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# Modelling of Heat Transfer in a Climbing-Film Evaporator: III. Application to an Industrial Evaporator*

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*ABSTRACT*

*A mathematical model developed to predict the heat transfer in a single-tube climbing-film evaporator is used to evaluate the performance of a three-stage shell-and-tube evaporator for concentrating 5000 kg $h^{-1}$ of pineapple juice from 13·5 to 62° Brix.*

## NOTATION

$A$ Cross-section of flow ($m^2$).
$g$ Acceleration of gravity ($m\ s^{-2}$).
$h$ Heat-transfer coefficient ($W\ m^{-2}\ K^{-1}$).
$H$ Distance (m).
$k$ Thermal conductivity ($W\ m^{-1}\ K^{-1}$).
$P$ Pressure (Pa).
$U$ Overall heat-transfer coefficient ($W\ m^{-2}\ K^{-1}$).
$V$ Velocity ($ms^{-1}$).
$W$ Mass flow rate ($kg\ s^{-1}$).
$\Delta x$ Wall thickness (m).
$z$ Tube length from bottom of evaporator (m).
$\alpha$ Void (volume) fraction of vapour.
$\rho$ Density ($kg\ m^{-3}$).

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**Subscripts**

c Contraction.
e Expansion.
G Glass single-tube evaporator.
i In space between sections.
M Metal (stainless steel) evaporator.
t In the tube.
z At position z (local).
1 Liquid phase.
2 Vapour phase.

## INTRODUCTION AND LITERATURE REVIEW

In food processing, evaporation is a unit operation where the concentration of the solution is increased by driving off a portion of the solvent in the form of vapour. In most cases the solvent vaporised is water and the concentrated solution is the desired product.

The problem is more complex than in the evaporation of chemical products as the end product is not only subject to physical and biological criteria but to organoleptic assessment. Such hard-to-define properties as flavour, aroma, colour and texture must be of prime concern in the process design as well as in the equipment design and operation.

The use of a climbing-film evaporator is often the best solution, as high rates of heat transfer can be maintained without the use of high temperature or long contact time. Quite often, because it is necessary to produce a highly concentrated finished product from a very weak liquor, a single-pass climbing-film evaporator will not be sufficient. One solution is to recirculate the liquor through the evaporator. Another solution is to pass the liquor in series through two or more evaporator units or calandrias. These calandrias may be formed by incorporating into a single steam shell two or more sections of tubes, each section having fewer tubes than the preceding one and being provided with a separator at the top. The flow of liquor passing through each section and each separator in series progressively diminishes, but, since the number of tubes is also reduced, an adequate flow is provided in each section to maintain a climbing-film condition. Such evaporators have been used for handling heat-sensitive liquors such as coffee, fruit juices, milk, tea, whey and wine (Slade, 1967).

The sloping calandria evaporator described by Brennan *et al.* (1976) can be considered similar to a recirculating climbing-film evaporator tilted in an inclined position. It was widely used in the dairy industry up to the mid-1950s when it was progressively replaced by more sophisticated evaporators (Scott, 1964).

Liquid food-stuffs very sensitive to heat can be evaporated at very low temperature in a low-temperature evaporator (Slade, 1967). These evaporators are of the recirculating climbing-film type and are operated at high vacuum. The boiling point of the liquor being of the order of 15°C, the temperature of the cooling water usually available is too high to condense the vapour formed. Consequently an auxiliary fluid (ammonia) is used. The low-temperature evaporator was widely used in the concentration of orange juice following the Second World War but was replaced by single-pass evaporators, where the disadvantage of the high temperatures was compensated for by the decrease in residence time leading to similar or even better organoleptic qualities of the concentrated juice.

A combination of the climbing and falling film may be used where a high percentage of evaporation is required and when the concentrated liquor tends to be viscous; as the most concentrated liquor is formed in the falling film section, the flow is improved by the force of gravity. It is generally used as a finisher evaporator following a climbing-film evaporator where the greatest part of the evaporation has been effected. It has been used to produce orange and tomato juice.

