CONTRIBUTIONS TO THE DEVELOPMENT OF A MULTI-STAGE FLASH/FLUIDIZED BED EVAPORATOR (MSF/FBE)

Proefschrift

Ter verkrijging van de graad van doctor in de technische wetenschappen aan de Technische Hogeschool Delft, op gezag van de Rector Magnificus, prof. ir. B.P.Th. Veltman voor een commissie aangewezen door het College van Dekanen, te verdedigen op donderdag 29 november 1984 om 14.00 uur.

door

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## CONTENTS

<table>
<thead>
<tr>
<th>LIST OF SYMBOLS</th>
<th>page</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>0.7</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>SUMMARY</th>
<th>page</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>0.14</td>
</tr>
</tbody>
</table>

### CHAPTER 1  INTRODUCTION AND MOTIVATION

1.1. General aspects

1.1.1. New developments

1.1.1.1. Vertical Tube Foam Evaporation

1.1.1.2. Low temperature Vapour Compression

1.1.1.3. Horizontal Aluminium Tubes Multi-Effect evaporator

1.1.1.4. Falling Film Multi-Stage Flash evaporator

1.1.1.5. Direct Contact Condensation Multi-Stage Flash distillation

1.1.1.6. Multi Effect Stack type of distillation process

1.2. The Multi-Stage Flash/Fluidized Bed Evaporator

1.3. Method of approach for the investigations

REFERENCES CHAPTER 1

### CHAPTER 2  WALL TO LIQUID HEAT TRANSFER IN A LIQUID-SOLID FLUIDIZED BED HEAT EXCHANGER

2.1. Introduction

2.1.1. Heat transfer in an MSF/FBE

2.1.2. Experimental approach

2.2. Test module for heat transfer measurements

2.2.1. The heater circuit

2.2.2. The heat exchanger circuit

2.2.3. The cooler circuit

2.2.4. Instrumentation of the test module

2.2.4.1. Volume flows

2.2.4.2. Pressure indicators

2.2.4.3. Temperatures

2.2.5. Data acquisition, storage and computation

2.2.6. Calibration

2.3. Background and principles of the measuring method A

2.3.1. The local heat transfer coefficient

2.3.2. The NUSSELT correlation

2.3.3. The calculation of the local heat transfer coefficient

2.3.4. The averaged heat transfer calculation over the entire tube length

2.4. Results obtained with measuring method A

2.4.1. Local heat transfer coefficients

2.4.2. Average heat transfer coefficients

<table>
<thead>
<tr>
<th>REFERENCES</th>
<th>page</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>1.20</td>
</tr>
</tbody>
</table>
2.5. Measuring method B
   2.5.1. Principle of the measuring method B
   2.5.2. The local heat transfer coefficient calculation
   2.5.3. The heat transfer calculation averaged over a part of the tube length

2.6. Results obtained with the measuring method B
   2.6.1. Local heat transfer coefficients
   2.6.2. Average heat transfer coefficients

2.7. Calculation of the error in the local heat transfer coefficient

2.8. Determination of the error in the porosity

2.9. Measurement of temperatures with imbedded thermocouples

2.9.1. FIESTA application for the determination of \( x_f \)

2.9.2. Determination of \( x_f \)

2.9.3. Analysis of the error in \( x_f \)

2.10. Remarks and conclusions

REFERENCES CHAPTER 2

CHAPTER 3 SCALE AND FOULING BEHAVIOUR OF A LIQUID-SOLID FLUIDIZED BED HEAT EXCHANGER

3.1. Introduction

3.2. Scale control

3.2.1. Polyphosphate compound control

3.2.2. Acidity control

3.2.3. Polymer additives control

3.2.4. Calcium sulphate scale control

3.3. The fluidized bed heat exchanger

3.4. Mechanism of crystallisation

3.4.1. Nucleation

3.4.2. Crystal growth

3.5. Sludge and fouling influence

3.6. Surface condition

3.7. Particle velocity in liquid-solid fluidized bed

3.7.1. Particle velocity model approach

3.7.2. Empirical relations

3.8. Scale control experiments

3.8.1. Schie canal water

3.8.2. Sea water

3.8.3. Artificial sea water calcium sulphate experiments

3.8.3.1. Experiments

3.8.3.2. Results and discussion

REFERENCES CHAPTER 3
CHAPTER 4  FLASH STAGE GEOMETRY

4.1. Introduction 4.0
4.2. Entrainment 4.1
4.3. Equilibration 4.3

4.3.1. Literature survey 4.5
4.3.2. Equilibration model 4.11

4.4. Experimental apparatus 4.14

4.4.1. Description of the test module 4.15
4.4.2. Operation of the test module 4.19
4.4.3. Calculation procedure 4.21
4.4.4. Pressure measurements 4.21
4.4.5. Temperature measurements 4.24
4.4.6. Salt concentration measurements 4.25
4.4.7. Brine level measurements 4.26
4.4.8. Mass flow measurements 4.26

