

# The relationship between flow pattern and thermal non-equilibrium in the multi-stage flash evaporation process

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(Received September 8, 1992)

## SUMMARY

The approach of a flashing horizontal stream to thermal equilibrium in a geometry typical to a flash chamber (stage) of a multi-stage flash (MSF) evaporator was modeled by combining the results of our past numerical analysis of turbulent isothermal flow patterns and pressure distributions in such a flash chamber with an empirical correlation developed here for determining the rate of temperature change of a flashing liquid emanating from an orifice and long tubes. The model was shown by experiments to work well as long as the flow was not significantly disturbed by separation or ebullition, which occurred here for the test conditions of the highest flow rate with the highest baffle plate placed closest to the orifice [ $W = (8.7)10^5$  kg/h · m,  $l_b = 0.15$  m,  $h_b = 0.20$  m] or with flash-down temperature differences ( $T_i - T_o$ ) exceeding 1.5 K. The delivery of the entering hot liquid to the flashing zone near the free surface, accomplished here by installing a baffle, was found to be a very effective way to enhance flash evaporation rates.

## SYMBOLS

|        |   |
|--------|---|
| $h$    | — height of liquid, m   |
| $h_b$  | — height of baffle plate, m   |
| $h_o$  | — height of orifice opening, m  |
| $k$    | — flash evaporation coefficient in Eqn. (3), $[(s \cdot K)^{-1}]$   |
| $l$    | — length of flash chamber, m  |
| $l_b$  | — distance of baffle chamber from orifice, m  |
| $NETD$ | — non-equilibrium temperature difference, $T_o - T_v$ , K   |
| $t$    | — time, s   |
| $T$    | — temperature of liquid, °C   |
| $T_i$  | — mean inlet temperature of liquid, °C  |
| $T_o$  | — mean outlet temperature of liquid, °C   |
| $T_s$  | — saturation temperature (for pure liquids) or boiling point (for solutions) corresponding to local liquid pressure, °C |
| $T_v$  | — mean temperature of emanating vapor, °C   |
| $u$    | — x-component of velocity, m/s  |
| $v$    | — y-component of velocity, m/s  |
| $W$    | — mass flow rate of liquid per unit width of flash chamber<br>[kg/h · m-width]  |
| $x$    | — horizontal coordinate, m  |
| $y$    | — vertical coordinate, m  |

*Greek*

|              |   |
|--------------|---|
| $\Delta T$   | — stage flash-down, $T_i - T_o$ , K   |
| $\Delta T_s$ | — initial apparent superheat, $T_i - T_v$ , K   |
| $\Psi$       | — dimensionless stream function = (stream function)/<br>(volumetric flow rate per unit stage width) |

*Abbreviations*

|        |  |
|--------|--|
| empty  | — flash chamber without a baffle plate |
| baffle | — flash chamber with a baffle plate    |

## INTRODUCTION

There is much interest and little fundamental information so far on flash evaporation from horizontal streams (cf. [1,2]). One of the applications which drives this interest is water desalination, in which the process of MSF

evaporation is used to desalt most of the water being desalinated to date (cf. [3]). The liquid being distilled (say, brine or seawater) in this process flows as a horizontal free-surface stream through a number of successive flash chambers (stages) with the vapor pressure decreasing from stage to stage in the direction of the liquid flow. As shown in Fig. 1, it flows through an inlet aperture into a region where the pressure is lower than its saturation pressure, and flashing is therefore initiated. The emanating vapor is condensed on a tube bundle positioned in the vapor space above the liquid stream. The distillate is collected in trays and removed. As the liquid evaporates, it proportionately also cools down, and evaporation thus proceeds in the flash chamber until thermodynamic equilibrium between the liquid and the vapor is attained. It is of great interest to determine the rate of approach to equilibrium in this process, both for its fundamental understanding and for optimizing stage design.