In all of the above types of equipment it is essential that the evaporation of water from the product by heat should be gentle enough to preserve its nutritive value and prevent browning reactions, either by caramelisation of sugars or by reaction between proteins and sugars. Undue denaturation of proteins and destruction of vitamins by excess or prolonged heating must also be avoided. This means that evaporators should work at conditions which minimise the above effects. Further, it is important for cost-related reasons to minimise energy consumption and hence to determine the conditions that best achieve both quality and economic objectives. This may be best accomplished by developing a mathematical model of the evaporator which can then be turned into a computer program and used to optimise objectives such as those described above. This paper describes the application of the mathematical model developed for a single tube evaporator (see Parts I and II (Bourgois and Le Maguer, 1983*a*, *b*)) to an industrial-size climbing-film evaporator.

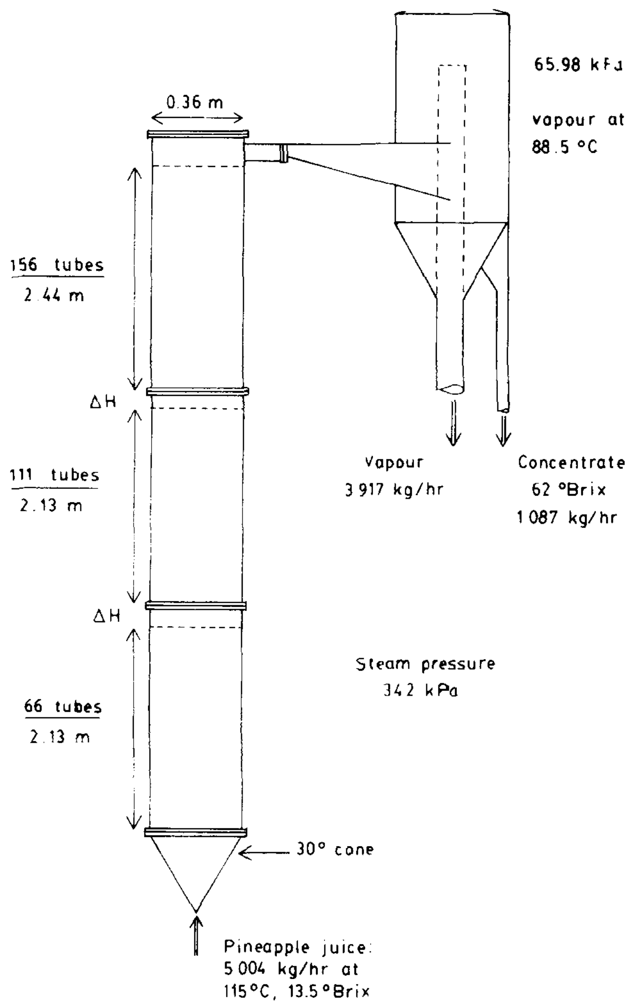


**Fig. 1.** Schematic diagram of the industrial evaporator.

## MATERIALS AND METHOD

### *Description of the evaporator*

A schematic diagram of the evaporator studied is given in Fig. 1. This evaporator is used to concentrate pineapple juice from 13·5 to 62° Brix at a feed rate of approximately 5000 kg $h^{-1}$. As can be seen from the diagram, the evaporator is composed of three consecutive sections, each section having more tubes than the preceding one to allow for the increasing volume of vapour and to ensure a climbing-film regime during the entire process. The available data are the inlet conditions, the outlet conditions and the details of the evaporator construction (tube diameter, tube length, wall thickness and number of tubes in each section). The characteristics are given in Table 1.

### *Evaluation of the heat-transfer coefficient*

For the single-tube glass evaporator, an overall heat-transfer coefficient between the steam in the jacket and the boiling film inside the tube was evaluated ($U_{Gz}$). The resistance to heat transfer is composed of three resistances in series: resistance of the film of condensed steam on the steam side, resistance due to the wall of the tube itself, and resistance of the film of liquid climbing the inside of the evaporator tube.

