. The boiling point of the solution may rise considerably as the solid content increase. For well defined solutions the elevation of the boiling points is proportional to the molar concentration of the solutions. For aqueous solutions the proportionality factor is 0.52 giving:

$$\Delta T_{BPE} = 0.52\ \xi \tag{21}$$

Where: ξ is the mole fraction of solution (Hallström, 1985).

Food liquids are normally more complicated and boiling point rise has to be determined experimentally. For milk, Kessler date gives the values in table 4.

Table 4. Values of boiling point elevation for milk concentrate

Concentration (% TS)	16	27.5	39	49	62	69	73
ΔT_{BPE} (°C)	0.5	1	1.5	2	3	4	5

Any boiling point elevation will reduce the effective temperature difference for heat transfer in the evaporator and lower the capacity.

In the falling film evaporator, the standing liquid head is almost eliminated. However, the boiling point elevation caused by pressure drops to vapour flow still exists, but it is relatively small. The pressure drops due to vapour flow are usually in the 0.1 to 0.6 kPa range, i.e. at typical food evaporation temperature, boiling point rises due to flow pressure drop are 0.1 to 0.6 °c (Schwartzberg, 1989).

2.5.6 Surface tension

Milk has a surface tension of between 42.3-52.1 mN/m. It is considerably lower than the value for water because of the presence of fat and natural surface-active agents (Lewis,1987). For whole milk (4% fat) and skim milk, the surface tension was found to decrease in an almost linear fashion as temperature increased. The results could be represented by the following equation (Bertsch, 1983):

$$\sigma = 1.8 \times 10^{-4}\, T^2 - 0.163\, T + 55.6 \tag{22}$$

Where: σ is surface tension in mN/m;
T is temperature in °C (18-135 °C).

There appears to be little published work on the effects of concentration on the surface tension of milk (Lewis, 1986).

The surface tension between the wall and the products determines the tendency of the film towards boiling. At low surface tension, the bubbles can easily detach from the surface so that a higher critical heat flux is obtained. If the products has a high surface tension, the bubbles will coalesce to a film at a lower heat flux, thus reducing heat transfer coefficients (Mannheim and Passy, 1974). Low surface tension also promote stable, uniform wetting in falling film evaporators.

2.6 Operating conditions affecting heat transfer in the falling film evaporators

Besides the factors discussed above, there are still many other variables affected the heat transfer. Armerding (1966) listed thirty seven variables which could be vital to the function of the evaporators. For the falling film evaporator, following variables are critical.

2.6.1 Distribution device

A major difficulty in the falling film evaporator is the maintenance of a continuous film of liquid over the heating surface. So the distribution of liquid on each tube walls is very important when designing and operating the falling film evaporators.

A distribution device must fulfil the following functions:

(1) It can distribute liquid uniformly onto all tubes to guarantee sufficient irrigation density;
(2) The liquid has to be distributed evenly over the tube circumference;
(3) It must function unrestrictedly during the total operating time of the evaporation. Its structure must be simple and it must be easily cleaned.

There are four types of distributor that have been used:

(1) nozzles;
(2) distribution plates;
(3) spiders or other flow distributors inserted in the tops of the tube;
(4) wires formed by tops of tube projecting out of the tube sheet.

The most common used one is the distribution plate because it guarantee very good irrigation density and wetting of all tubes. The problem, however, is that distribution plate is sensitive to clogging or warping or distortion. So it is necessary to check distribution plate regularly.

2.6.2 Irrigation density

Complete wetting of the heater surface is an important requirement for avoiding trouble and for efficient heat transfer in falling film evaporators. In other words, it must be ensured in the design stages that a sufficiently high value is selected for the irrigation density, as otherwise the tube will not be completely covered by the liquid and dry spots will occur leading to crust formation and capacity losses, and the values for the overall heat transfer coefficient may be too low. On the other hand, however, if a too high value is chosen, the concentration ratio in single pass effect is lower and the required final concentration is not reached. The final concentration could be reached by multiple passes in each effect, but the residence time would be longer, and the capital and pumping costs would be higher.

The extent to which the heating surfaces are wetted depends on a number of factors, viz, the nature of the surfaces, the physical properties of the liquid, the design of the liquid distributor, and - in particular - the irrigation density and the heating rate (Billet, 1989).

Heat flux tends to increase critical Reynolds number (Re_{CR}, which is the transition point from laminar to turbulent) and vapour flow tends to decrease it. Hence heat flux by itself tends to promote incomplete wetting, but vapour flow produced by heating favours complete wetting. Design criteria that account for such effects should be developed to permit more reliable selection of irrigation density that ensure complete wetting in falling film evaporators. For instance, for concentrating fruit juice in the falling film evaporator, the irrigation density is always above 0.085 kg/m.s in the first effect and 0.25 kg/m.s in the last effect, and setting irrigation density > 0.085 kg/m.s $(\mu_n/\mu_1)^{1/5}$ in other stages (n being the effect number) (Schwartzberg, 1988).

2.6.3 Flash

If the temperature of liquid entering the top of the tubes is higher than the evaporating temperature in that effect, the flash will occur. The flash of liquid is considered to contribute the distribution of liquid on the heating tube, and than benefit for heat transfer coefficient (Billet, 1988).

But Sinek (1962) found that the overall heat transfer coefficients for run with 10 °C feed superheat are all roughly 10% lower than the corresponding coefficient with zero feed superheat in 24-ft tubes. He explained that this unforeseen event is possibly that flashing at the tube entrance produces such high velocities that falling-film flow is only established a few feet further down, the net effect being a reduction in heat transfer area. This explanation seems unlikely correct. It seems likely that the flashing at the tube entrance produces the laminar falling film flow due to the expanding of vapour volume, in a result of lowering heat transfer coefficients.

So maybe there is a relation between the heat transfer coefficients and the degree of flash. In addition, the effect of flash on the heat transfer coefficients maybe also dependent on the type of distributor.

Chapter 3

Commercial Milk Evaporator Test and Whole Milk Data Processing

3.1 The test of a three effect twelve pass milk evaporator

Although there is a lot of data on milk evaporators reported by the students who studied for Diploma of Dairy Science and Technology, there is a lack of the data on the three effect twelve pass milk evaporator. So a test was carried out at the powder plant II of Kiwi Dairy Company in Hawera.

3.1.1 The method

3.1.1.1 Equipment

The following equipment was used in this test:

(1) A portable Brookfield viscometer (LVTD);
(2) A portable thermometer (Jenway, Model 2003);
(3) A oven (Watvic, OM-24);
(4) An analytical balance (Mettler, AE-200);
(5) A steel tape and several containers.

3.1.1.2 Procedure

By using the simple equipment list above, the effects of operating variables on the performance of the evaporator can be determined.

The typical procedure is as follows:

(1) Take the samples from different passes in each effects on the evaporator, as well as the feed and the products from their silos;
(2) Measure the temperatures immediately after the samples were taken;
(3) Record feed flow rate and all temperatures on the control panel;

(4) Measure the total solids of all samples by means of the oven-balance;

(5) Measure the dimensions of each effects of the evaporator;

(6) Use a computer program to compute the data.

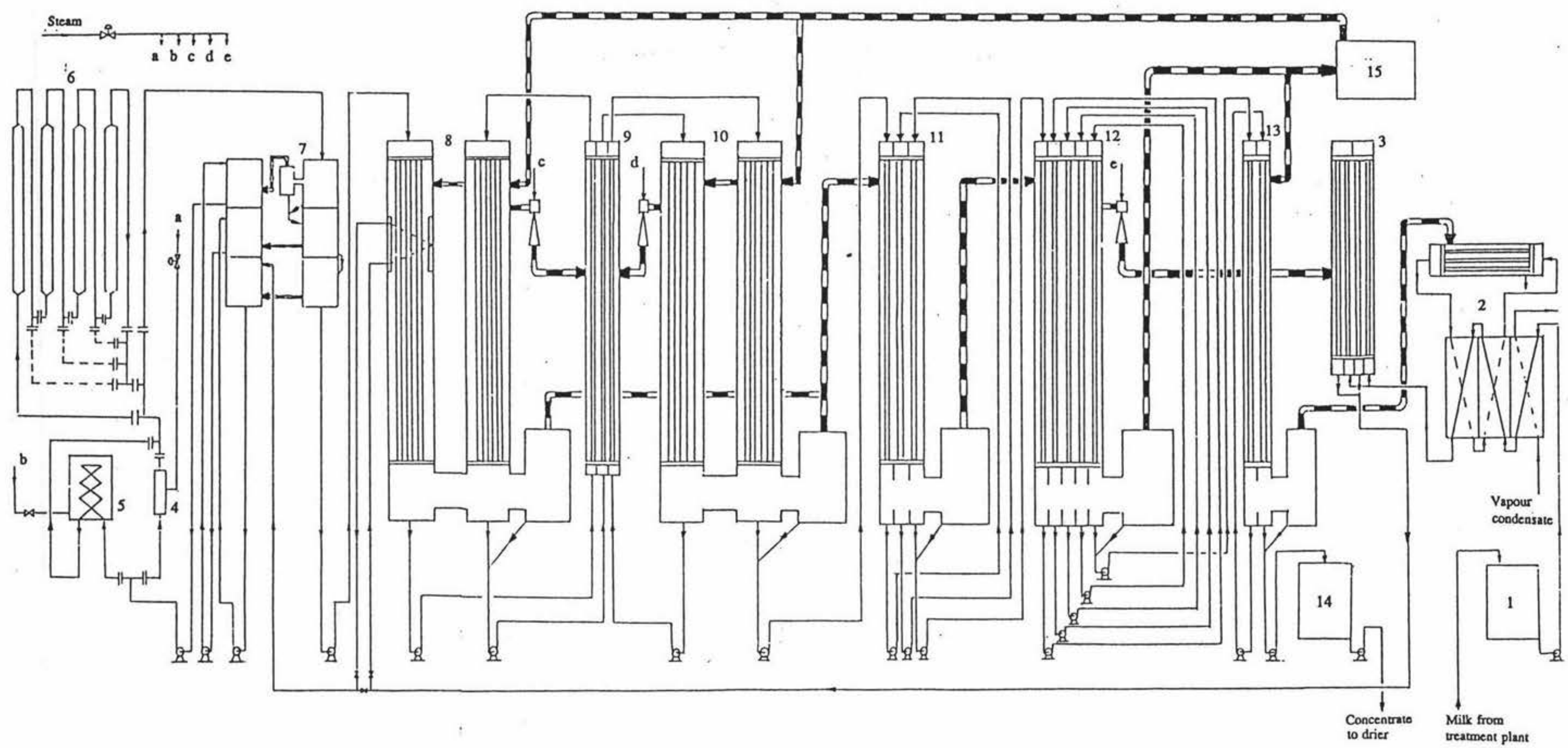
3.1.1.3 The three effect twelve pass milk evaporator

The flow-sheet of the three effect twelve pass milk evaporator with the mechanical vapour recompression (MVR) and a finisher is shown in Fig.1.

After measuring the diameter and length of the tubes and having counted the number of the tubes in each passes, the heat transfer area in each passes were determined. All these are presented in table 5.

Table 5. Dimension of heat transfer area in the 3 effect 12 pass evaporator

Effect number	Pass number	Length of tube(m)	Diameter of tube(mm)	Number of tube	Area (m^2)
1	1	7	32	742	522.16
1	2	7	32	518	364.53
1	3	7	32	478	336.38
1	4	7	32	440	309.64
2	5	11	42	268	388.98
2	6	11	42	216	313.51
2	7	11	42	186	269.96
3	8	11	42	213	309.15
3	9	11	42	157	227.87
3	10	11	42	123	178.52
3	11	11	42	96	139.34
3	12	11	42	84	121.92
finisher	13	7	32	74	50.08
finisher	14	7	32	50	35.19



1. Feed tank	4. Direct steam injector	7. Flash vessel	10. Second effect	13. Finisher
2. Heat exchangers	5. Indirect steam heater	8. First effect	11. Third effect	14. Concentrate tank
3. Preheater	6. Holding tube	9. Intermediater heater	12. Forth effect	15. MVR

Fig. 1 The flow sheet of the three effect twelve pass milk evaporator

3.1.1.4 Computation

A computer program prepared by Jebson (personal communication) written in Basic was used to carry out the calculation. The detail of the computer program with several allowances made for the calculation can be found in Jebson's papers (1988, 1990, 1991). Because there are 5 passes in the third effect (normally only 3 or 4 pass in one effect), the program was modified to handle this flow pattern.

After the mass balance and the heat balance were made, the steam consumption and the heat transfer coefficient for each pass or effect was calculated.

The velocity as well as the momentum of vapour in the tubes, the irrigation density, the Reynolds number are also calculated.

The measured values of viscosity of the concentrate were used in the calculation rather than values estimated by using formula proposed by Buckingham (1978) and Bloore & Boag (1981).

3.1.2. The results and discussions

(1) All results of calculation for the three effect twelve pass evaporator were shown in table 6.

Table 6. Results of calculation for the 3 effect 12 pass evaporator

Effect number	Pass number	T_E (°C)	ΔT (K)	Measured Solid(%)	U ($kW/m^2.K$)
1	1	68.9	3.5	9.699	1.280
1	2	68.9	3.5	10.639	1.474
1	3	68.9	3.5	11.353	0.829
1	4	68.9	3.5	11.934	0.968
2	5	65.1	3.8	13.448	1.995

2	6	65.1	3.8	15.488	1.541
2	7	65.1	3.8	17.726	1.946
3	8	59.4	5.7	21.547	1.357
3	9	59.4	5.7	25.567	0.971
3	10	59.4	5.7	29.896	1.379
3	11	59.4	5.7	36.193	1.710
3	12	59.4	5.7	41.091	0.492
finisher	13	52.4	7	43.821	0.740
finisher	14	52.4	7	45.660	0.711

Continue Table 6.