The evaporation rate, due to both free-surface evaporation and ebullition, is integrally linked with the fluid mechanics and heat and mass transfer phenomena occurring in the flash chamber. The fluid mechanics are especially complex due to the effects of the inlet orifice and of the downstream geometry, in some cases leading to a hydraulic jump and in others complicated by devices placed in the flash chamber to enhance equilibration rates. In the authors' previous study [4] a numerical analysis was performed to determine turbulent isothermal flow patterns and pressure distributions in

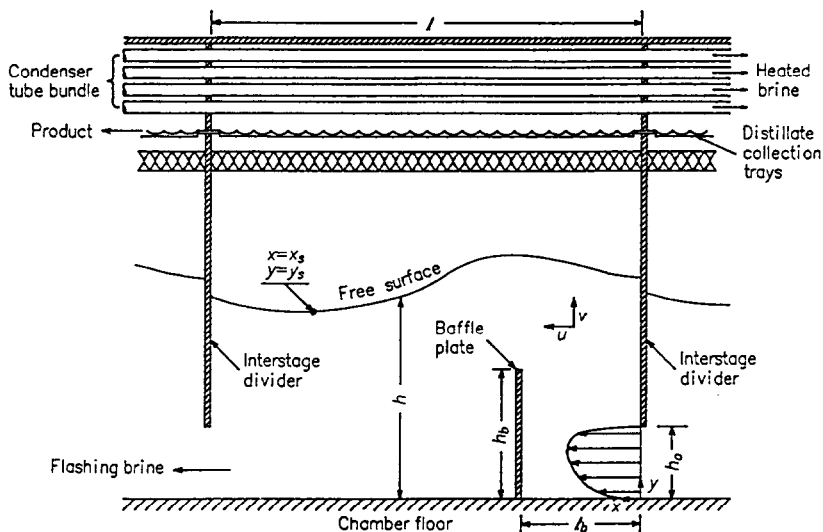


Fig. 1. Flash chamber (stage) model description.

a flash chamber of this geometry, with and without a downstream vertical baffle plate (sill). The analysis was validated by experiments. Based on this solution, a numerical analysis was performed here to predict the non-equilibrium temperature difference at the outlet of the flash chamber, and experiments were performed to validate the analysis.

The non-equilibrium temperature difference (*NETD*) and the initial apparent superheat ( $\Delta T_s$ ) are defined, respectively, as

$$NETD = T_o - T_v \quad (1)$$

and

$$\Delta T_s = T_i - T_v \quad (2)$$

where  $T_i$  and  $T_o$  are the mean temperature of the liquid at the flash chamber inlet and outlet, respectively, and  $T_v$  is the mean temperature of the emanating vapor which is essentially equal to the saturation temperature (for pure liquids) or to the boiling point (for solutions) corresponding to the pressure in the vapor space above the liquid stream. The *NETD* is a good indicator of the approach to equilibrium in a flash chamber and serves as an important parameter in this study. as  $NETD \rightarrow 0$ , the liquid approaches equilibrium where it cannot release any more vapor. In industrial practice it is naturally desirable to minimize the *NETD* with the most economical stage design, often coinciding with a flash chamber (stage) of least length.

## THE NUMERICAL ANALYSIS

### *The mathematical model*

As observed in both experiments and plant operation (cf. [1,2,5]), the flash evaporation process in a flash chamber may, depending on the local liquid temperature ( $T$ ) and initial apparent superheat ( $\Delta T_s$ ) of the liquid, exhibit behavior which ranges from vigorous, almost explosive ebullition and wavy flow in the stages with high temperatures and superheats, to more tranquil flow with evaporation occurring primarily near the free surface for the low ones. The former process was not modeled analytically yet because of its complexity. The latter, more tranquil process will be modeled here by using the velocity and pressure distribution results obtained by numerical analysis in the previous study [4] of two-dimensional turbulent isothermal flow and by using for each streamline an empirical correlation, also devel-

oped by the authors, which relates the rate of flashing liquid temperature decrease to the local superheat ( $T - T_s$ ). It is assumed in the model here that the flash chamber of length  $l$  has a baffle of height  $h_b$  placed at distance  $l_b$  from the inlet orifice, as shown in Fig. 1.

As an introduction to the description of the model, Fig. 2 shows three of the stream line and isobar maps obtained in our previous study [4], describing the limiting cases of the study: Fig. 2a for an empty flash chamber ("empty"), i.e., without a baffle; 2b for a flash chamber with the shortest baffle placed closest to the orifice; and 2c for a flash chamber with the highest baffle placed furthest from the orifice.