**TABLE 1**
Characteristics of the Evaporator(s)

*Tubes*
Stainless steel, 19 mm inside diameter.
Thermal conductivity: $k_M = 15$ W $m^{-1}$ $K^{-1}$.
$\Delta x_M = 1{\cdot}27$ mm.

*Tube length*
Section 1: 2·13 m. Section 2: 2·13 m. Section 3: 2·44 m.

*Intersection space*
Length: 0·153 m. Diameter: 0·36 m.

Thermal conductivity of glass: $k_G = 1{\cdot}163$ W $m^{-1}$ $K^{-1}$.
$\Delta x_G = 1{\cdot}53$ mm.

The values obtained for the single-tube glass evaporator can be used to estimate the heat-transfer coefficient of the industrial evaporator. Since the diameter of the tubes is much greater than their wall thickness, the local heat-transfer coefficient is estimated using the equation:

$$1/U_{Mz} = 1/U_{Gz} + (\Delta x/k)_M - (\Delta x/k)_G \tag{1}$$

where subscripts M and G refer to metal and glass, respectively. This assumes that the heat-transfer coefficients on the steam side are similar in both cases and that the heat transfer through the climbing film on the inside is a function of the void fraction $\alpha$ only.

Using the thermal conductivity values provided in Table 1, $U_{Mz}$ is given by the following equations:

$$U_{Mz} = 2539 \text{ W m}^{-2} \text{ K}^{-1} \qquad \text{for } \alpha < 0{\cdot}619 \tag{2}$$

$$U_{Mz} = 4657{\cdot}1 - 3423{\cdot}7\alpha \qquad \text{for } 0{\cdot}619 < \alpha < 1 \tag{3}$$

*Estimation of inlet pressure*

From results obtained with the glass climbing-film evaporator in Part II one can estimate the pressure at the inlet of the evaporator using a linear drop of 2·4 kPa m$^{-1}$ of tube. So a first estimate of the expected pressure drop will be close to 15·93 kPa (neglecting entry and exit losses at the intermediate plates), which added to the outlet pressure (65·98 kPa) gives an approximate inlet pressure of 81·9 kPa. This corresponds to a boiling temperature of 94°C. Therefore, with an inlet temperature of 115°C, flashing may be expected to occur in the conical section preceding the first set of tubes, and a mixture of vapour and liquid would enter the first section of the evaporator ensuring two-phase flow throughout the tubes in each section. The exact pressure will have to be determined starting from the above estimate and taking into account, by using the model, the actual pressure drop occurring through the equipment.

## Method of calculation

*Evaluation of the amount of flashing*

From the original estimate of the pressure at the inlet and the composition and temperature of the entering feed the ratio of vapour to liquid can be estimated through a direct flash calculation. Subsequently an iterative evaluation converging on the outlet vapour temperature is used.

*Heat and mass transfer in each section*

It will be assumed that the flow is equally distributed between the tubes in each section, so that the flow in each tube is the total flow divided by the number of tubes in that section. The original model is then applied to each tube taking into account the characteristics of the tube: length, diameter and heat-transfer coefficient. The pressure at the top of that section is then obtained, the vapour and concentrate rates are determined and by multiplying the results for one tube by the number of tubes in that section the total rates may be calculated.

*Evaluation of the pressure drop between two sections*

The pressure drop is composed of three terms:

(1) The friction losses due to the expansion when going from the inside of the top of the lower tube to the space between sections. These are given by:

$$\Delta P_e = (1 - A_t/A_i)^2 (\rho_1 V_1^2/2 + \rho_2 V_2^2/2) \tag{4}$$

where $V_1$ and $V_2$ refer to velocities of liquid and vapour, respectively, at the top of the tube.