4.5. Experimental results 4.29

4.5.1. Introductory remarks 4.29
4.5.2. Entrainment 4.32

4.5.2.1. Basic geometry A 4.32
4.5.2.2. Basic geometry B 4.35
4.5.2.3. Basic geometry C 4.39

4.5.3. Equilibration 4.41

4.6. Some design criteria and concluding remarks 4.44

4.6.1. Entrainment separation 4.44
4.6.2. Temperature loss 4.50
4.6.3. Equilibration 4.53

REFERENCES CHAPTER 4 4.56

CHAPTER 5  OPERATIONAL ASPECTS OF A MULTI-STAGE FLASH/FLUIDIZED BED EVAPORATOR (MSF/FBE) 5.0

5.1. Introductory remarks 5.1
5.2. Construction of the pilot plant 5.1

5.2.1. Plant lay-out 5.1
5.2.2. Manufacturing and erection of the pilot plant 5.9

5.3. Instrumentation of the pilot plant 5.11

5.3.1. Instrumentation for measurements 5.11
5.3.2. Instrumentation for process control 5.12

- 0.5 -
5.4. Specific items of investigation

5.4.1. Hydraulic behaviour
5.4.2. Automatic control of an MSF/FBE
5.4.3. Constructional changes

5.4.3.1. Flash device
5.4.3.2. Inspection and measurement features
5.4.3.3. Throttling device
5.4.3.4. Stirring device

5.4.4. Non-condensables venting

5.4.4.1. Non-condensables venting system in the pilot plant
5.4.4.2. Operational experience

5.4.5. Heat transfer coefficient

5.5. Design developments

5.5.1. 500 m$^3$/day demonstration MSF/FBE
5.5.2. 100 m$^3$/day commercial MSF/FBE
5.5.3. 4000 m$^3$/day MSF/FBE (design)