Pass number	Vapour velocity (m/s)	Vapour momentum (kg.m/s.m)	Measured viscosity (cP)	Γ (kg/m.s)	Re
1	6.035	0.15	0.561	1.457	10788.020
2	5.686	0.20	0.549	1.834	13867.220
3	3.200	0.06	0.635	1.746	11714.020
4	3.735	0.08	0.821	1.716	8771.614
5	11.095	1.01	0.661	1.543	11215.330
6	8.213	0.60	0.426	1.442	14057.000
7	10.373	0.96	0.445	1.275	11906.450
8	10.260	0.63	0.469	0.516	4578.367
9	6.797	0.32	0.484	0.497	4257.253
10	9.651	0.65	0.504	0.461	3811.429
11	11.963	1.00	0.613	0.400	2777.752
12	0.344	0.08	2.848	0.360	1225.393
13	13.978	0.41	6.163	0.763	534.034
14	11.255	0.38	104.686	1.043	308.075

(2) If this evaporator were considered to be a single three effect falling film evaporator, the steam consumption in this evaporator in term of kg steam used per kg evaporation would be 0.367.

(3) The heat transfer coefficients plotted against the pass numbers was shown in Fig. 2.

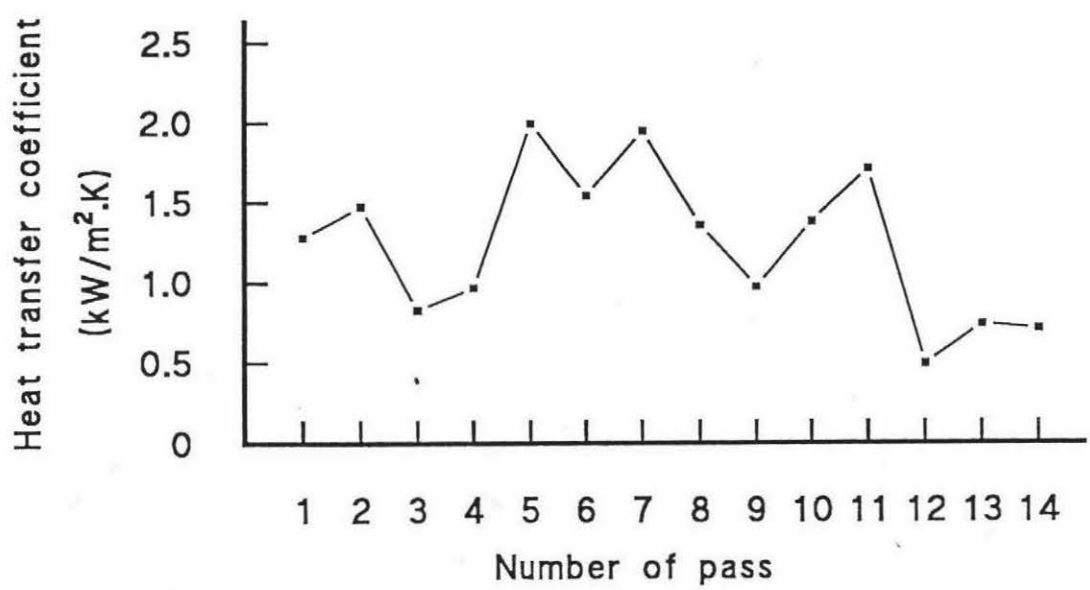


Fig. 2 Heat transfer coefficient versus the pass number.

It was calculated that the average heat transfer coefficient in the first effect is 1.138 kW/m^2.K, in the second effect is 1.827, in the third effect is 1.18.

(4) The regression equations were obtained by using the Minitab statistic package computer program and presented as follows:

$$U = 1.05 + 22.2\, M_v + 2.4\times10^{-5}\, Re - 0.103\, \Delta T \qquad (23)$$
$$(R^2 = 93.8\ \%)$$

$$U = 0.443 + 20.9\, M_v - 1.11\times10^{-3}\, \mu - 3.5\times^{-2}\, \Gamma + 5.0\times10^{-5}\, Re \qquad (24)$$
$$(R^2 = 93.3\ \%)$$

Where: M_v is vapour momentum, it is expressed as mass times velocity per meter tube perimetric length (kg.m/s.per m).

(5) It can be seen that the temperature difference had a negative effect on heat transfer coefficient and the momentum of vapour had a positive effect on the heat transfer coefficient. Both the viscosity and the irrigation density had a negative effect, but the Reynolds number had a positive effect. It is easy to understand the negative effect of viscosity, but not to irrigation density. From the results, it was found that the irrigation density in this evaporator was relatively higher, especially in the first effect, and hence the liquid falling film was thicker. Therefore the heat transfer coefficients were lower. If this is true, the heat transfer coefficients might be increased just by reducing the feed flow rate or increasing the heating area in the first effect, in turn reducing the irrigation density, in a result of increasing the heat transfer coefficients.

3.2 Whole milk evaporator data processing

There are 17 data sets of whole milk evaporator spread in the different dairy company in New Zealand available from students' report over 10 years. All these data entered into a computer program to evaluate the heat transfer coefficient.

The computer program used previously for skim milk evaporation was modified by using the formula proposed by Wood (1982) to estimate the whole milk viscosity at different solids levels.

By using the values of configuration of evaporator, the vapour momentum in the tube, the irrigation density and Reynolds number were calculated. But all values were not available, there are only 7 data sets to be used for regression.

3.2.1. Original data

The original data for regression came from following dairy plants:

R - Te Rapa;

W - Waitoa, NZCDC;

T - Te Awamutu, NZCDC.

3.2.2 The results and discussions

(1) For the whole milk evaporators, the heat transfer coefficients varied from 0.24 to 3.99, the vapour momentums varied from 22 to 0.004, the Reynolds numbers were in the range of 248-14680, the maxim temperature difference was 18 °C, the minim was 3 °C.

(2) The heat transfer coefficients regress to vapour momentum, Reynolds number and temperature difference for each data. The general form of regression equation is as follows:

$$U = a + b\, M_v + c\, Re + m\, \Delta T \tag{25}$$

The summary of regression equations are presented in table 7.

Table 7. Summary of regression equations for whole milk evaporators

Factory & year	R^2	F	a	b	c ($\times 10^{-4}$)	m
R85	100	4	4.22	3.32	0.24	-0.246
R88	99.1	4	3.13	-4.45	1.73	-0.137
R85	99.8	4	3.85	5.32	1.16	-0.29
W83	99.5	4	1.40	9.43	0.36	-0.138
W86	94.2	5	3.60	-16.3	1.18	-0.270
T90	95	10	1.90	9.86	0.2	-0.252
T90	96.7	10	1.84	8.31	0.26	-0.232

(3) Expecting two data sets, a positive effect of vapour momentum on the heat transfer coefficient could be seen. As the vapour generated from the falling film and travelled down the tubes, the influence of vapour on the falling film flow could be expected. It is not hard to image a high wind blowing over a water

surface and causing waves (Jebson, 1988). This "wind over water" effect on the falling film creating or increasing the turbulence in the film and hance increasing the heat transfer coefficient.

(4) A positive effect of Reynolds number and a negative effect of temperature difference on the heat transfer coefficient can be found out from the equations. These effects will be discussed more detail in the Chapter 5.

Chapter 4

Experimental

Although the data obtained from commercial evaporator have given general information about the performance of falling film evaporators in the New Zealand dairy industry, they were not enough to explain the precise performance of the falling film evaporators and the phenomena occurring in the heating tubes. Because the spot samples were taken from the evaporator, which cannot give a clear picture of the whole system. The measurements had to be made at different times in the course of operation, as there would be different degrees of deposition of milk on the heating tubes leading to inaccuracies in the results. Therefore, a pilot scale single-tube falling film evaporator was used for further experimental work to gain more understanding of the heat transfer in the falling film evaporator.

4.1 Objectives

The objectives of this part experimental work are to assemble, debug, operate the single-tube falling film evaporator and to evaluate the effects of operating variables on overall heat transfer coefficients in this falling film evaporator.

4.2 The pilot evaporator

In order to save apparatus' investment, the pilot scale single-tube falling film evaporator made of stainless steel was linked to a Centri-Therm evaporator (CT1B-2, ALFA-LAVAL), which already exited at the pilot plant. That means, excepting of the heating column and vapour chamber and temperature measurement systems, all parts of apparatus used the Centri-Therm's equipment, i.e. feed pump, concentrate pump, condensate pump, preheater, condenser, vacuum unit, and steam regulation system. The whole apparatus is shown in Fig. 3. Fig. 4 shows the schematic diagram of the single-tube falling film evaporator.