(2) The hydrostatic head of the mixture in the space between adjacent plates which is given by:

$$\Delta P_h = \Delta H [\alpha \rho_2 + (1 - \alpha) \rho_1] g \tag{5}$$

(3) The friction losses due to the contraction when going from the space between tube sections to the inside of the bottom of the upper tube. These are given by:

$$\Delta P_c = 0{\cdot}5 (\rho_1 V_1^2/2 + \rho_2 V_2^2/2) \tag{6}$$

where $V_1$ and $V_2$ refer to velocities of liquid and vapour, respectively, at the bottom of the tube.

The pressure drop between adjacent sections is the sum of these three terms.

$$\Delta P = \Delta P_e + \Delta P_h + \Delta P_c \tag{7}$$

Figure 2 illustrates the procedure followed by the program to converge on the desired exit temperature.

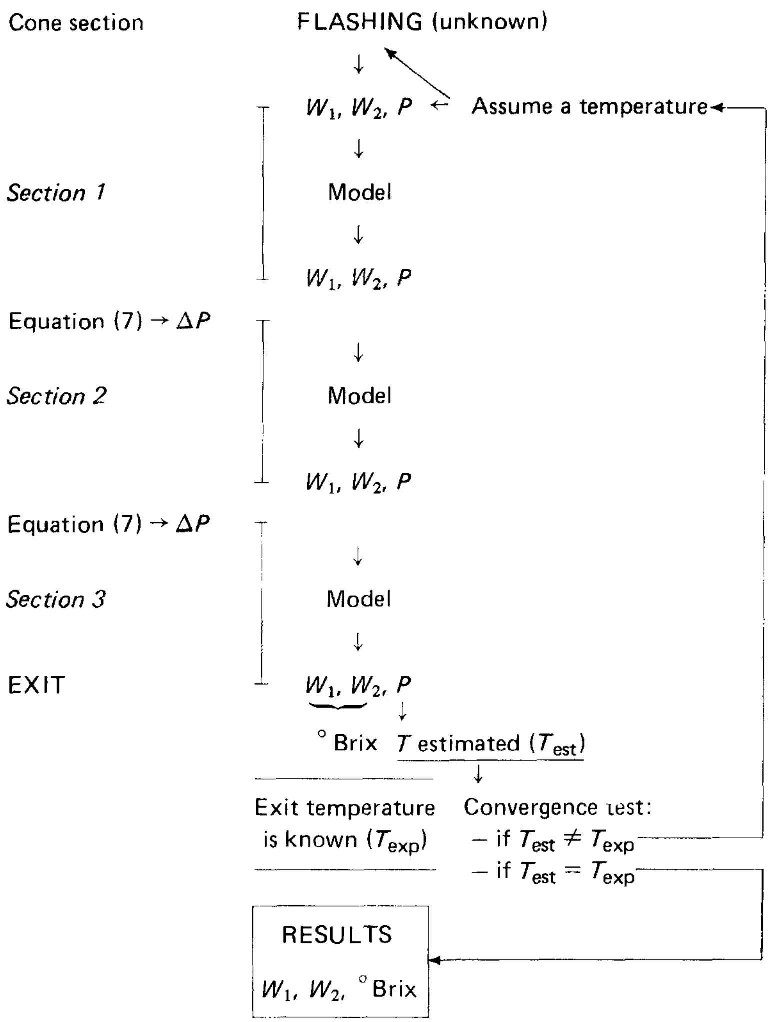


**Fig. 2.** Diagram of the program to converge on the exit temperature. The initial conditions at the inlet of the evaporator are known: temperature, feed rate, °Brix.