A more compact summary of the results is shown in the upper part of Fig. 3. The correlated shape of the free surface is the upper solid line. The broken line represents a typical computed streamline of the main flow (flash chamber with a baffle) which enters through the interstage orifice at the bottom right corner, is propelled by the baffle upwards towards the free surface, and then descends to exit through another orifice on the left bottom end of the stage and thus to enter the next stage. The lower solid line is a typical computed isobar, which is directly convertible to a saturation temperature (for pure liquids) or boiling-point (for solutions) isotherm for the liquid in use. That isobar (i.e., saturation or boiling-point isotherm) shows how the pressure is significantly reduced as the liquid ascends towards the free surface due both to a proportionate decrease in hydrostatic head and

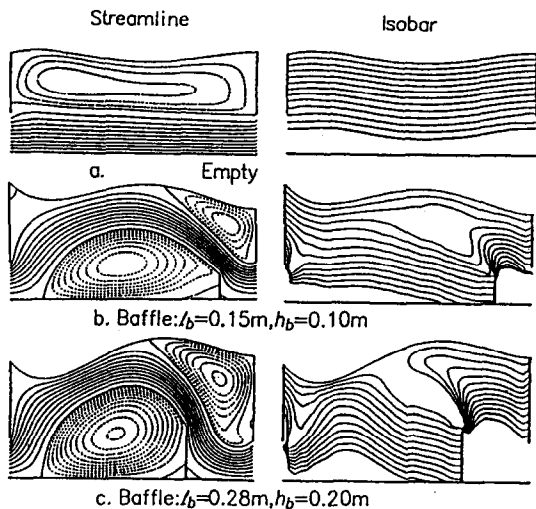


Fig. 2. Some computed streamline and isobar maps for a flash chamber without a baffle (top) and flash chambers with baffles of different height and location (center and bottom), from [4], for  $W = 8.7 \times 10^5$  kg/h  $\cdot$  m.

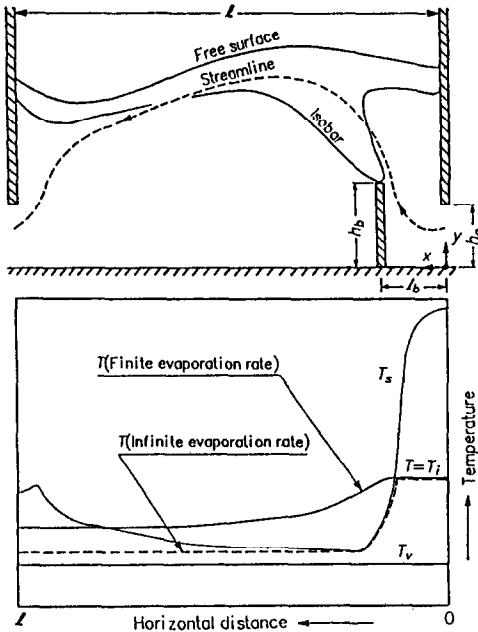


Fig. 3. A compact presentation of the computed streamlines, isobars, and the associated temperature fields in a flash chamber with a baffle, from [4].

to acceleration; then it is increased due both to an increase in hydrostatic head and to deceleration, descent towards the outlet orifice, and impingement on the left-hand interstage divider. Finally it decreases once again due to acceleration into the outlet orifice. A typical corresponding saturation temperature profile ( $T_s$ ) is shown in the lower part of Fig. 3. This also shows the vapor temperature above the liquid ( $T_v$ ), usually constant in such a flash chamber, and the mean liquid temperature for finite and infinite evaporation rates. The entering liquid temperature ( $T_i$ ) is assumed to be uniform at the inlet orifice because of the contraction and mixing of the flow at the upstream stage outlet. Evaporation and consequent cooling of the liquid start when the saturation temperature or boiling point determined by the pressure drops to the entering liquid temperature ( $T_i$ ), here when  $T_s < T_i$  and the liquid reaches the superheated state. Evaporation stops and the liquid temperature ( $T$ ) remains constant when  $T$  drops below  $T_s$  and the liquid returns to the subcooled state.