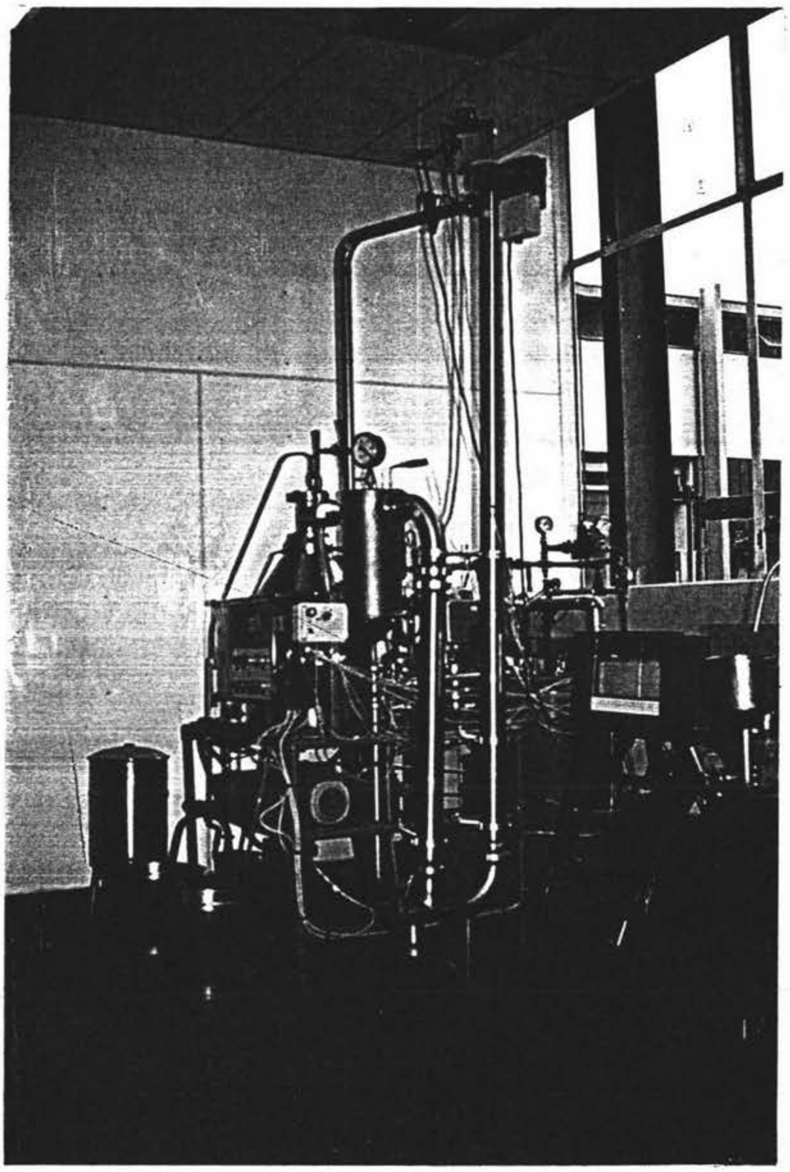


Fig. 3 The whole apparatus of the single-tube FFE

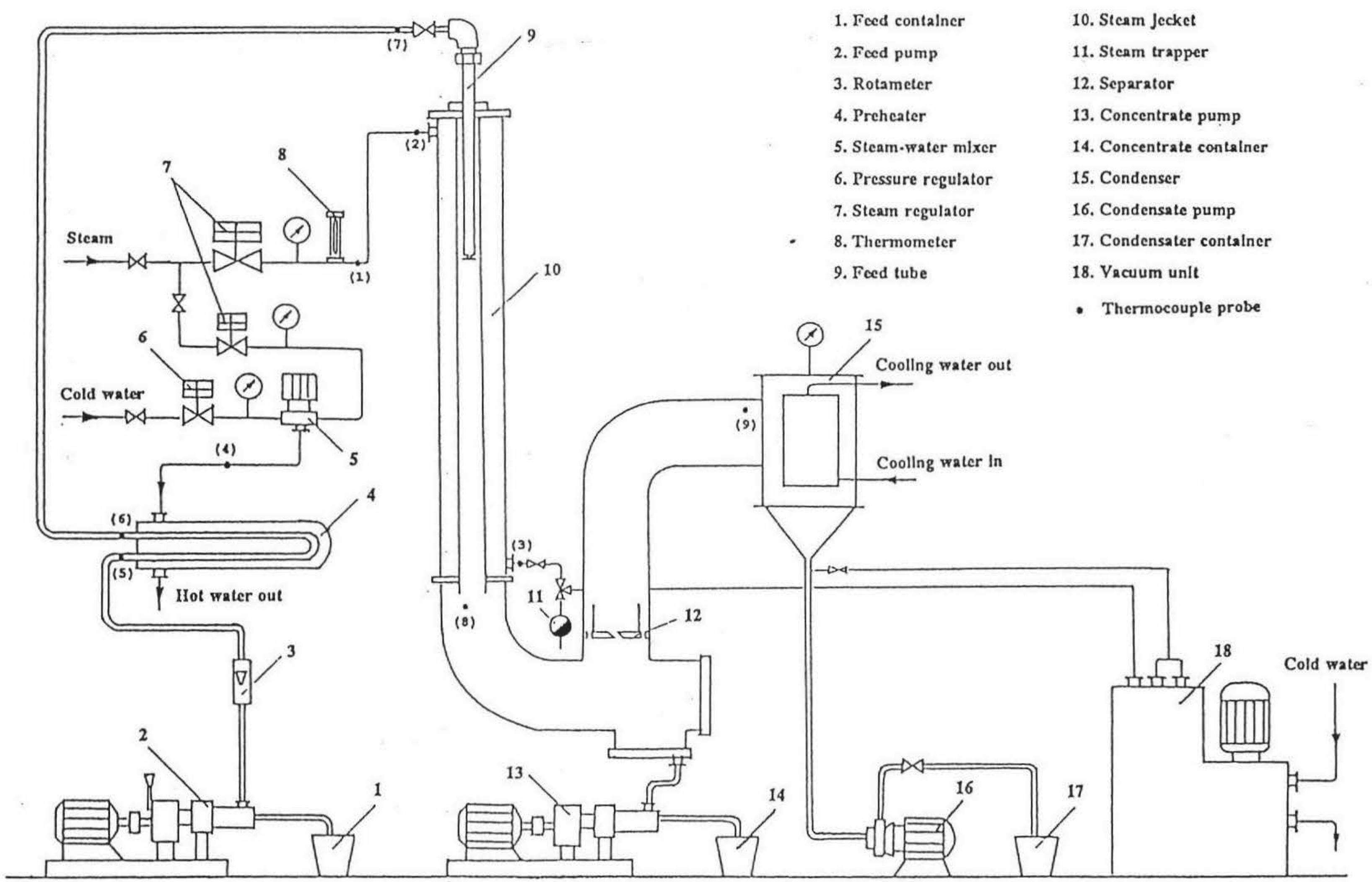


Fig. 4 The schematic diagram of the single tube falling film evaporator

The heating column contains a 32.00 mm O.D., 1.6 mm thick, 2 meter length heating tube, which was coaxially placed inside a 73 mm I.D., 2 meter length tube. Therefore, a steam jacket was formed between these two tubes. The whole heating area referring to outside surface of heating tube is 0.206 m^2.

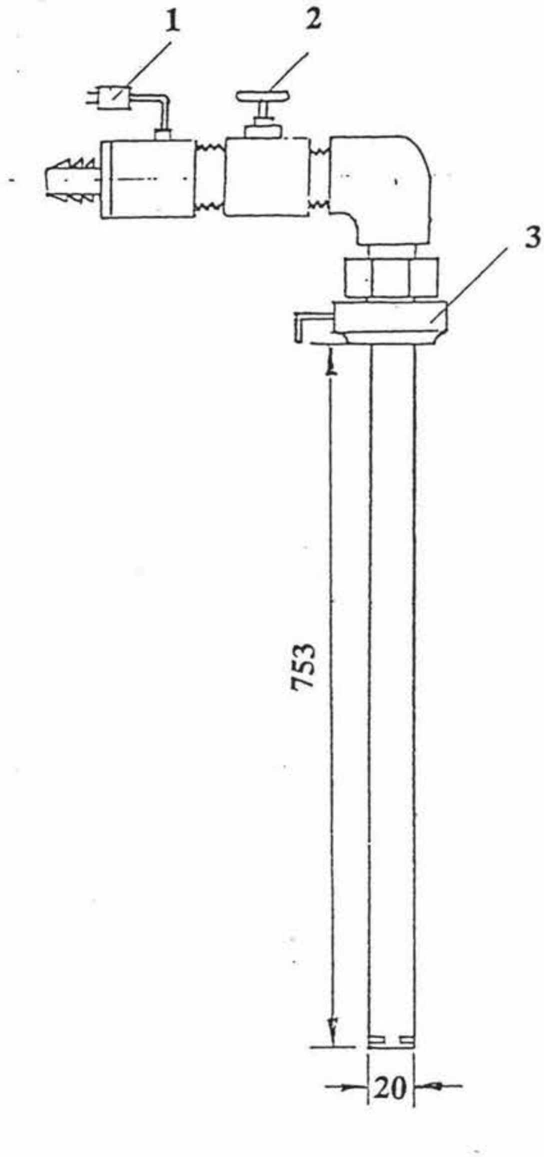
Another 20 mm O.D., 0.75 meter length tube (feed tube) was centrally inserted into the heating tube to act as a distributor. The tube was plugged up at the bottom with three slots (2 mm wide). The structure of the feed tube is shown in Fig. 5. This feed tube was adjustable on the evaporator so the heat transfer area on the heating tube can be varied.

Details of design calculation of the heating column are presented in the appendix I.

In order to rotate the heating tube for further experiment, the problem of sealing the steam jacket became more serious. After several trials, a stainless steel house which housed a seal was made and a teflon strip of about 0.7 mm thickness by 20 mm wide was used for a bearing.

A angled slots separator was mounted in the vapour clamber. The angled slot made the vapour spin when the vapour passed it, which provided the centrifugal force for the vapour. Therefore, the liquid particles entrained in the vapour could be directed against the wall of vapour chamber. A weir above the separator provided the static heat for the accumulated liquid and let the liquid flow down to concentrate. The structure of the angled slots separator was shown in Fig. 6 and the design calculation of it can be found in the appendix II.

The liquid to be evaporated was pumped by a variable speed feed pump (mono pump type 1) through a Spiraflo heat exchanger (Model TT 0.75/0.5-3), where the liquid was heated up by hot water to evaporating temperature, and then into feed tube. After passing through the feed tube, the liquid flowed down the inside of heating tube as a thin evaporating film. The heat issued from the inside of heating tube wall to make a portion of the falling film evaporate. The unvaporized liquid and vapour flowed straight to the bottom of the vapour chamber where the unvaporized liquid was pumped out by



1. Thermocouple probe
2. Valve
3. Fixer

Fig. 5 The stucture of feed tube

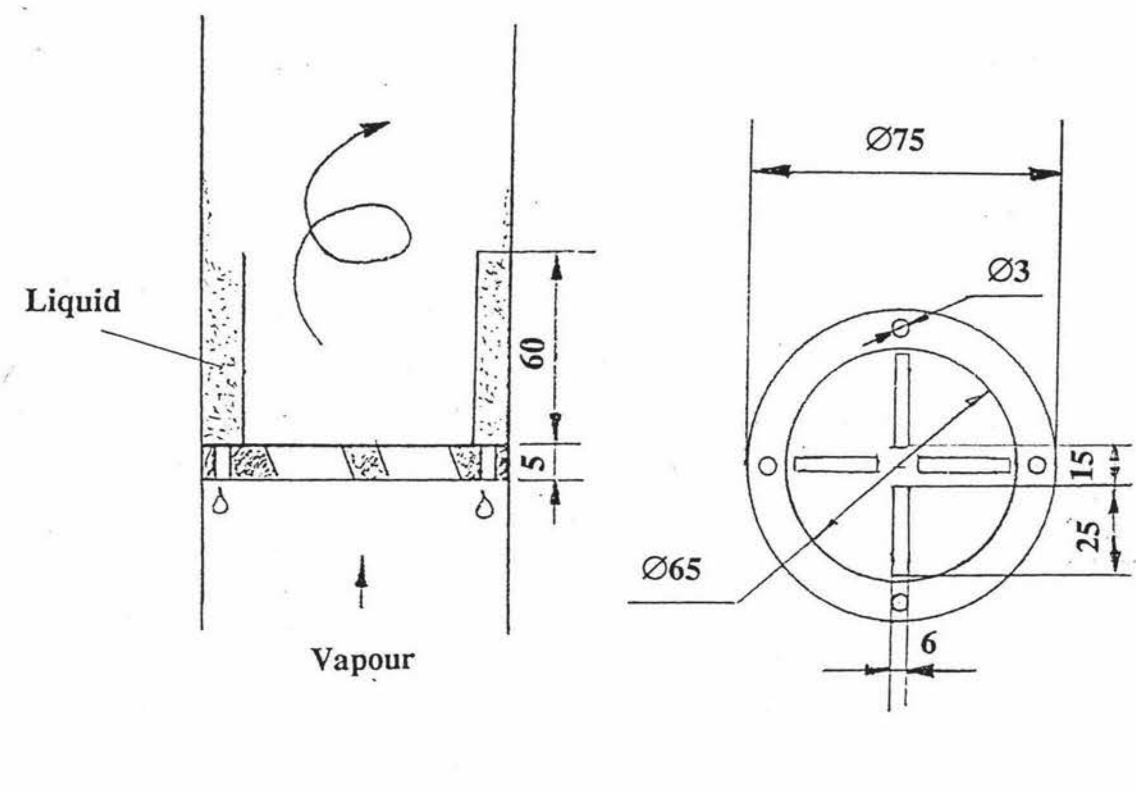


Fig. 6 The structure of the angled slots separator

a concentrate pump (mono pump type 1) and collected in the bucket. The vapour passed through a angled slot separator and into a spiral condenser, where the vapour was condensed. The condensate was pumped out by a centrifugal pump (type GM) and was collected in the bucket for measurement. Cold tap water flowing inside the spiral tubes of the condenser provided the heat sink for the condensation.

Steam was fed to the top of the steam jacket. The heat flux which was released by condensing steam outside of the heating tube transferred to heating tube wall and conducted through the wall. The steam condensate flowed down the outside of the heating tube and was removed by means of an ejector in the vacuum unit from the bottom of the steam jacket.

The hot water used to heat up liquid in the Spiraflo heat exchanger was from a hot water mixer (Type MK 4A. Serial No.A) into which steam and cold tap water were fed. The pressures of steam and cold water were controlled by means of a steam regulator valve (Type GD-30) and a pressure regulator valve (Type E, Series 3) respectively.

Vacuum was provided by an Vacuum unit (Type VB 2-3). This unit consists of a container for water, a circulation pump and three ejectors with built in check valves. When the water was pumped by the circulation pump through the ejectors, a vacuum was created. One of the ejectors was used to remove condensate of steam from steam jacket. Other two were used to remove non-condensables from evaporating side. (One of them used to supply vacuum to steam regulating valve, which is supplied now with the centre vacuum line).

The vacuum can be adjusted manually for operation and released at the end of the operation by means of a vent cock on the venting line.

The heating section and vapour stream tube lines were assembled with removable clamps and rubber gaskets. The liquid flow lines were made of the transparent plastic hose. The system was carefully sealed and checked for air leakage every time before each run.

4.3 Instrumentation

Steam temperature as well as pressure were controlled by a steam regulating valve (type SRV-5) with a temperature regulator. The steam regulating valve was mounted with a vertical rod and a valve housing uppermost. The coil spring aimed at pressing the valve cone against the valve seat, but the vacuum in the vacuum champer below the rubber membrane aimed at opening the valve. There were two connections at the bottom valve, one could communicate with a service source (the vacuum), another one could communicate with the air nozzle in the temperature regulator, which determined the size of the servo vacuum, and thereby the degree of opening of the valve. The air nozzle in the temperature regulator was mounted in a water proof aluminium box, and a balance lever was pressed against it. Springs were fitted at both ends of lever. Expansion bellows were also fitted to the lever and they were connected to the steam line at the position after the steam regulating valve. When the steam pressure increased with a increased temperature, the bellows expanded, which caused the balance lever to be lifted. Then the air flowed through the air nozzle increased, the vacuum in the chamber would reduce, which resulted in the valve throttling the steam supply. After being set at required temperature on the seal, the regulating system would maintain the temperature in the steam jacket.

The adjustment of the steam regulating valve as well as the temperature regulator and more detail information can be found in the ALFA-LAVAL Centri-Therm instruction book.

The inlet feed flow rate was measured by means of a stainless steel variable area rotameter with a capacity of 0.4-4.5 l/min. of water at 20 °C. The rotameter was calibrated by discharge weight versus time measurements at operating temperature.

Temperatures were measured with iron/constantan thermocouple probes (type J), which were connected to a recorder (32 channel recorder/dataiogger, MOLYTEK). The thermocouple probes were monitored at nine different points, i.e. No. 1 measured the steam temperature after regulating; No. 2 measured the steam temperature entering the steam jacket; No. 3 measured the steam condensate temperature; No. 4 measured the hot water temperature. No. 5 and 6 measured the feed temperature at inlet and outlet preheater; No. 7 measured the feed temperature just before entering the feed tube; No. 8 measured the concentrate temperature; No. 9 measured the vapour temperature. All these are indicated in Figure 1. Prior to starting the experiments, all thermocouples were calibrated in a constant temperature bath at boiling and freezing points, and intermediate points against a standard mercury thermometer. The recorder plotted all measured temperatures on the chart in 10 second intervals.

A vacuum gauge was used to monitor vapour pressure. Two steam pressure gauges were used to monitor steam line pressure for heating column and hot water mixer respectively. A water pressure gauge was used to monitor water line pressure to the hot water mixer. The positions of gauges are shown in Fig. 3.