REFERENCES CHAPTER 5

ACKNOWLEDGEMENTS

SAMENVATTING
<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>A</td>
<td>area of heat transfer surface (Ch.2)</td>
<td>m²</td>
</tr>
<tr>
<td>A</td>
<td>surface area of the crystal (Ch.3)</td>
<td>m²</td>
</tr>
<tr>
<td>A</td>
<td>total interface surface per unit of liquid mass (Ch.4)</td>
<td>m²/kg</td>
</tr>
<tr>
<td>A_t</td>
<td>cross area of the interstage orifice</td>
<td>m²</td>
</tr>
<tr>
<td>a</td>
<td>cross section area of the heat exchanger tube</td>
<td>m²</td>
</tr>
<tr>
<td>C</td>
<td>nucleation constant</td>
<td>m/kg</td>
</tr>
<tr>
<td>C*</td>
<td>equilibrium saturation concentration</td>
<td>mol/m³</td>
</tr>
<tr>
<td>C</td>
<td>vapour bubble growth coefficient</td>
<td>m/[s⁰.⁵]</td>
</tr>
<tr>
<td>C²</td>
<td>constant</td>
<td></td>
</tr>
<tr>
<td>c_d</td>
<td>constant in relation for vapour bubble growth</td>
<td>m/[³⁰C s⁰.⁵]</td>
</tr>
<tr>
<td>c_d∞</td>
<td>drag coefficient</td>
<td></td>
</tr>
<tr>
<td>c_i</td>
<td>drag coefficient of a sphere in a stagnant infinite liquid</td>
<td></td>
</tr>
<tr>
<td>c_i</td>
<td>solute concentration in the solution at the crystal solution interface</td>
<td>mol/m³</td>
</tr>
<tr>
<td>c_p</td>
<td>mean specific heat of the liquid</td>
<td>J/kg ³⁰C</td>
</tr>
<tr>
<td>c_p</td>
<td>specific heat of the liquid</td>
<td>J/kg ³⁰C</td>
</tr>
<tr>
<td>P_l</td>
<td>specific heat of the liquid at location z</td>
<td>J/kg ³⁰C</td>
</tr>
<tr>
<td>c_pz1</td>
<td>specific heat of the liquid at location z</td>
<td>J/kg ³⁰C</td>
</tr>
<tr>
<td>c_pz2</td>
<td>specific heat of the liquid at location z</td>
<td>J/kg ³⁰C</td>
</tr>
<tr>
<td>c_w</td>
<td>friction coefficient</td>
<td></td>
</tr>
<tr>
<td>D</td>
<td>equivalent hydraulic diameter of evaporator stage</td>
<td>m</td>
</tr>
<tr>
<td>D_i</td>
<td>inner diameter of the heat exchanger tube</td>
<td>m</td>
</tr>
<tr>
<td>D_m,a_n</td>
<td>deflection of mercury column for atmospheric measurement and for nᵗʰ stage</td>
<td>m</td>
</tr>
<tr>
<td>D_m,b_n</td>
<td>deflection of mercury column for actual measurement and for nᵗʰ stage</td>
<td>m</td>
</tr>
<tr>
<td>D_O</td>
<td>outer diameter of the heat exchanger tube</td>
<td>m</td>
</tr>
<tr>
<td>D_T</td>
<td>temperature between the inside wall measuring point and the bulk of the liquid, averaged over a certain tube length</td>
<td>³⁰C</td>
</tr>
<tr>
<td>D_v</td>
<td>coefficient of diffusion of steam in non-condensible gases</td>
<td>m²/s</td>
</tr>
<tr>
<td>d_i</td>
<td>inner diameter of the throttling device tube</td>
<td>m</td>
</tr>
<tr>
<td>d_d</td>
<td>diameter of entrained droplet</td>
<td>m</td>
</tr>
<tr>
<td>d_p</td>
<td>particle diameter in the fluidized bed</td>
<td>m</td>
</tr>
</tbody>
</table>
E diffusion coefficient $m^2/s$
$E_x$ mean perpendicular component of the kinetic energy $kg \ m^2/s^2$
$e_x$ expansion factor of the fluidized bed $-$
$f_i$ fouling factor for the inner surface of the condenser tubes $m^2 \ 0\degree\ C \ s/J$
$f_o$ fouling factor for the outer surface of the condenser tubes $m^2 \ 0\degree\ C \ s/J$
$\Delta G$ overall excess free energy $J$
$\Delta G_{crit}$ free energy change associated with homogeneous nucleation for critical radius of nucleus $J$
$\Delta G'_{crit}$ free energy change associated with heterogeneous nucleation for critical radius of nucleus $J$
$\Delta G_s$ excess free energy between surface of a particle and the bulk of the particle $J$
$\Delta G_v$ excess free energy between a very large particle and the solute per unit of volume $J/m^3$
$q_g$ acceleration of gravity $m/s^2$
$H$ evaporator stage height (Ch.4) $m$
$H$ undisturbed length of the condensate film (Ch.5) $m$
$H_l$ brine level $m$
$H_{mf}$ bed height at incipient fluidization point $m$
$H_n$ brine height in stage n $m$
$H_S$ total stage height $m$
$h$ overall heat transfer coefficient $J/m^2 \ 0\degree\ C \ s$
$h^*$ representative passage dimension $m$
$h'$ vapour aperture under the baffle plate $m$
$h_{b,n-1}$ enthalpy of brine in stage n-1 entering the stage n $J/kg$
$h_{d,n}$ enthalpy of vapour produced in stage n $J/kg$
$h_i$ overall heat transfer coefficient related to the inside surface $J/m^2 \ 0\degree\ C \ s$
$h_i$ overall heat transfer coefficient related to the inside surface and averaged over a certain tube length $J/m^2 \ 0\degree\ C \ s$
$h_{lg}$ latent heat of evaporation $J/kg$
$h_o$ overall heat transfer coefficient related to the outside surface $J/m^2 \ 0\degree\ C \ s$
$I$ momentum transfer from the particle to the wall $kg/m \ s$
>ID internal diameter of the throttling device tubes $m$
$K$ overall heat transfer coefficient based on the outer exchanger surface $J/m^2 \ 0\degree\ C \ s$
$K_G$ overall crystal growth coefficient $m/s$
$k_r$ rate constant for surface reaction $m/s$
$K_d$ coefficient of mass transfer by diffusion $m/s$
\( L \)  
flash stage length (Ch.4) \( m \)