In the case of infinite evaporation rate, shown in Fig. 3 just to demonstrate a limiting (but unattainable) case, the liquid temperature instantly drops down to  $T_s$  as long as evaporation proceeds, and then remains constant

when  $T_s$  rises above it due to the above-explained rise in liquid pressure further downstream from the baffle. The real situation is characterized in Fig. 3 by the temperature profile for finite evaporation rates.

### *The temperature decrease rate of a flashing liquid*

Experiments with water flashing into a space of lower saturation pressure from an orifice of 60 mm diameter at 60°C [6] and long tubes of 3.5–8.2 mm diameter in the temperature range of 40–80°C [7,8] were examined here and correlated well with the equation

$$dT/dt = -k(T - T_s)^2 \quad (3)$$

where  $k$  is a flash evaporation coefficient with values of 3.9, 4.6, and 5.6  $(s \cdot K)^{-1}$  for  $T_i = 40^\circ C$ ,  $60^\circ C$ , and  $80^\circ C$ , respectively. More specifically, the value of  $k = 4.6 (s \cdot K)^{-1}$  was determined from the data of [6] (at  $60^\circ C$ ) and the values of  $k = 3.9$  and  $5.6 (s \cdot K)^{-1}$  (at  $40^\circ C$  and  $80^\circ C$ , respectively) were estimated from the temperature dependence of the experimental data of [7] and [8] based on  $k = 4.6 (s \cdot K)^{-1}$  at  $60^\circ C$ . Since typically nuclei for bubble formation are present in the liquid entering the flash chamber, the experience is that flash evaporation indeed starts at  $t = 0$ , i.e. (at the position where the entering liquid reaches the superheated state).

### *The method for calculating the NETD*

Since the rate of temperature change depends only on  $t$  and  $T_s$ , and not on the shape of the streamline, it was assumed that Eqn. (3) is valid along each streamline of the liquid flowing through the flash chamber. One may regard Eqn. (3) as having been developed indeed for a “single streamline” of a flashing liquid since the experiments basically were for flashing from round, 60 mm diameter jets. The previous paper [4] provided the location and magnitudes of the streamlines and of the saturation temperature or the boiling point isotherms in that liquid for isothermal flow. The main flow domain was divided here into 20 streamlines in the non-dimensional range of  $0 \leq \Psi \leq 1$  (i.e.,  $\Delta\Psi = 0.05$ ). Fig. 4 shows two streamlines,  $\Psi = (n-1)\Delta\Psi$  and  $n\Delta\Psi$ , on the  $x$ - $y$  computational grid. The fluid temperature along the streamline  $\Psi = (n-1/2)\Delta\Psi$  represents in this model the temperature of the fluid filament between the two neighboring streamlines.

Eqn. (3) is solved to obtain the temperature drop along each streamline  $(n-1/2)\Delta\Psi$  in intervals described typically by points A and B in Fig. 4 by

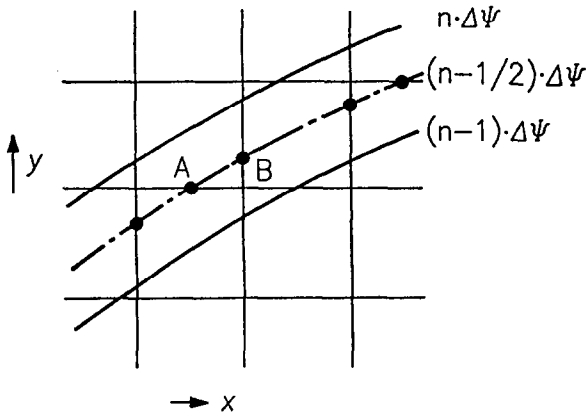


Fig. 4. A detail of the computational grid used to determine the *NETD*.

using the Runge-Kutta-Gill method as long as  $T \geq T_s$ . Wherever  $T < T_s$ , it is assumed that  $dT/dt = 0$ . The time interval ( $\Delta t_{AB}$ ) for the fluid to flow from A to B was calculated by

$$\Delta t_{AB} = \frac{2 \sqrt{(x_B - x_A)^2 + (y_B - y_A)^2}}{\sqrt{u_A^2 + v_A^2} + \sqrt{u_B^2 + v_B^2}} \quad (4)$$

where  $u$  and  $v$  are the liquid velocity components in the  $x$  and  $y$  directions, respectively.