## RESULTS

The values of the concentrate rate, vapour rate, temperature, °Brix and volume fraction of vapour at the exit of each section are given in Table 2. They differ from the measured values by 2·7 and 0·8% for the pro-

**TABLE 2**
Values Predicted by the Model[a]

*Section 1*

Average heat transfer = 1604 W $m^{-2}$ $K^{-1}$.
Liquid velocity $V_1$ = 1·27 m $s^{-1}$; vapour velocity $V_2$ = 37·6 m $s^{-1}$.

| *Concentration rate (kg $s^{-1}$)* | *Vapour rate (kg $s^{-1}$)* | *Vapour temperature (°C)* | $\alpha$ | *°Brix* |
|---|---|---|---|---|
| 1·092 | 0·298 | 91·82 | 0·957 | 17·18 |

*Section 2*

Average heat transfer = 1342 W $m^{-2}$ $K^{-1}$.
Liquid velocity $V_1$ = 1·69 m $s^{-1}$; vapour velocity $V_2$ = 48·4 m $s^{-1}$.

| *Concentration rate (kg $s^{-1}$)* | *Vapour rate (kg $s^{-1}$)* | *Vapour temperature (°C)* | $\alpha$ | *°Brix* |
|---|---|---|---|---|
| 0·746 | 0·644 | 90·96 | 0·987 | 25·15 |

*Section 3*

Average heat transfer = 1022 W $m^{-2}$ $K^{-1}$.
Liquid velocity $V_1$ = 1·97 m $s^{-1}$; vapour velocity $V_2$ = 59·0 m $s^{-1}$.

| *Concentration rate (kg $s^{-1}$)* | *Vapour rate (kg $s^{-1}$)* | *Vapour temperature (°C)* | $\alpha$ | *°Brix* |
|---|---|---|---|---|
| 0·311 | 1·079 | 88·50 | 0·997 | 60·34 |

[a] Initial flashing = 4·0%; initial void fraction $\alpha = 0{\cdot}761$.

duct concentration and vapour flow rate, respectively. This demonstrates the adequacy of the model developed for the single-tube glass evaporator.

The calculated amount of flashing-off occurring in the cone is 4·0% of the mass, ensuring a two-phase flow from the start. Table 2 shows that the initial void fraction $\alpha$ is greater than 0·619 and the heat-transfer coefficient is consequently evaluated using eqn (3). It decreases in each consecutive section when the void fraction increases. This trend was observed experimentally with the glass evaporator and is presumably due to the increasing resistance to heat transfer on the product side. At high values of $\alpha$, the vapour velocity is very high and the flow is