4.4 Variables

As reviewed previously, there are many variables affecting heat transfer in the falling film evaporator. Based on the results of commercial milk evaporator test and literatures (Agarwala, 1975, Kroll & McCutchan,1968, Sinek & Young,1962), following variables were selected for study in this experiment:

(a) Overall temperature difference;
(b) Evaporating temperature;
(c) Heating tube length;
(d) Reynolds numbers.

Scaling of heating surface cannot be avoided while evaporating milk. During the course of running, more deposits will be built up on the surface resulting in lowering of the heat transfer coefficient. Therefore, tap water and sugar solution were used rather than milk for convenience and to minimize the scale deposition problem.

4.5 Selection of conditions

After the apparatus was assembled, it took a lot of time to debug the equipment, especially for measurement of temperature. Because the temperature difference was used to calculate the heat transfer coefficient, the accurate measurement is critical.

After many runs were done to test the capacity of this evaporator, the ranges of variables were chosen.

The temperature difference range used in commercial evaporator usually fall into 3 to 16 °C, so the temperature difference range was selected to be 3 to 18 °C in this experiment. The evaporating temperatures range was selected to be 70 to 90 °C, because the vacuum chamber is too small to use at lower evaporating temperature. Otherwise the entrainment would be excessive.

Due to the capacity of the rotameter and the concentrate pump, the inlet flow rate range was limited from 400 to 1400 ml/min. Because of the height of ceiling, the heating tube length was limited below 2 meters. So the lengths used were 1.2, 1.5 and 2 meters.

The ranges of experimental variables are listed in table 8.

Table 8. Ranges of experimental variables

Variable		Range
Flow rate	(ml/min)	400, 600, 800, 1000, 1200, 1400
Evaporating	(°C)	70, 80, 90
Heating tube length	(m)	1.2, 1.6, 2
Temperature difference	(°C)	3, 5, 8, 10, 15, 18
Feed liquid	(No.1)	Tap water
Feed liquid	(No.2)	10% sugar solution

4.6 Operating procedure of equipment

4.6.1 Starting

(1) Close all valve on the line and set the feed tube at correct position;

(2) Turn on water to the condenser and the vacuum unit;

(3) Turn on the temperature recorder;

(4) Start the vacuum pump;

(5) Check the sealing of steam jacket and evaporating side (check that pressure P = -95 kPa);

(6) Start the feed pump and check that the product enters the evaporator, adjust the feed flow rate to the desired value;

(7) Start the condensate pump, adjust the vacuum of evaporating side to corresponding regulating temperature;

(8) Open the drain valve and drain condensate from the steam supply line, close the drain valve;

(9) Open the steam valve slowly, adjust the steam pressure on the regulator until the correct temperature is obtained;

(10) Open the cold water valve to hot water mixer and then open the steam valve to it, adjust the hot water temperature to the desired value, which is correspond to feed temperature just before entering feed tube.

4.6.2 Cleaning

(1) Release the vacuum in the evaporating side;

(2) Rise the equipment with tap water;

(3) Use 0.3-1% Acid solution to recirculate for 10 minutes;

(4) Rise with tap water;

(5) Use 0.05-0.1% alkali solution to recirculate for 10 minutes;

(6) Rise with tap water.

4.6.3 Stopping

(1) Set the steam regulator at minium;
(2) Close valves on the steam line and the cold water line to the hot water mixer;
(3) Close the steam supply valve;
(4) Release the vacuum in the evaporating side;
(5) Stop the vacuum pump;
(6) Stop the feed pump and concentrate pump;
(7) Stop the condensate pump;
(8) Close for cooling water to condenser and vacuum pump;
(9) Switch off the temperature recorder.

4.6.4 Operating caution

(1) Never run the feed pump or the concentrate pump without liquid. If allowed to run dry, the rubber stator in the pump will be destroyed;
(2) Maxim steam pressure before the steam regulating valve is 600 kPa, maxim steam pressure after the steam regulating valve is 100 kPa. corresponding to 120 °C;
(3) The condenser should be supplied with enough cooling water to keep the outgoing water temperature 10 °C below the evaporation temperature.
(4) Always keep eyes on temperature recorder during the course of running, because adjustment will be needed when desired temperature shaft due to the fluctuation of steam pressure and cold water pressure.

4.7 Experimental procedure

In order to find the effect of individual variables on the heat transfer coefficients, only one parameter was varied during each test, while other parameters were kept constant.

A typical procedure was as follows:

(1) Starting the evaporator, and setting the steam temperature, evaporating temperature, feed flow rate, feed temperature, and heating tube length at the desired values.

(2) After about five minutes, when a steady state had been reached, the flow rate of vapour condensate were measured by means of bucket-stopwatch. The measurements and readings were taken during three minutes periods at five minutes intervals during each test run. That means that the flow of condensate was diverted to a empty bucket and at the same time a stopwatch is started. After 3 minutes, the amounts of condensate in the bucket was measured. For each run, three times measurements were usually made. If the error between the each measurement was greater than about 5%, further measurements were made.

(3) During the course of the run, all temperatures were recorded on the temperature recorder. All data were recorded on a log sheet.

The temperature of the feed was very difficult to control due to fluctuation of the pressure of cold water and steam in the steam-water mixer. It was even more difficult when the feed temperature was low entering the preheater. So the concentrate and the vapour condensate were feed back to feed tank after their weights were measured. This procedure also kept the concentration of feed constant.

The feed temperature was kept ± 1 °C around evaporating temperature during each measurement. The temperature difference between probes No.8 and No.9 was kept less than 0.5 °C. If this temperature difference was greater than 0.5 °C, which indicate that the concentrate might be accumulated at the bottom of the vapour chamber, and that adjustments should be made.

4.8 Data processing

Mass balance of the feed, concentrate and condensate was down at beginning of each run to check proper vapour condensation.

Based on the measured feed rate and vapour condensate flow rate, irrigation density as well as Reynolds number at the top and bottom of the heating tube were computed.

The viscosity of sugar solution was computed according to the equation proposed by Campanella (1991).

The heat transfer coefficients were computed based on the condensate of vapour and the overall temperature different (the temperature difference between probes No.3 and No.8). U value is calculated from the following formula:

$$U = \frac{(\frac{W}{v})\ h_v}{A\ \Delta T} \qquad (26)$$

Where: W is vapour condensate (l/s);

v is specific volume of liquid (l/kg);

h_v is latent heat of vapour condensation, which could be found out in the steam tables (J/kg).

An example of calculations is presented in the appendix III.

"As easy as" spread sheet and Minitab computer programs were used to do all data processing.

Chapter 5

Results and Discussions

5.1 The summary of experimental results

The typical experimental results are summarised in table 9, the others are attached to the appendix V.

Table 9. Summary of experimental results

L (m)	T_E (°C)	ΔT (°C)	V (ml/min)	Liquid (No.)	Γ (kg/m.s)	W (ml/min)	U (kW/m^2.K)
2	70	8	1000	1 2	0.170 0.194	111 90	2.63 2.24
2	80	8	1000	1 2	0.168 0.192	119 106	2.78 2.55
2	90	3	1000	1 2	0.173 0.192	58 55	3.59 3.53
2	90	5	1000	1 2	0.170 0.195	88 80	3.27 3.08
2	90	8	1000	1 2	0.167 0.191	122 117	2.83 2.82
2	90	10	1000	1 2	0.165 0.189	150 140	2.79 2.70
2	90	15	1000	1 2	0.159 0.183	212 193	2.63 2.48
2	90	18	1000	1 2	0.159 0.180	253 227	2.61 2.43
2	90	10	1200	1 2	0.201 0.229	140 134	2.60 2.58
1.6	70	8	1000	1 2	0.172 0.194	95 87	2.81 2.62
1.2	70	8	1000	1 2	0.173 0.195	80 73	3.15 2.93

5.2 The falling film configuration

As the liquid entering the evaporating side at its evaporating temperature, there was neither heating area needed to heat up liquid to evaporating temperature nor flash. After passing through the narrow slots on the feed tube, the liquid sprayed onto the heating tube and fell down as a thin evaporating falling film. At the entry the film flow regime would be highly turbulent due to the high velocity of the liquid sprayed from the feed tube. After some distance, the liquid film would become stable and maybe have a wavy laminar flow regime. As liquid film fell down, continued addition of heat caused more and more vapour to be generated. The liquid film decreased in thickness as a result of both vaporization and further fluid acceleration. The increasing vapour velocity caused the film flow to become turbulent. So at the bottom the film flow regime would be turbulent once again. If the temperature difference was big enough the nucleate sites might appear on the tube wall and bubbles would present in the falling film. From the top to the bottom of heating tube the liquid film thickness became thinner and the liquid concentration became higher gradually, the film velocity and vapour velocity are increasing progressively. Eventually dry spots maybe begin to appear on the tube wall.

The film flow configuration is drawn in Fig. 7.

5.3 Film coefficients

The condensing side mechanism was observed to be of the dropwise condensation type. As mentioned in chapter 2., this type of condensation has a film heat transfer coefficient 6 to 18 times that of the film condensation type (Perry, 1984). Under the conditions of this experiment, the film heat transfer coefficient was calculated to be of order 2-6 $kW/m^2.K$ (see appendix IV). Hence the measured overall heat transfer coefficient will closely approximate the evaporating side film coefficients.

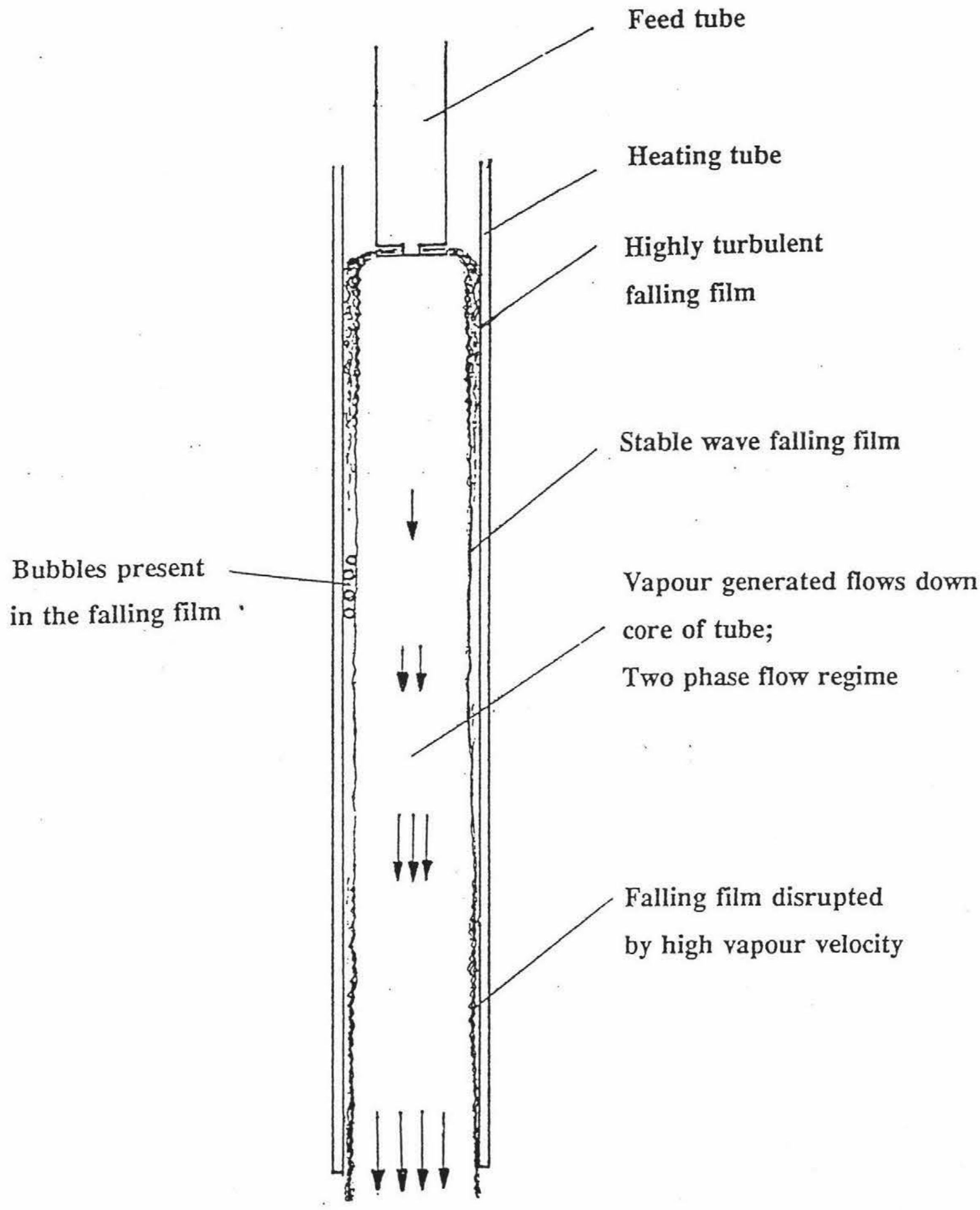


Fig. 7 The diagram of falling film configuration

5.4 The effect of temperature difference

From the heat transfer equation ($q = U \Delta T$), it can be seen that the heat transfer rate (q) depends on the overall heat transfer coefficient and the overall temperature difference. The higher the overall heat transfer coefficient, the bigger overall the temperature difference, the greater the heat transfer rate. But the over heat transfer coefficient and the overall temperature difference are not independent parameters, the overall temperature difference has a great influence upon the overall heat transfer coefficient. It was found in commercial evaporator trials, there was a tendency of increasing the overall heat transfer coefficient with decreasing the overall temperature difference (Jebson, 1988, 1991). It is true in this experiment. From the Fig. 8 and 9, it is observed that a definite increase in the overall heat transfer coefficient with decreasing temperature difference. Fig. 8. shows typical curves,which are of the overall heat transfer coefficient versus the temperature difference, for 10% sugar solutions and water. Fig. 9 shows that overall heat transfer coefficient rose with lowering temperature difference for 10% sugar solution at different evaporating temperatures.