Naming the liquid temperature for each of these mid-streamlines at the flash chamber end as  $(T_{x=1})_n$ , the average temperature of the liquid at the exit,  $T_o$ , is calculated by

$$T_o = \sum_{n=1}^{20} (T_{x=1})_n \Delta \Psi \quad (5)$$

Knowing  $T_o$ , *NETD* is calculated from Eqn. (1) for given values of  $T_v$  and  $T_i$  (i.e.,  $T_v$  and  $\Delta T_s$ ).

#### *The range of the numerical analysis*

Corresponding with the flash chamber geometry of the experimental apparatus, computations have been made for  $l = 1.0$  m,  $h_o = 0.15$  m,  $l_b = 0.15 - 0.28$  m,  $h_b = 0 - 0.20$  m and the standard height of liquid in this study



$h=y_s=0.40$  m at  $x=x_s=0.775$  m (Fig. 1) for water flow rates  $W=(4.3)10^5$  to  $(8.7)10^5$  kg/h·m width, as shown in Table I.

TABLE I

Flash chamber geometry and conditions of the numerical analysis and of the experimental study

| $l_b$ [m] | $h_b$ [m] | $W$ [ $10^5$ kg/h·m] | $T_w(T_o)$ [°C] |
|-----------|-----------|----------------------|-----------------|
| -         | 0         | 4.3, 6.5, 8.7        | 40, 60, 80      |
| 0.15      | 0.20      | 4.3, 6.5, 8.7        | 40, 60, 80      |
| 0.22      | 0.20      | 8.7                  | 40, 60, 80      |
| 0.28      | 0.20      | 8.7                  | 40, 60, 80      |
| 0.15      | 0.15      | 8.7                  | 40, 60, 80      |
| 0.15      | 0.10      | 8.7                  | 40, 60, 80      |

(  $l=1.00$  m,  $h_o=0.15$  m) (  $x_s=0.775$  m,  $y_s=0.40$  m)

## THE EXPERIMENTAL APPARATUS AND PROCEDURE

The experimental apparatus is a three-stage flash evaporator, the second stage being the test stage, with each stage consisting of a flash chamber and a condenser. The three flash chambers, each 1.0 m long and 0.046 m wide, are separated from each other by vertical plate dividers; a rectangular orifice, 0.15 m high and 0.046 m wide, for the transport of liquid is located at the bottom of each divider.

Air is evacuated from the system prior and during the evaporation experiments. Separate loops are used for the condenser cooling water and for the flash evaporator. The temperature of the water in each of these two loops can be regulated independently. Before admission to the lowest temperature stage condenser, the condenser coolant can be cooled by means of a chiller to a level necessary for condensation of the vapor produced in that stage. Heat is added to the flash evaporator water in an external heater to bring its inlet temperature to the desired level. Next the hot water is admitted to the highest temperature stage where it flashes and continues to flash in the second (test) stage and the third stage because of exposure to decreasing vapor pressure in these stages.

Temperature measurements were made with copper-constantan sheathed thermocouples, 1 mm diameter. They were calibrated in reference to a standard thermometer, and the measurement accuracy was  $\pm 0.05^\circ\text{C}$ .

The flow rate of the water was measured by a rotameter which was calibrated to an accuracy of  $\pm 3\%$  through the full range of water temperatures used in the experiment.

A more detailed description of the experimental apparatus and procedure and of the error analysis is provided in [4]. The experimental conditions correspond to those used in the numerical analysis and are presented in Table I.

## RESULTS

Fig. 5 presents comparisons of the computed non-equilibrium temperature difference ( $NETD$ ) for  $k=4.6 \text{ (s}\cdot\text{K)}^{-1}$  with experimental measurements. The limiting case for infinite evaporation rate  $k=\infty$ , as well as the asymptote which characterizes the limiting case of no evaporation in the flash chamber ( $NETD=\Delta T_s$ ), are also shown. For any  $W$ ,  $l_b$ , and  $h_b$ ,  $NETD$  decreases as  $T_o$  or  $T_v$  are increased due to an increase in the saturation pressure difference corresponding to the specified temperature difference, that is, due to an increase in the depth of superheated liquid region.