\( L_1 \)  
flash length from entrance of stage till development of dispersed jet \( m \)

\( L_B \)  
length of baffle plate \( m \)

\( L_{1,1} \)  
length of a fluidized bed \( m \)

\( L_{m,b} \)  
length of the throttling device tubes \( m \)

\( L_o \)  
mercury column during actual measurement \( m \)

\( L_{w,a,n} \)  
initiation length in evaporation process \( m \)

\( L_{w,b} \)  
water column in manometer during atmospheric measurement and for \( n \)th stage \( m \)

\( L_{w,b,n} \)  
water column during actual measurement \( m \)

\( l_{tot} \)  
water column in manometer for measurement in the \( n \)th stage \( m \)

\( m_p \)  
effective heat exchanger tube length (Ch.2) \( m \)

\( m_{P,N} \)  
mass of solids deposited in time \( t \) \( kg \)

\( m_N \)  
mass of particles \( kg \)

\( N \)  
rate of decrease of bubble population \( 1/s \)

\( n \)  
exponent in equation for swarm of particles (Ch.2, Ch.5) \( - \)

\( n \)  
overall growth rate order (Ch.3) \( - \)

\( n \)  
evaporation index (Ch.4) \( m^3\ C/s/\ [kg \ J] \)

\( p \)  
volume fraction of the particles in a fixed bed \( - \)

\( \Delta p_{\text{temp}} \)  
error in pressure due to temperature differences along the manometer legs \( N/m^2 \)

\( p_a \)  
atmospheric pressure \( N/m^2 \)

\( p_{n,n} \)  
saturated vapour pressure in stage \( n \) \( N/m^2 \)

\( p_r \)  
manometer reading \( \text{bar} \)

\( p_{t,m} \)  
total vapour pressure of the mixture \( N/m^2 \)

\( \Delta p \)  
total pressure difference for insterstage brine transport \( N/m^2 \)

\( \Delta p_1 \)  
pressure drop over a fluidized bed with throttling device at the incipient fluidization point \( N/m^2 \)

\( \Delta p_2 \)  
pressure drop over a fluidized bed with throttling device in fully fluidized situation \( N/m^2 \)

\( \Delta p_p \)  
pressure drop over the fluidized bed \( N/m^2 \)

\( \Delta p_{\text{loss}} \)  
pressure loss between saturation pressure in the stage and the saturation pressure at the condenser inlet \( N/m^2 \)

\( \Delta p_{\text{man}} \)  
error in pressure and due to manometer slide gauge reading \( N/m^2 \)