It can be seen from Fig. 8 and 9 that the overall heat transfer coefficient reducing rapidly in the range of temperature difference 3-8 °C, and then more slowly. The results suggest that there is a change of mechanism of vapourization in the falling film.

As mentioned in the literature review, there are four mechanisms of vapour evaporation from a falling film. I think there are three mechanisms involved in ranges used in this experiment. The vapour-liquid interface evaporation would occur in the range of temperature difference 3-5 °C. The bubble formation may commence at about 5 °C. Chun and Seban (1971, 1972) found that a superheat of 3.7 °C was required for bubble formation in water at atmospheric pressure, and that the necessary superheat increased as pressure decreased. Therefore, the low rate nucleate evaporating would occur in the range of temperature difference 5-8 °C. After the temperature difference is greater than about 8-10 °C, the high rate nucleating evaporating might be the dominant mechanism. Angeletti (1983) pointed out that the mechanism of direct liquid-vapour interface evaporation prevails when the total temperature difference is less 10 °C. These results

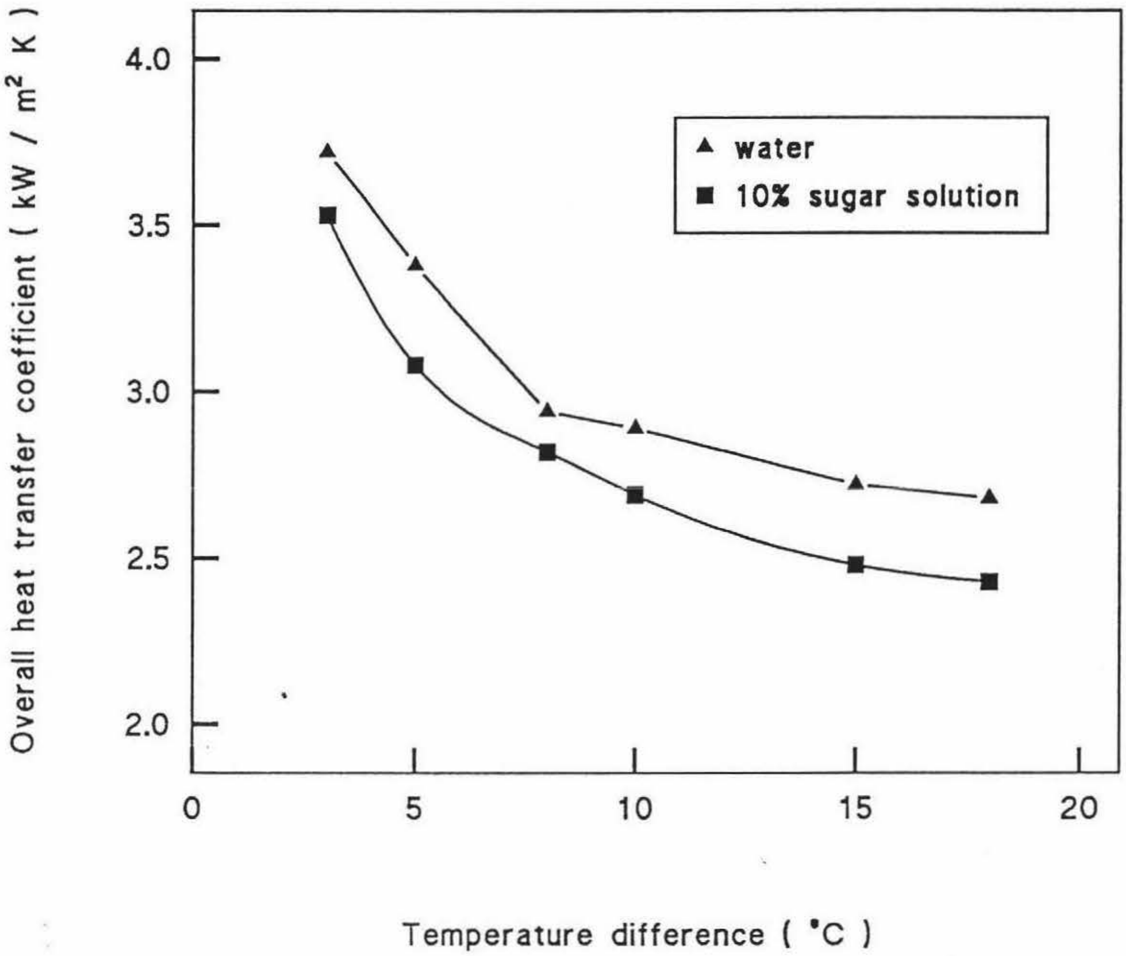


Fig. 8 Effect of temperature difference on overall heat transfer coefficient for water and 10 % sugar solution

Evaporating temperature: 90 °C, Feed flow: 1000 ml/min

Heating tube length: 2 m

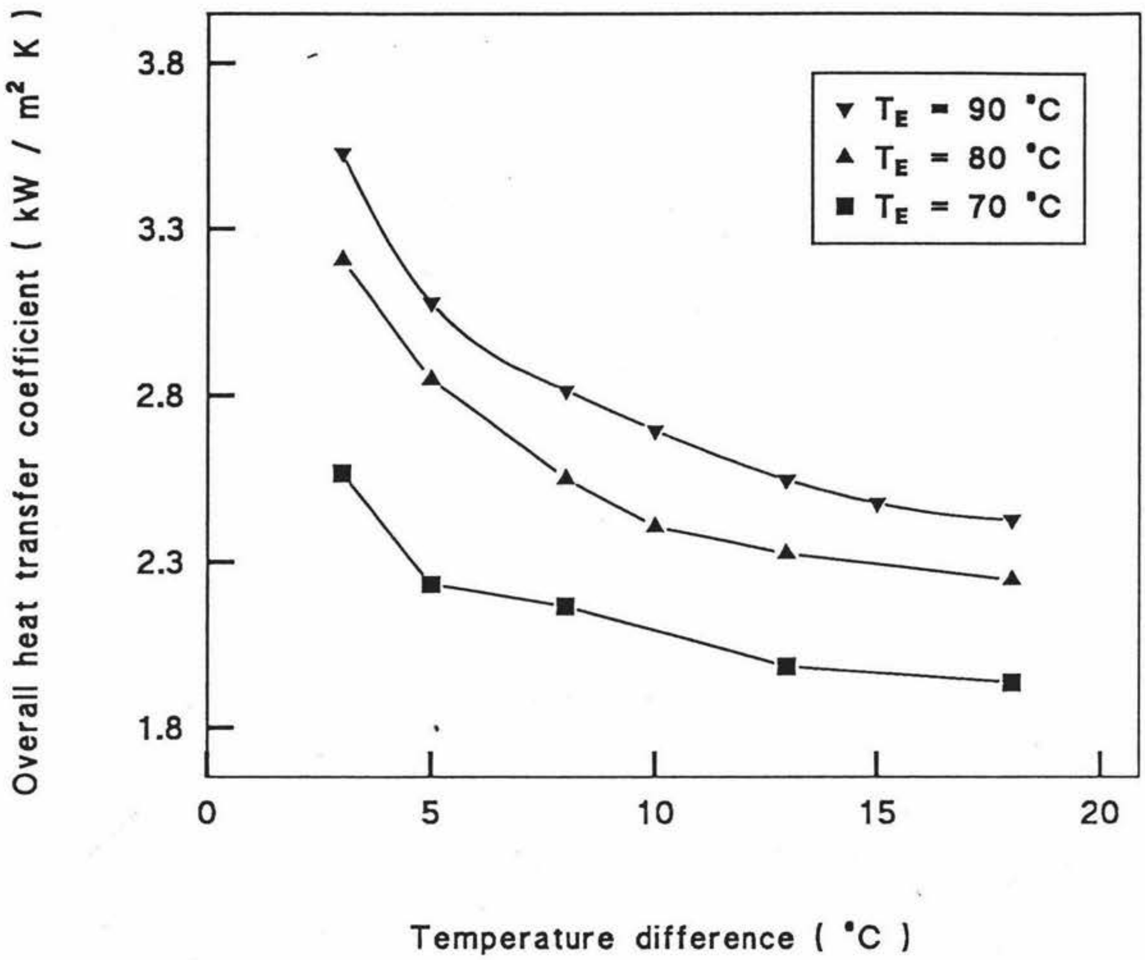


Fig. 9 Effect of temperature difference on overall heat transfer coefficient for 10 % sugar solution at different evaporating temperatures.

Heating tube length: 2 m, Feed flow: 1000 ml/min

would be probably explained as that the presence of bubbles would greatly increase the film thickness, decreasing the rate of heat transfer to the liquid-vapour interface. Although the presence of bubbles would make the liquid turbulent, the contribution of direct liquid-vapour interface evaporation (which is the more effective) would rapidly diminish with increasing temperature difference until bubble formation was the main effect. At this stage the heat transfer coefficients would decrease only slowly with increasing temperature difference because the temperature difference exceed the degree of superheat required to induce nucleate boiling. The number of nucleation sites apparently does not increase with increasing temperature difference.

Steiner and Ozawa (1982) stated that bubble formation reduced heat transfer benefits induced by high irrigation densities and that turbulence promoters were not likely to improve heat transfer significantly in falling film evaporation. This opinion arrears to be supported by the results presented here.

Low temperature differences are desirable when there is interest in multiple effect operation, and the total available temperature difference is not large. Commercial falling film evaporators used in the dairy industry usually operate at low temperature differences, in the range of 3-10 °C per effect. Therefore, higher heat transfer coefficients at lower temperature differences would be an additional advantage for multiple effect evaporation.

5.5 The effect of evaporating temperature

Typical examples of the measured effect of evaporating temperature on overall heat transfer coefficient are shown in Fig. 10. The basic trend is the same for water and sugar solution, the over heat transfer coefficient increasing as evaporating temperature increased, but the rate of increase for the sugar solution is greater than for water. The reason may be the higher viscosity of the sugar solution. As the temperature was increased the Reynolds number increases because of the decrease in viscosity. The viscosity decrease was a little larger for sugar solution than for water. Fig. 11 shows the effect of evaporating temperature on overall heat transfer coefficients for 10% sugar