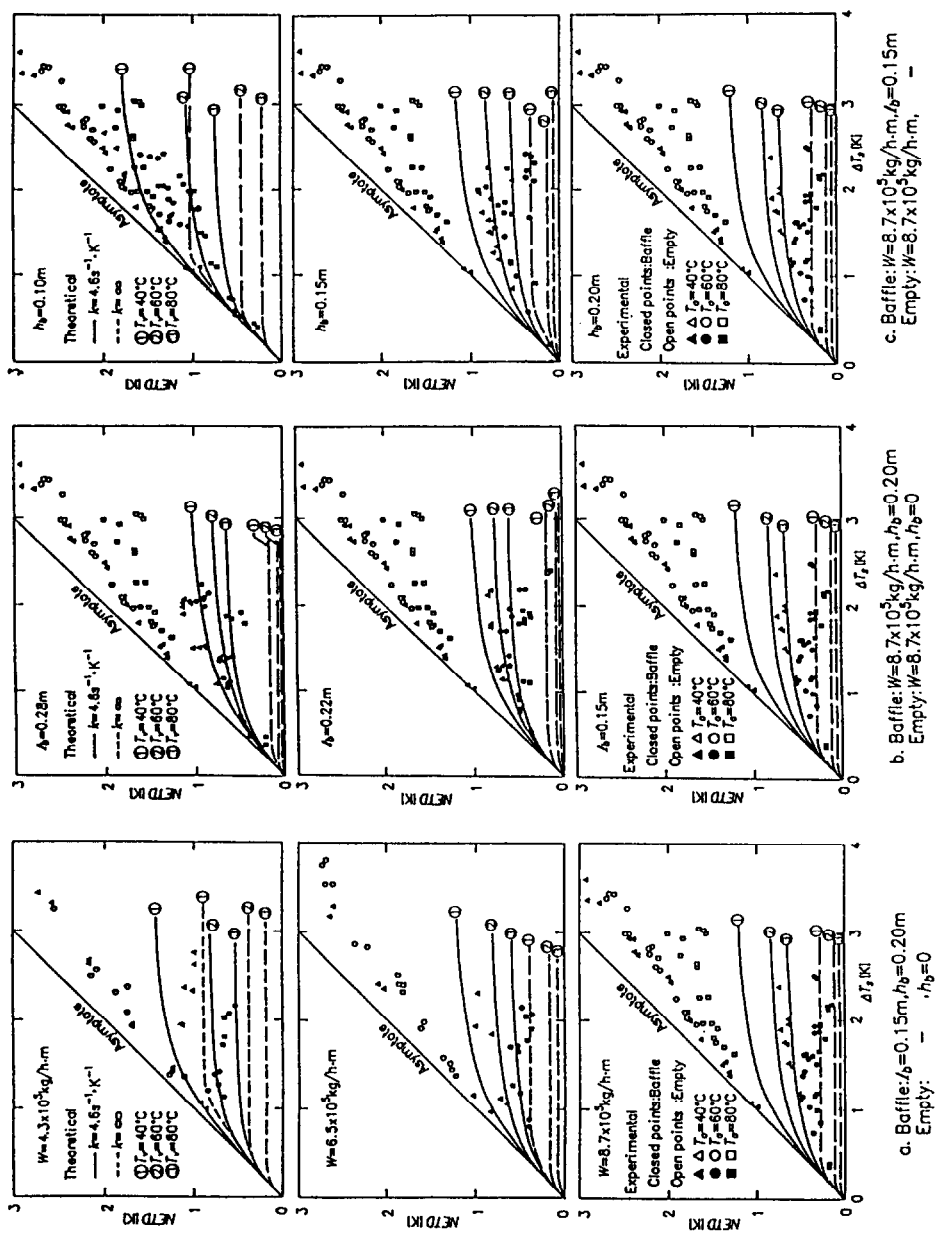


Fig. 5. Comparison of the computed non-equilibrium temperature difference (NETD) with the experimental measurements.