\( \Delta p_{\text{thr}} \)  
pressure loss over the throttling device at the inlet of the tube \( N/m^2 \)
ΔP_w  pressure loss over the fluidized bed due to the weight of the bed  N/m²
P_1  partial vapour pressure of steam  N/m²
P_2  partial vapour pressure of non-condensable gases  N/m²
R_1  gas constant for steam  J/[°K mol]
R_b  radius of vapour bubble  m
Re_{dp}∞  Reynolds number for single particle in an infinite volume  -
r  radius of scale nucleus  m
r_c  critical radius of scale nucleus  m
ΔT  temperature difference between brine temperature and saturation temperature  °C
ΔT_C  mean temperature difference over the condensate film  °C
ΔT_O  initial superheat of the brine entering the evaporator stage  °C
T.D.S.  Total Dissolved Solids  ppm
T_a  temperature of the steam  °C
T_b  brine temperature  °C
T_{b, en}  brine temperature at the entrance of the evaporator stage  °C
T_{b, L_1}  brine temperature at the start of the dispersed jet  °C
T_{b, l}  liquid temperature in the bulk of the liquid  °C
T_{b, l, z}  liquid temperature in the bulk at location z  °C
T_{b, l, z_1}  liquid temperature in the bulk at location z_1  °C
T_{b, l, z_2}  liquid temperature in the bulk at location z_2  °C
T_{i, x®, z}  temperature at the center of the thermocouple at the inner side of the tube wall  °C
T_{l, in}  liquid temperature at the entrance of the heat exchanger tube  °C
T_{l, out}  liquid temperature at the exit of the heat exchanger tube  °C
T_{o, x®, z}  temperature of the center of thermocouple at the outer side of the tube wall  °C
T_S  saturation temperature at the surface of the condensate  °C
T_v  vapour temperature of the mixture  °C
T_{v, sat}  saturation temperature of the vapour  °C
T_W  mean wall temperature at location z  °C
T_{W, s}  temperature at the surface of the heat exchanger tube (Ch.5)  °C
T_{W, c, z}  wall temperature condensation side at location z  °C
wall temperature at the liquid side $T_{w,l}$ °C
wall temperature liquid side at location $z$ $T_{w,l,z}$ °C
temperature of the center of thermocouple at location $z$ $T_{x,z}$ °C
time $t$ s
velocity of the vapour mixture $u_{vm}$ m/s
terminal velocity of a single sphere in an infinite volume $u_i$ m/s
terminal velocity of a single sphere in the confined volume of the tube $u$ m/s
liquid velocity in the minimum cross section $u_{l,s}$ m/s
superficial liquid velocity $u_{l,s,z}$ m/s
superficial liquid velocity at location $z$ $u_{l,s,z_2}$ m/s
incipient fluidization velocity $u_{mf}$ m/s
terminal velocity of a swarm of spheres in a stagnant liquid $u_{sw}$ m/s
vapour velocity $v_v$ m/s
mean bulk fluid velocity $v_v^-$ m/s
mean impact velocity perpendicular to the wall $v_l$ m/s
horizontal particle velocity $v_h$ m/s
brine velocity in the interstage orifice $v_{1_{o}}$ m/s
radial particle velocity $v_r$ m/s
resulting particle velocity $v_t$ m/s
tangential particle velocity $v_{tan}$ m/s
vertical particle velocity $v_v$ m/s
width of the evaporator stage $w_0$ m
fictive distance for temperature measurements $x_f$ m
undisturbed length of the condensate film $z$ m
surface reaction exponent (Ch.3) $z$ -
Greek symbols

\( \alpha \) condensation heat transfer coefficient with the presence of non-condensable gases \( \text{J/m}^2 \text{ } ^0\text{C } \text{s} \)

\( \alpha_c \) condensation heat transfer coefficient without the presence of non-condensable gases (equation 5-8) \( \text{J/m}^2 \text{ } ^0\text{C } \text{s} \)

\( \alpha_c \) condensation heat transfer coefficient \( \text{J/m}^2 \text{ } ^0\text{C } \text{s} \)

\( \alpha_{c,z} \) condensation heat transfer coefficient at location \( z \) \( \text{J/m}^2 \text{ } ^0\text{C } \text{s} \)

\( \alpha_l \) wall to liquid heat transfer coefficient \( \text{J/m}^2 \text{ } ^0\text{C } \text{s} \)

\( \bar{\alpha}_{w,l} \) wall to liquid heat transfer coefficient, averaged over a certain tube length \( \text{J/m}^2 \text{ } ^0\text{C } \text{s} \)

\( \alpha_{w,l} \) wall to liquid heat transfer coefficient \( \text{J/m}^2 \text{ } ^0\text{C } \text{s} \)

\( \alpha_{w,l,z} \) wall to liquid heat transfer coefficient at location \( z \) \( \text{J/m}^2 \text{ } ^0\text{C } \text{s} \)

\( \beta \) approach to equilibrium

\( \beta_1 \) equilibration factor at the start of the dispersed jet

\( \beta_2 \) equilibration factor in the dispersed jet area

\( \delta_a \) averaged entrained particle diameter \( \text{m} \)

\( \delta_{ob} \) observed entrained particle diameter \( \text{m} \)

\( \varepsilon \) porosity of the fluidized bed

\( \varepsilon_m \) mean porosity

\( \varepsilon_{mf} \) fluidized bed porosity at incipient fluidization point

\( \zeta \) pressure loss coefficient for combined entrance losses and discharge losses of the throttling device tubes

\( \eta_c \) mean dynamic viscosity of the condensate \( \text{Pa s} \)

\( \eta_l \) dynamic viscosity of the liquid \( \text{Pa s} \)

\( \eta_v \) dynamic viscosity of vapour \( \text{Pa s} \)