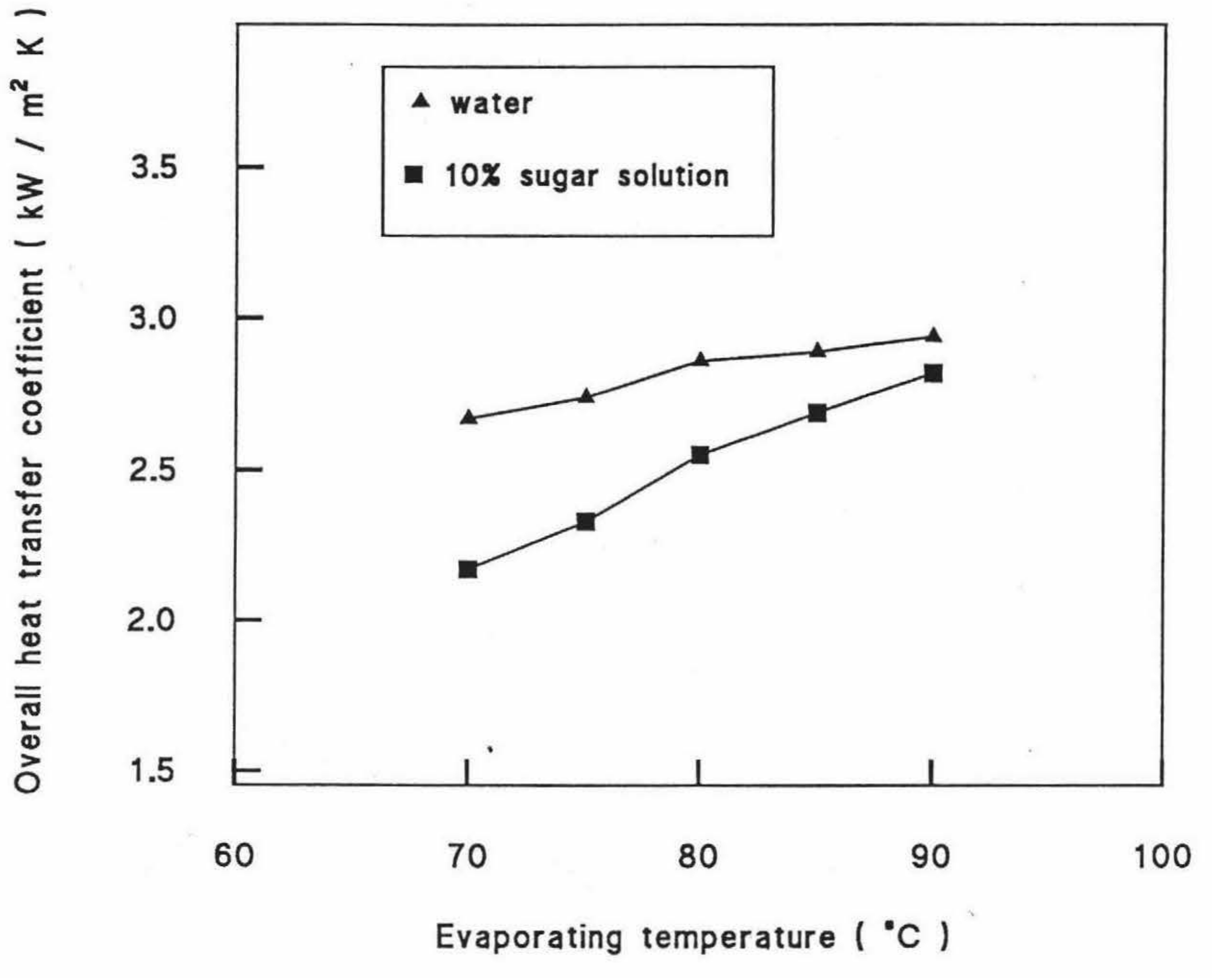


Fig. 10 Effect of evaporating temperature on overall heat transfer

coefficient for water and 10 % sugar solution

Temperature difference: 8 °C, Feed flow: 1000 ml/min

Heating tube length: 2 m

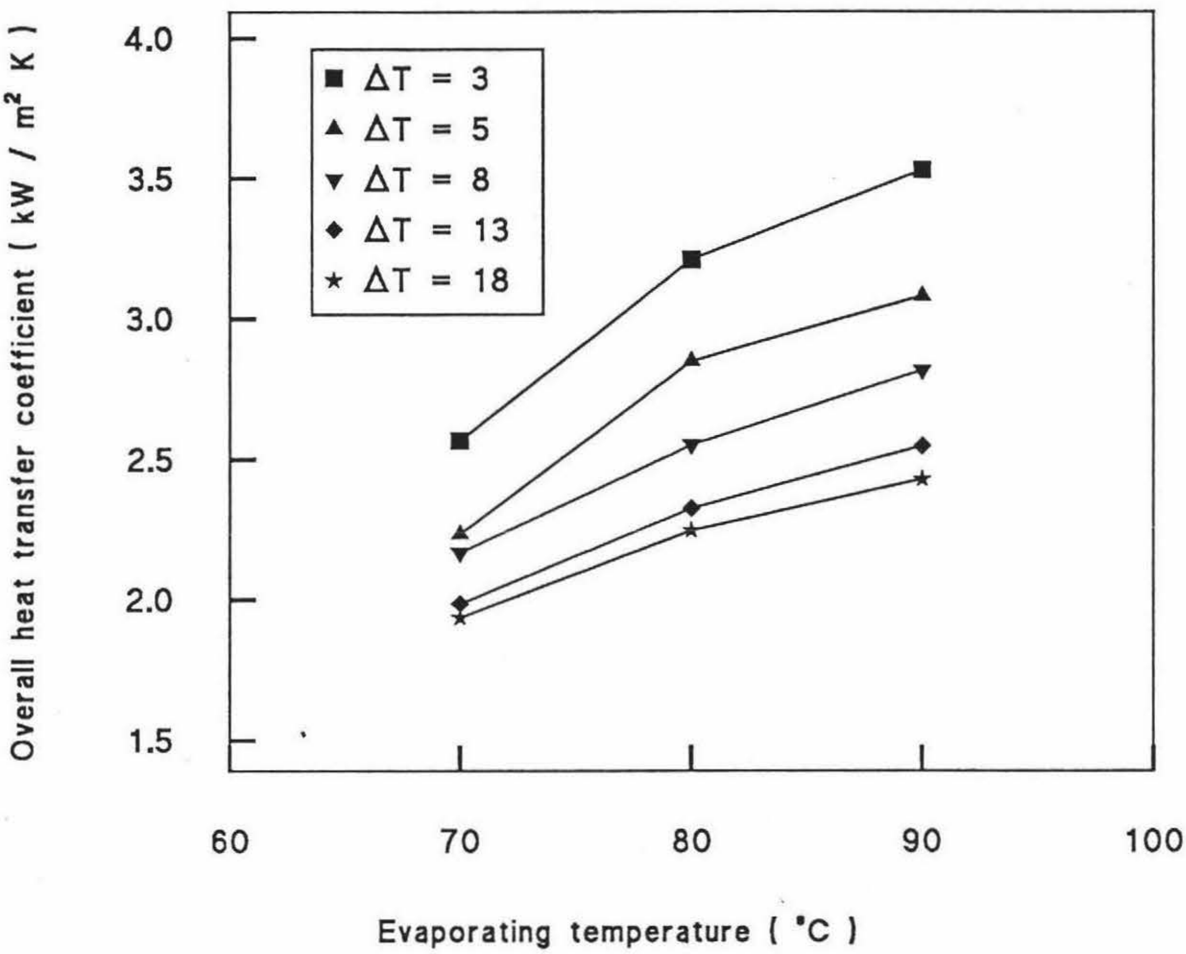


Fig. 11 Effect of evaporating temperature on overall heat transfer coefficient for 10 % sugar solution at different temperature differences.

Heating tube length: 2 m, Feed flow: 1000 ml/min solution at different temperature differences. It can be seen that the rate of increase is higher for low temperature difference than for high temperature difference. This is because vapour-liquid interface evaporation occurs at low temperature differences and the heat transferred from the heating tube wall would mainly conducted through the falling film, while the bubbles formation occurs at high temperature different and the most heat convected to the falling film. So the decrease of viscosity induced by increasing evaporating temperature would be more significant for the vapour-liquid interface evaporation than for the nucleate evaporating.

As the specific volume of vapour is reduced at higher evaporating temperatures, the vapour velocity is decreased, thereby affecting the film thickness which in turn could affect the individual heat transfer coefficients. But this effect was not observed in the pilot scale evaporator. It may be due to the only a small amount of vapour generated from the falling film as the tube was short. So there was non significant influence on overall heat transfer coefficients.

5.6 The effect of heating tube length

At initial stage, it was thought that the longer the heating tube length, the higher heat transfer coefficient because the vapour generated in the long tube should affect the film thickness much more. In fact, the reverse result was obtained in this experiment.

Fig. 12 and 13 shows that the overall heat transfer coefficient decreases with increasing heating tube length. This may be due to the entry effect. As described before, the film flow regime at the entry will be highly turbulent. After some distance, the liquid film will have a lower but stable Reynolds number. The heat transfer coefficient near the entry must be higher. Therefore, the entry effect in a short heating tube is greater than in a long heating tube.

The mechanism of liquid distribution on the top of heating tube in commercial evaporators is different. The distributor plate used in commercial evaporators to let liquid drip onto the plate between the heating tubes and then liquid flowed down on the

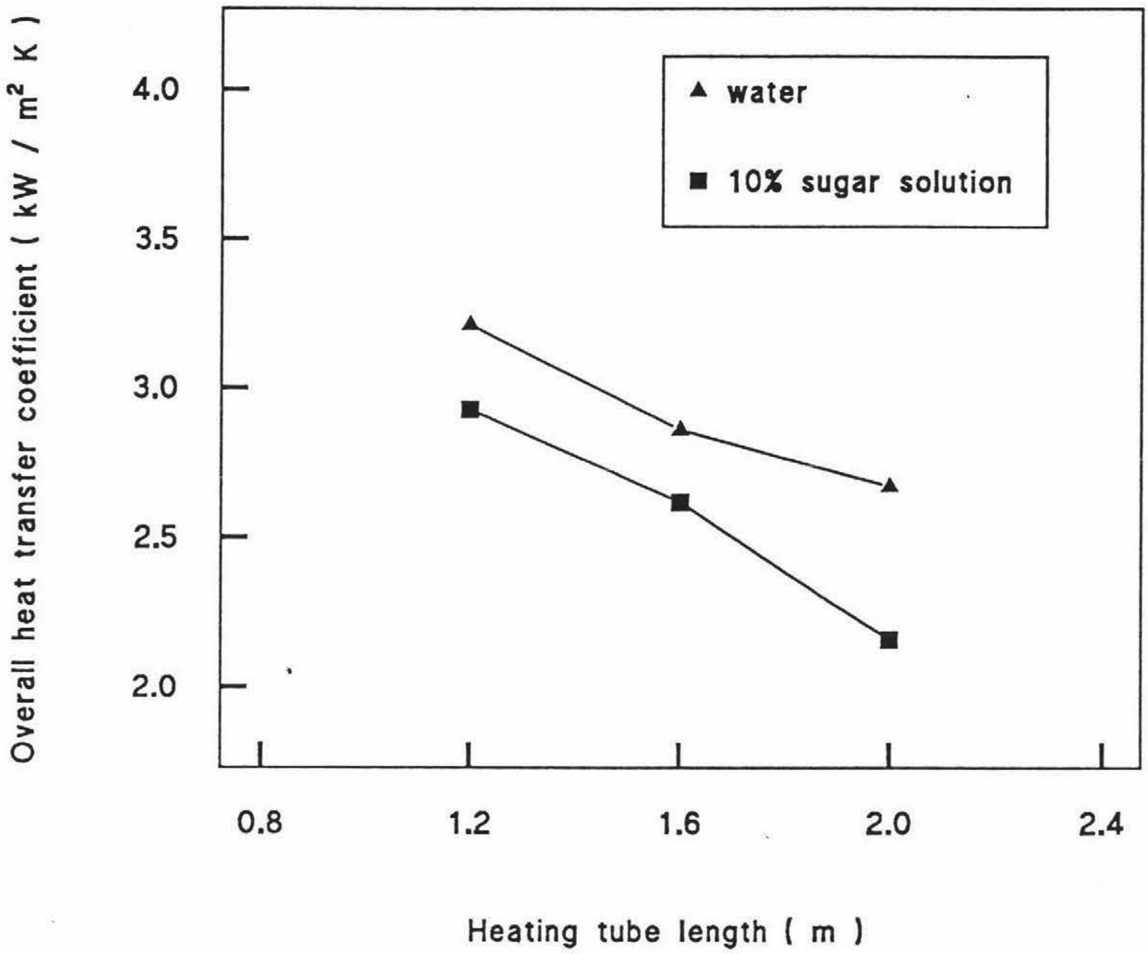


Fig. 12 Effect of heating tube length on overall heat transfer coefficient for water and 10 % sugar solution.

Temperature differences: 8 °C, Feed flow: 1000 ml/min

Evaporating temperature: 70 °C

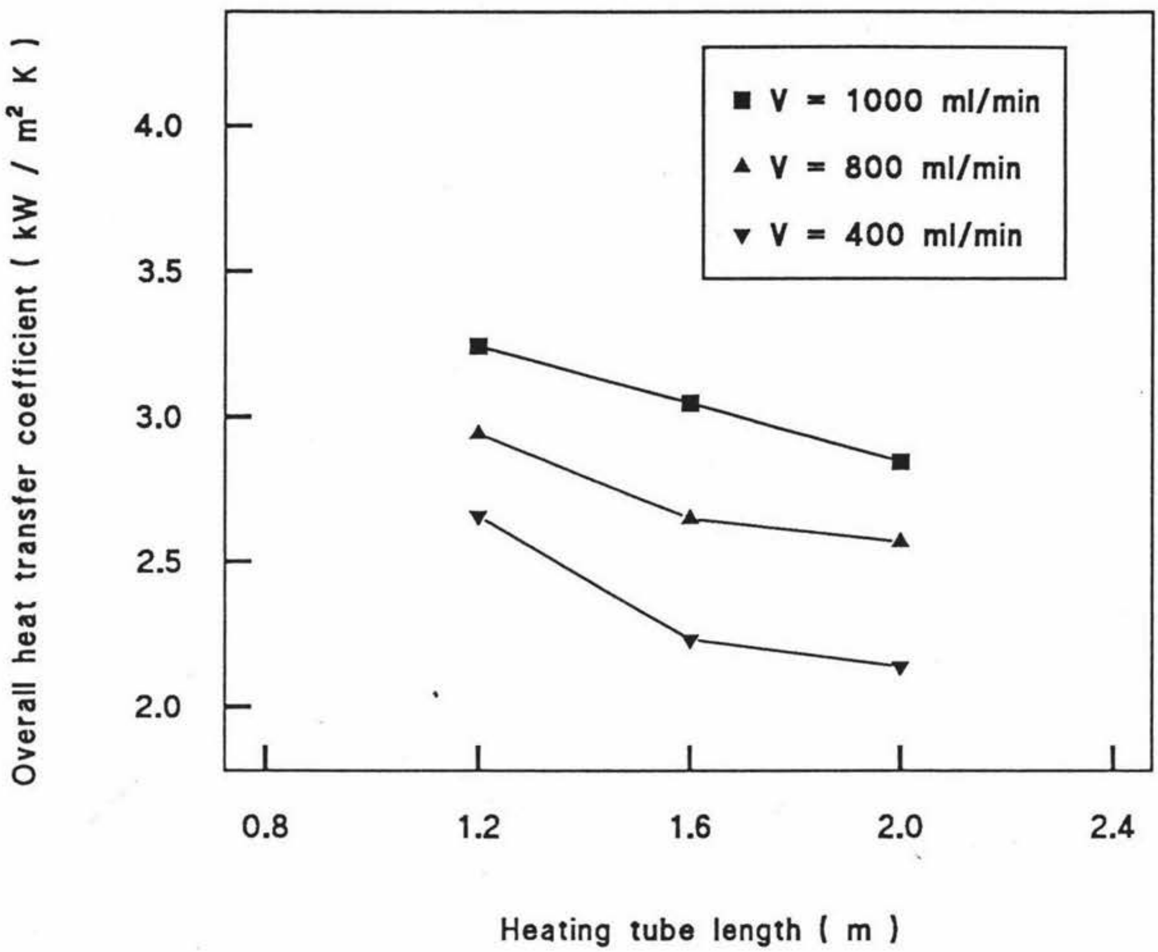


Fig. 13 Effect of heating tube length on overall heat transfer coefficient for 10 % sugar solution at different feed flows.