Fig. 5a illustrates the effect of the mass flow rate  $W$  for fixed values of  $l_b$  and  $h_b$ . Despite the decrease in the residence time of the liquid in the flash chamber,  $NETD$  slightly decreases with increasing  $W$  because the liquid is propelled higher and thus becomes exposed to a lower pressure region downstream from the baffle plate as shown in our previous paper [4]. Figs. 5b and 5c illustrate the effect of  $l_b$  and  $h_b$ , respectively, for fixed values of  $W$ . Inspection of the figures reveals that the installation of a higher baffle plate nearer to the inlet orifice reduces  $NETD$ .

The experimental results agree substantially with the calculated values with the exception of one extreme case,  $W=(8.7)10^5$  kg/(h·m),  $l_b=0.15$  m,  $h_b=0.20$  m, where a break-up of the liquid stream was observed downstream of the top of the baffle plate. In this case flash evaporation becomes similar to that in a superheated liquid jet (i.e., spray flash evaporation [7,8]). The agreement is also less satisfactory when the flash-down is large, for example, when  $\Delta T=(\Delta T_s=NETD)>1.5$  K, primarily because the flow is more strongly affected by bubble formation and movement and thus the non-evaporative isothermal flow model becomes inapplicable. It may be noted that in high-performance MSF evaporators (having typically more than 35 stages) or in those utilizing low temperature waste heat, the flash-down  $\Delta T \approx 2$  K.

The computations were repeated for  $k=3.9$  and  $k=5.6$  (s·K)<sup>-1</sup> (at 40°C and 80°C, respectively), and it was found that the values of  $NETD$  as a function of  $\Delta T_s$  were negligibly different from those computed with  $k=4.6$  (s·K)<sup>-1</sup> in the full range of parameters considered in this study.

In the case of the empty flash chamber ("empty") (i.e., without a baffle), the measured  $NETD$  is much larger, up to about an order of magnitude than that measured for a flash chamber with a baffle ("baffle"), as indicated by the open points in Fig. 5. This can be explained from the experiments [1] and the predictions of our isothermal model [4] as follows:

1. The submerged hot jet of liquid emanating from the inlet orifice was observed to be overlaid by a cold recirculating flow (see Fig. 2a), resulting in the fall of its temperature, and to flow along the bottom of the flash chamber to the outlet orifice;
2. A portion of that wall-jet enters into the recirculating flow and travels back upstream near the free surface, accompanied by weak flash evaporation; and
3. Another portion of it flows out of the flash chamber without liberation of vapor.

To attain low *NETD* it is necessary to elevate the entering hot liquid towards the free surface (where the saturation temperature is lower) as rapidly as possible with minimal temperature loss. This assures high flash evaporation rates. In contrast, without a baffle in the flash chamber, much of the hot liquid remains near the flash-chamber floor, submerged by the recirculating flow; and furthermore, the portion of the liquid which is elevated to the free surface cools down (by mixing with that recirculating flow) before it reaches close to the free surface. This deficiency is particularly acute in short stages, and a baffle (or another device which serves to propel the entering liquid upwards effectively) is indispensable if low *NETD* is desired.

## CONCLUSIONS

Within the range of parameters investigated in this study, the non-equilibrium temperature difference in a flash chamber with a baffle (sill) can be predicted well by a combination of a numerical analysis model for isothermal turbulent flow in the flash chamber [4] with an empirical correlation of the temperature decrease rate in flash evaporation, as long as a combination of a high flow rate and a large flash-down temperature difference does not cause a significant disruption of the flow by separation or ebullition. From all examined cases this occurred here only for (1) the highest flow rate,  $(8.7)10^5$  kg/h · m, with the highest baffle placed closest to the orifice ( $l_b=0.15$  m,  $h_b=0.20$  m) or (2) for a flash-down of  $T_i - T_o > 1.5$  K.

The delivery of the entering hot liquid to the flashing zone near the free surface was found to be a very effective way to enhance flash evaporation rates. Installation of a simple baffle plate in the flash chamber has accomplished this purpose well. Furthermore, a baffle must be installed in relatively short (1 m) flash chambers for efficient vapor release.

The non-equilibrium temperature difference, *NETD*, is significantly reduced as the temperature of the liquid or the baffle height is increased and less significantly reduced as the flow rate is increased or the baffle distance decreased.

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