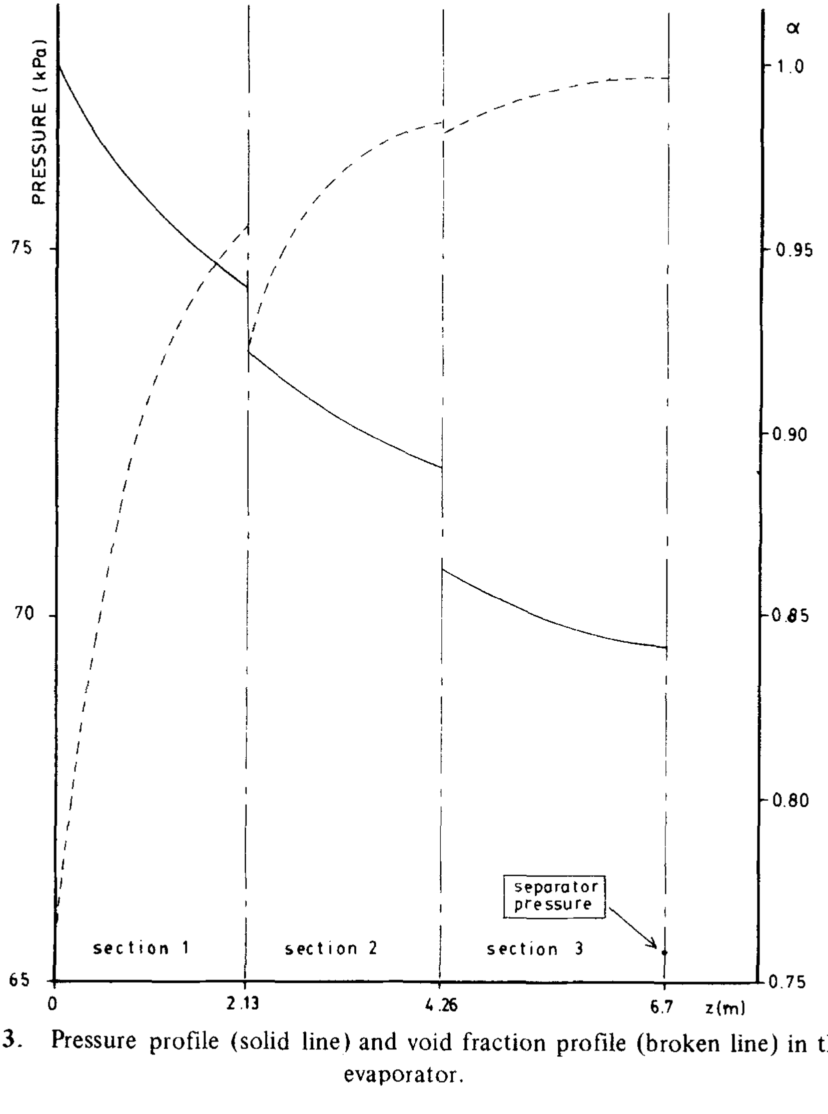


**Fig. 3.** Pressure profile (solid line) and void fraction profile (broken line) in the evaporator.

no longer truly annular. The central vapour core contains many entrained droplets of liquid which decrease the heat transfer in the liquid layer. In the event of a total disappearance of the film, the heat transfer may drop by a factor of ten.

The velocities at the top of the evaporator (in the third section) are 1·97 m $s^{-1}$ for the liquid phase and 59·0 m $s^{-1}$ for the vapour core. From the bottom part to the top section the slip ratio varies by no more than 5% which is consistent with theory.

A profile of the pressure along the evaporator is shown in Fig. 3 together with the profile of the void fraction $\alpha$. The void fraction increases sharply to values over 0·9 which, in turn, decrease the heat-transfer coefficient as discussed above. The average value of the heat-transfer coefficient in section three is only two-thirds of the value for the first section (see Table 2). In practice it would therefore be advantageous to use multiple-stage evaporation with separation of the vapour after each stage. This would decrease the volume fraction in the next stage giving all the benefits of a higher heat-transfer coefficient.

With this in mind, the model was used to predict that the same concentration could be achieved with a two-stage system, the first stage composed of 156 tubes 2·44 m long with a separator at the top and the second stage of 89 tubes 2·44 m long. The average heat-transfer coefficient would be 1541 W $m^{-2}$ $K^{-1}$ for the first stage and 1561 W $m^{-2}$ $K^{-1}$ for the second stage as compared to 1604 and 1342.

From a design standpoint, the addition of an extra separator after the first stage is offset by the elimination of one stage and the reduction in height of the system.

## CONCLUSION

The model successfully represented this particular evaporator and provided insight into the intermediate stages of the total concentration process and the limiting variables. It provides a useful tool with which to optimise and modify evaporators either existing or in the design stages, as demonstrated above.

Subsequently such a model could be used in process control situations to optimise process conditions according to economic and energy criteria.

## REFERENCES

Bourgois, J. and Le Maguer, M. (1983*a*). Modelling of heat transfer in a climbing-film evaporator: Part I. *J. Food Engineering*, **2** (1), 63.

Bourgois, J. and Le Maguer, M. (1983*b*). Modelling of heat transfer in a climbing-film evaporator: Part II. *J. Food Engineering*, **2** (3), 225.

Brennan, J. G., Butters, J. R., Cowell, N. D. and Lilly, A. E. V. (1976). *Food Engineering Operations.* 2nd Edn, Applied Science Publishers Ltd, London.

Scott, R. (1964). Evaporators and evaporation. *Dairy Industries*, **29**, 749.

Slade, F. H. (1967). *Food Processing Plant*. Vol. 1, CRC Press, Cleveland.