Temperature differences: 18 °C,

Evaporating temperature: 70 °C

inside of the heating tubes. Hence the effect found here is not likely to occur in commercial evaporators. However, it is indicate that heat transfer on the top part of heating tube could likely be improved by modifying the distribution device.

In the commercial trials, the vapour momentum had a positive effect on the heat transfer coefficient, but in this pilot scale evaporator the vapour momentum in very small because the tube length is not longer enough (max. 0.65 kg.m/s.m compered to 4 kg.m/s.m in commercial evaporators, Jebson, 1991). Therefore, there was no evidence which showed vapour momentum to contribute the heat transfer coefficient in this evaporator. It is still thought that if the heating tube length was longer, the effect of vapour momentum on the heat transfer coefficient would be significant.

The condensation of steam maybe also contribute the decreasing of over heat transfer coefficient as increasing the tube length. For the long tube, more condensate will be drip down on lower part of tube. Although the condensation is the dropwise type, the drops will coalesce with their neighbours and grow into rivulets, the more condensate will form more rivulets, which is in result of reducing the bare areas exposed to the steam. Therefore, for the long tube, the condensing side heat transfer coefficient will be low at the lower part of tube. But this film heat transfer coefficient is very large compared with the evaporating side film heat transfer coefficient.

5.7 The effect of Reynolds number

The tendency of increasing over heat transfer coefficient with increasing Reynolds number can be seen in Fig. 14. In this experiment the Reynolds number was changed by two variables; the irrigation density, and the viscosity.

A Minitab statistical package was used on the data in Fig. 5. The following equation was obtained:

$$U = 0.939 + 1.58 \times 10^{-3} Re - 3.5 \times 10^{-7} Re^2 + 6.45 \times 10^{-4} \mu \qquad (27)$$

$$(R^2 = 97.8\%)$$

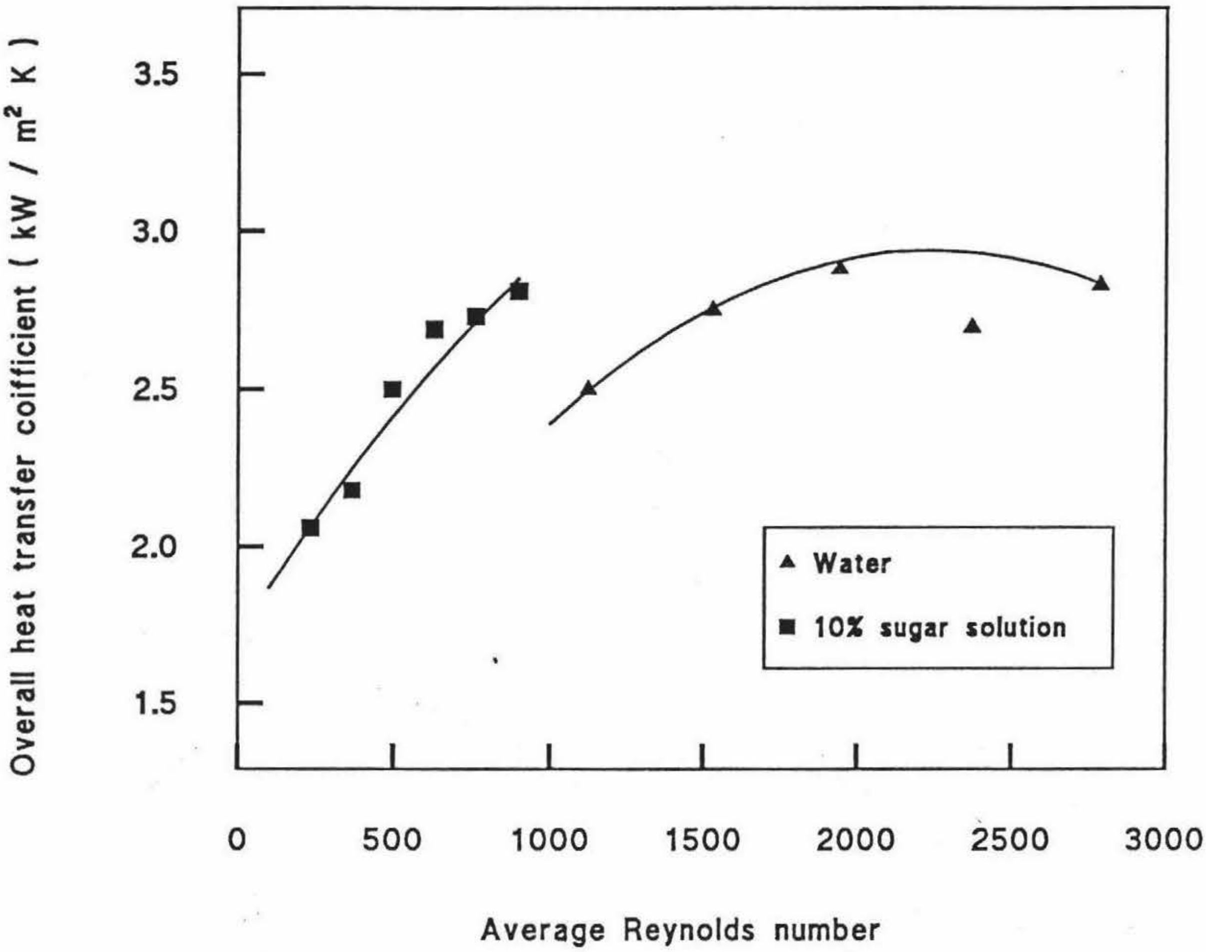


Fig. 14 Effect of average Reynolds number on overall heat transfer coefficient for water and 10 % sugar solution.

Temperature difference: 10 °C, Heating tube length: 2 m,

Evaporating temperature: 90 °C

It seems that the viscosity contributes to Reynolds number but also appears to alter the heat transfer coefficient in its own right. This may be due to the contribution of viscosity to Prandtl number, which has a positive effect on falling film heat transfer coefficient (Chun & Seban, 1971). In this experiment it is 2.32 for water and 8.35 for sugar solution. As the viscosity increases, the Prandtl number also increases but the Reynolds number decreases. Hence the net effect of viscosity on heat transfer coefficient in this experimental range is positive, and is in agreement with Chun & Seban.

At the higher Reynolds numbers two conflicting effects may be occurring. As the Reynolds number increases, the film thickness would also increase, decreasing the effect of liquid-vapour interface evaporation, this would lower heat transfer coefficient. However, the increasing Reynolds number would increase turbulence which would increase the heat transfer coefficient. It is possible that the scatter of experimental points at higher Reynolds number is a result of the combining of these effects and not experimental error. Hence the correlation would not adequately describe the heat transfer in the higher Reynolds number range.

In falling film evaporators irrigation density is very important, because dry spots will be formed on the heating tube wall if the irrigation density is too low. As the heat transfer coefficient increases with increasing irrigation density, high irrigation densities should be used, but for commercial evaporators high irrigation densities would require the use of large capacity pumps, and more energy consumed in the pumping system.

Schwartzberg (1988) summarised several researchers' work and reported that at Re < 500 over heat transfer coefficient increased linearity as Reynolds number increased, at 500 < Re < 1600, over heat transfer coefficient remained fairly constant or decreased slightly, at Re > 1600, over heat transfer coefficient increased slightly as Reynolds number increased. This is in agreement with the results reported here. For sugar solution the Reynolds number range is from 230 to 900. The over heat transfer coefficient increased as the Reynolds number increased. For water the Reynolds numbers from 1100 to 2700. The over heat transfer coefficient increased slightly in this range.

Chapter 6

Conclusions and Recommendations

6.1 Conclusions

It can be concluded from this experimental work that:

(1) A definite increase in the overall heat transfer coefficient resulted in decreasing the overall temperature difference was found. The overall temperature differences have a great influence on the heat transfer mechanism in the falling film, in tern on the heat transfer coefficient. When the overall temperature difference is less than 5-8 °C, the vapour-liquid interface evaporation is dominant, while the overall temperature difference is greater than 5-8 °C, the nucleating evaporation prevails. The higher heat transfer coefficient at lower overall temperature difference would be an additional advantage for the multiple effect evaporation.

(2) On the top part of the heating tube, the falling film flow regime on the tube wall will be affected by entry length effect. Then the heat transfer coefficient will also be affected. So the modification of liquid distribution to improve the heat transfer on the top of evaporator could probably be made.

(3) The dependence of the overall heat transfer coefficient on the evaporating temperature was observed in the range of 70 °C to 90 °C.

(4) Both the irrigation density and the liquid viscosity appeared to alter the heat transfer coefficient. At the $Re < 500$, the heat transfer increased linearly as the Reynolds number increased, at the $Re > 500$, the heat transfer coefficient increased slightly as the Reynolds number increased.

6.2 Recommendations

In future work on this subject, it is recommended that:

(1) The effect of flash on the heat transfer coefficient could be explored. From this aspect, the effect of vapour momentum on the heat transfer coefficient may also be found out.

(2) As the Reynolds numbers in most effects of commercial milk evaporators are of in the range of 1000-10000, it is worth to investigate the higher Reynolds number. Therefore the higher feed flow rate should be used in this evaporator by replacing the feed pump and concentrate pump.

(3) It is better to measure the heating tube wall temperature to work out the evaporating side film coefficients, which should explain the phenomena of the falling film evaporation more accurately and directly.

Appendices

Appendix I

The design calculation of heating column

Due to the height of ceiling at the pilot plant, the maxim heating tube length can only be 2 m. The tube diameter was selected to be 32 mm O. D., which is commonly used in the commercial evaporators. Therefore, the maxim heating area is determined. The design calculation was just to check the capacity of vapour condenser.

The maxim capacity of vapour condenser is about 0.03 kg/s according to the Alfa-Laval Centri-Therm instruction book.

It is assumed that the maxim temperature difference used in the single-tube falling film evaporator is 20 °C. The overall heat transfer coefficient is assumed to be 3 kW /m^2K. Therefore, The condensate of vapour can be calculated as follows:

$$W_{max} = U\ A\ \Delta T/\ h_v = 3 \times 0.2 \times 20\ /\ 2300 = 0.0052\ \mathrm{kg/s} < 0.03\ \mathrm{kg/s}$$

So the vapour condenser has sufficient capacity to fit the heating column.

Appendix II

The design calculation of the angled slots separator

(1) Based on the measurements of the commercial evaporator used at Tui Dairy Company, the vapour velocity, time, distance for particle and particle size in the separator were checked.

The dimensions of the separator used for the first effect is shown in Fig. 15.

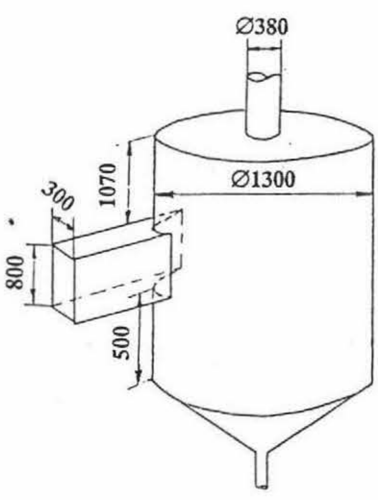


Fig. 15 The dimensions of the separator used in a commercial evaporator

There were two bodies in the first effect in which the vapour generated was 2.84 kg/s. The evaporating temperature was 72 °C. The specific volume of vapour at 72 °C is 4.6832 m^3/kg.

Therefore, vapour volume = $2.84 / 2 \times 4.6832 = 6.65$ m^3/s

Vapour velocity interring the separator = $6.65 / (0.3 \times 0.8) = 27.7$ m/s

Internal diameter = $1300 / 2 - 300 = 350$ mm = 0.35 m

It was assumed that the vapour had six circuits in the separator, thus

Internal time = $3.14 \times 0.35 \times 6 / 27.7 = 0.238$ s

Internal acceleration = $u_v^2 / r = 27.7^2 / 0.35 / 2 = 4385$ m/s^2

The distance of particles travelled = 0.3 m, time = 0.238 s

Thus, the particle velocity = 0.3 / 0.238 =1.26 m/s

According to Stoke's Law, i.e.

$$u_v = \frac{a\ D_p^2(\rho_p - \rho_v)}{18\mu} \tag{28}$$

$1.26 = 4385 \times D_p^2 \times (1024 - 1 / 4.6832) / (18 \times 10.94)$

$D^2 = 3.92 \times 10^{-5}$

The particle diameter D_p = 6.3 mm

(2) Design of the angled separator

Separator diameter was 75 mm, the length was about 900 mm.

The maxim vapour generated was 0.0178 m^3/s.

It was designed for this separator to use the same vapour velocity in the commercial separator, thus the vapour passed area is

$0.0178 / 27.7 = 6.43 \times 10^{-4}\ m^2$

If 6 mm wide slot was used, the total length = $6.43 \times 10^{-4} / (6 \times 10^{-3}) = 0.107$ m

There are a 10 mm margin on the edge and a 15 mm margin in the centre of the separator.

So the slot length = (75 - 25) / 2 = 25 mm

For the 6 mm × 25 mm slots, the number of slots = 0.107 / 0.025 = 4.28 ≈ 4

The angle of slots was selected to be 30°.

The distance of particle travelled per turn = 3.14 × 15 / cos 30° = 54.39 mm

The hight per turn = 3.14 × 15 × tan 30° = 27.2 mm

The number of turns = 600 / 27.2 = 22

Total distance of particles travelled = 54.39 × 22 /1000 = 1.20 m

The time = 1.20 / 27.7 = 0.043 s

The particle setting velocity = 0.025 / 0.043 = 0.582 m/s

Acceleration = $(27.7 \times \cos 30°)^2 / 0.025 = 23018\ m/s^2$

According to Stoke's Law, i.e.

$0.582 = 23018 \times D^2 \times 1024 / (18 \times 10.94)$

$D_p^2 = 4.86 \times 10^{-6}$

The particle diameter $D_p = 2.2$ mm < 6.3 mm

(3) The height of weir

The temperature drop of the vapour passed through the angled separator was assumed to be maxim 1 °C, the pressure drop correspond to 1 °C is about 0.0005 MPa, i.e.

$$0.0005 \text{ MPa} = 500 \text{ Pa} = 500 / 9810 = 0.05 \text{ m } H_2o = 50 \text{ mm } H_2o$$

Therefore, the height of weir above the separator was selected to be 60 mm.

The structure of the angled slots separator are shown in Fig. 6.

Appendix III

An example of the calculation of heat transfer coefficients

(1) The heating area

The outside diameter of heating tube is 0.032 mm and the heat transfer area is calculated as follows:

$$A = \pi \, D \, L$$

If L = 2 m, the heating area is

$$A = 3.41 \times 0.32 \times 2 = 0.2 \ m^2$$

(2) The heat flux

The amount of vapour condensate was measured and the heat flux is calculated as follows:

$$Q = (W / \nu) \, h_v$$

If W = 111 ml/min. = 1.85×10^{-3} l/s, at 70 °C, ν = 1.023 l/kg, h_v = 2334 kJ/kg, the heat flux is

$$Q = (1.85 \times 10^{-3} / 1.023) \times 2334 = 4.22 \ kW$$

(3) The heat transfer rate

$$q = Q / A = 4.22 / 0.2 = 21.1 \ kW/m^2$$

(4) The heat transfer coefficient

If ΔT = 8 °C, the heat transfer coefficient is

$$U = Q / A \, \Delta T = q / \Delta T = 21.1 / 8 = 2.63 \ kW/m^2.K$$

(5) The Reynolds number

On the top

$$\Gamma_t = V / \pi d, \quad Re_t = 4 \Gamma_t / \mu$$

At the bottom

$$\Gamma_b = (V - W) / \pi d, \quad Re_b = 4 \Gamma_b / \mu$$

The average Reynolds number is

$$Re = (Re_t + Re_b) / 2$$

If $V = 1000$ ml/min $= 1.667 \times 10^{-2}$ l/s, $W = 111$ ml/min $= 1.85 \times 10^{-3}$ l/s, for water at 70 °C, $\mu = 400$ µPa.s= 0.0004 Pa.s, $\rho = 0.978$ kg/l, thus

$$\Gamma_t = 1.667 \times 10^{-2} \times 0.978 /(3.14 \times 0.0288) = 0.18 \text{ kg/m.s}$$

$$Re_t = 4 \times 0.18 / 0.0004 = 1800$$

$$\Gamma_b = (1.667 \times 10^{-2} - 1.85 \times 10^{-3}) \times 0.978 /(3.14 \times 0.0288) = 0.16 \text{ kg/m.s}$$

$$Re_b = 4 \times 0.16 / 0.0004 = 1600$$

$$Re = (1800 + 1600) / 2 = 1700$$

(6) The momentum of vapour

The velocity of the vapour at the bottom

$$u_v = (W / v) v_v /(\pi d^2 / 4)$$

So the momentum of vapour is

$$M_v = u_v W /(\pi d)$$

If $W = 111$ ml/min $= 1.85 \times 10^{-3}$ l/s, for vapour and its condensate at 70 °C, $v_v = 5.048$ m^3/kg, $v = 1.023$ l/kg.

$$u_v = 1.85 \times 10^{-3} \times 5.048 \times 4 /(1.023 \times 3.14 \times 0.288^2) = 14.02 \text{ m/s}$$

$$M_v = 14.83 \times 1.85 \times 10^{-3} /(1.023 \times 3.14 \times 0.0288) = 0.28 \text{ kg.m/s.m}$$

(7) The heat released from vapour chamber surface

The vapour chamber's outside diameter is 75 mm and length is about 1 meter. So the exposed area can be calculated as follows:

$$A = \pi D L = 3.14 \times 0.075 \times 1 = 0.236 \text{ m}^2$$

The ambient temperature was assumed at 20 °C, and the highest chamber surface temperature was 90 °C. The film coefficient of air for vertical tube can be found from Perry's Chemical Handbook, i.e.

$$H_{af} = b\ (\Delta T_{af})^m\ L^{3m-1}$$

Under the conditions of this experiment, the values of b and m selected as follows:

$$b = 1.37,\ m = 0.25$$

Thus, the film coefficient is

$$H_{af} = 1.37\ (70)^{0.25}\ 1^{3 \times 0.25 - 1} = 3.96\ W/m^2.K$$

Therefore, the heat released can be calculated, i.e.

$$Q' = H_{af}\ A\ \Delta T = 3.96 \times 0.2355 \times 70 = 65.28\ W$$

The error of heat released to total heat flux is

$$Q' / Q = 65.28 / 4220 = 1.5\ \%$$

(8) The viscosity of sugar solution

The equation proposed by Campanella is

$$\mu = a \exp k\ (1+m_s)^n \exp \frac{b+c(1+m_s)}{T^m} \tag{29}$$

Where: $a = 8.1940 \times 10^{-2}$

$b = 2.5927 \times 10^{7}$

$c = 4.4433 \times 10^{6}$

$k = 9.1760 \times 10^{-2}$

$m = 2.8774$

$n = 4.9029$

If m_s = 10 %, T = 70 °C, the viscosity is

$$\mu = 0.504\ cP.$$

Appendix IV

The heat transfer coefficient for condensation film

According to the Nusselt equation from Perry's Chemical Engineering Handbook, the heat transfer coefficient for condensation film could be written as follows:

$$\frac{h_{cf}\,L}{k} = 0.925\ \left(\frac{L^3 \rho_s^2\ g}{\mu\ \Gamma}\right)^{\frac{1}{3}} \tag{30}$$

Under the conditions of this experiment, the parameters at 80°C are:

$L = 2$ m, $g = 9.81$ m/s^2, $k = 0.67$ W/m.K, $\mu = 3.51 \times 10^{-4}$ Pa.s,

$\Gamma_s = V_s / \pi D = 181 / (1000 \times 60 \times 3.14 \times 0.032) = 0.0299$ kg/m.s,

$\rho_s = 0.0293$ kg/m^3

Therefore, the heat transfer coefficient for condensation film is

$h_{cf} = 5.4$ kW/m^2.K.

Appendix V

Experimental results

1. For tap water

L (m)	T_E (°C)	ΔT (K)	V (ml/min)	W (ml/min)	q (kW/m^2)	U (kW/m^2.K)	Γ (kg/m.s)	Re	M_v (kg.m/s.m)
2	90	10	1400	147	27.3	2.73	0.223	3035	0.112
2	90	10	1000	150	27.9	2.79	0.151	2116	0.117
2	90	10	800	143	26.6	2.66	0.117	1667	0.106
2	90	10	600	130	24.2	2.42	0.084	1224	0.088
2	90	8	1000	122	22.7	2.83	0.156	2148	0.077
2	75	8	1000	144	21.4	2.67	0.159	1812	0.121
2	85	8	1000	120	22.4	2.80	0.157	2033	0.090

2. For 10% sugar solution

L (m)	T_E (°C)	ΔT (K)	V (ml/min)	W (ml/min)	q (kW/m^2)	U ($kW/m^2.K$)	Γ (kg/m.s)	Re	M_v (kg.m/s.m)
2	70	18	1000	181	34.8	1.94	0.184	614.6	0.373
2	70	13	1000	134	25.8	1.98	0.189	630.6	0.205
2	70	5	1000	58	11.2	2.23	0.197	656.2	0.039
2	70	3	1000	40	7.70	2.57	0.199	662.3	0.018
2	80	18	1000	210	40.4	2.25	0.181	604.9	0.337
2	80	13	1000	157	30.2	2.32	0.187	622.8	0.189
2	80	5	1000	74	14.2	2.85	0.195	650.8	0.042
2	80	3	1000	50	9.62	3.21	0.198	658.9	0.019
2	80	10	1000	125	24.1	2.41	0.190	633.6	0.120
2	90	10	1400	146	28.1	2.81	0.269	896.8	0.111
2	75	8	1400	97	18.7	2.33	0.274	913.4	0.087
2	85	8	1400	112	21.6	2.69	0.272	908.3	0.079

Continue Appendix V

L (m)	T_E (°C)	ΔT (K)	V (ml/min)	W (ml/min)	q (kW/m^2)	U ($kW/m^2.K$)	Γ (kg/m.s)	Re	M_v (kg.ms./m)
1.2	70	18	1000	182	58.4	3.24	0.184	614.3	0.377
1.2	70	18	400	149	47.8	2.66	0.066	220.0	0.253
1.2	70	18	800	165	52.9	2.94	0.145	484.9	0.310
1.6	70	18	1000	228	54.9	3.05	0.180	598.8	0.509
1.6	70	18	400	167	40.2	2.23	0.064	213.9	0.318
1.6	70	18	800	198	47.7	2.65	0.142	473.8	0.446
2	70	18	400	200	38.5	2.14	0.061	202.8	0.455
2	70	18	800	240	46.2	2.57	o.138	459.6	0.656
2	90	13	1000	172	33.1	2.55	0.185	617.7	0.129
2	90	10	400	107	20.6	2.06	0.070	234.2	0.060
2	90	10	600	113	21.8	2.18	0.110	367.3	0.067
2	90	10	800	130	25.0	2.50	0.149	496.7	0.088

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Nomenclature

Symbols	Definition and units
A	Heat transfer area (m^2)
a	Acceleration of particles or constant
b	Constant
C_p	Specific heat of liquid (J/kg.K)
c	Constant
D	Outside diameter of tube (m)
D_p	Diameter of particles (m)
d	Inside diameter of tube (m)
F	Degree of freedom
Fr	Froude number (expressed as μ^2/gd)
f_k	A constant of physical properties of the condensate film
g	Gravitational acceleration (m/s^2)
H_{af}	Film coefficient for condensation films ($W/m^2.K$)
H_{cf}	Film coefficient for air films ($W/m^2.K$)
h_V	Condensation enthalpy of the vapour at the pressure concerned (J/kg)
K	Consistency index
k	Constant
L	Length of tube (m)
M_v	Vapour momentum, it is expressed as mass times velocity per unit of perimetric length (kg.m/s.per.m)
m	Constant
m_s	Total solids percentage (%)
m_w	Water percentage (%)
Nu'	Modified Nusselt number (expressed as $\alpha_1/\lambda\ [u^2/\rho^2 g]^{1/3}$)
n	The number of effect or constant
P	Pressure (kPa)
Pr	Prandtl number (expressed as $c\mu/\lambda$)
Q	Heat flux (W)

Q'	Heat released from vapour chamber surface (W)
q	Heat transfer rate (W/m^2)
Re	Reynolds number (expressed as $4\Gamma/\mu$)
Re_b	Reynolds number at the bottom of heating tube
Re_{CR}	Critical Reynolds number (transition point from laminar to turbulent)
Re_t	Reynolds number at the top of heating tube
R_L	Inside heat transfer resistance of heating tube ($m^2.K/W$)
R_V	Outside heat transfer resistance of heating tube ($m^2.K/W$)
r	Radius of separator (m)
T	Temperature (K)
T_E	Evaporating temperature (°C)
T_S	Steam temperature (°C)
ΔT_{af}	Temperature difference cross the air film (K)
ΔT	Overall temperature difference (K)
ΔT_{BPE}	Liquid boiling point elevation (°C)
ΔT_V	Temperature difference cross the condensate film (K)
U	Overall heat transfer coefficient ($W/m^2.K$)
u	Velocity of film flow (m/s)
u_v	Velocity of vapour (m/s)
V	Inlet liquid flow rate (ml/min)
V_s	Steam condensate flow rate (ml/min)
W	Vapour condensate flow rate (l/s)
We	Weber number (expressed as $[\rho\ u^2\delta/\sigma]^{1/2}$)

Greek symbols

α_L	Surface heat transfer coefficient on the side of the liquid (falling film) ($W/m^2.K$)
α_V	Surface heat transfer coefficient on the side of the vapour condensation ($W/m^2.K$)
Γ	Average irrigation density, it is defined as mass flow rate per unit of perimetric length (kg/m.s)

Γ_b	Irrigation density at the bottom of heating tube (kg/m.s)
Γ_s	Irrigation density of steam condensate (kg/m.s)
Γ_t	Irrigation density at the top of heating tube (kg/m.s)
γ	Flow behaviour index
δ	Film thickness (m)
δ_w	Thickness of tube wall (m)
λ	Thermal conductivity of liquid (W/m.K)
λ_w	Thermal conductivity of tube wall (W/m.K)
μ	Liquid viscosity (Pa.s)
ν	Specific volume of liquid (l/kg)
ν_v	Specific volume of vapour (m^3/kg)
ξ	The mole fraction of solution
ρ	Liquid density (kg/m^3)
ρ_p	Particle density (kg/m^3)
ρ_s	Steam density (kg/m^3)
ρ_v	Vapour density (kg/m^3)
σ	Surface tension of the liquid (N/m^2)