\( \eta_{vm} \) dynamic viscosity of the vapour mixture \( \text{Pa s} \)

\( \Delta \theta \) saturation temperature between two evaporator stages \( ^0\text{C} \)

\( \theta_{in} \) liquid temperature at the entrance of heat exchanger tube \( ^0\text{C} \)

\( \theta_{out} \) liquid temperature at the outlet of heat exchanger tube \( ^0\text{C} \)

\( \lambda \) wall friction factor

\( \lambda_c \) thermal conductivity of the condensate \( \text{J/m } ^0\text{C } \text{s} \)

\( \lambda_{CN} \) thermal conductivity of cupper nickel alloy \( \text{J/m } ^0\text{C } \text{s} \)

\( \lambda_l \) heat conductivity coefficient of the liquid \( \text{J/m } ^0\text{C } \text{s} \)

\( \lambda_t \) heat conductivity coefficient of the tube material \( \text{J/m } ^0\text{C } \text{s} \)

\( \lambda_w \) thermal conductivity of the heat exchanger tube material \( \text{J/m } ^0\text{C } \text{s} \)

\( \xi \) friction factor
\( \rho_c \) density of the condensate \( \text{kg/m}^3 \)

\( \rho_L \) density of the liquid \( \text{kg/m}^3 \)

\( \rho_1 \) density of the liquid \( \text{kg/m}^3 \)

\( \rho_m \) density of mercury \( \text{kg/m}^3 \)

\( \rho_s \) density of the fluidized bed particles \( \text{kg/m}^3 \)

\( \rho_v \) density of the steam \( \text{kg/m}^3 \)

\( \rho_{v,m} \) density of the vapour mixture \( \text{kg/m}^3 \)

\( \rho_w \) density of water in the manometer legs \( \text{kg/m}^3 \)

\( \rho_{z_1} \) density of the liquid at location \( z_1 \) \( \text{kg/m}^3 \)

\( \rho_{z_2} \) density of the liquid at location \( z_2 \) \( \text{kg/m}^3 \)

\( \sigma \) surface energy \( \text{N/m} \)

\( \sigma \) surface tension of droplet (Ch.4) \( \text{N/m} \)

\( \phi \) correction factor for homogeneous nucleation

\( \phi \) heat flux to the liquid \( \text{J/s} \)

\( \phi_l \) mass flow through the interstage orifice \( \text{kg/s} \)

\( \phi_m \) mass flow through the heat exchanger averaged over a certain tube length \( \text{kg/s} \)

\( \phi_{m_1} \) mass flow through the heat exchanger tube \( \text{kg/s} \)

\( \phi_{m_2} \) mass flow through the fluidized bed tubes \( \text{kg/s} \)

\( \phi_{m,n} \) mass flow of brine from stage \( n \) to \( n+1 \) \( \text{kg/s} \)

\( \phi_{m,n-1} \) brine mass flow from stage \( n-1 \) to stage \( n \) \( \text{kg/s} \)

\( \phi_{m,n} \) distillate mass flow produced in stage \( n \) \( \text{kg/s} \)

\( \phi_{v,n} \) vapour flow produced in evaporator stage \( \text{kg/s} \)

\( \phi_{v,v} \) vapour volume flow \( \text{m}^3/\text{s} \)

\( \phi_w \) heat flux to the liquid, averaged over a certain tube length \( \text{J/s} \)

\( \phi_{w,1} \) heat flux to the liquid per unit of length (averaged) \( \text{J/s} \)

\( \phi_{w,tot} \) total heat flux to the liquid in the heat exchanger tube \( \text{J/s} \)

\( \phi_{w,z} \) local heat flux to the liquid at location \( z \) \( \text{J/s} \)

\( \phi_z \) local heat flux to the liquid at location \( z \) \( \text{J/s} \)

\( \psi \) molecular weight fraction for vapour mixtures (Ch.5)

\( \psi_1 \) discharge coefficient

\( \omega_{ob} \) entrainment coefficient for observed particle diameter range

- 0.13 -
SUMMARY

This thesis deals with the development of the Multi-Stage Flash/Fluidized Bed Evaporator (MSF/FBE).

After a clarification is given of the Multi-Stage Flash process, used as a reference, in Chapter 1 the development of the MSF/FBE is put into perspective by reviewing other developments in thermal desalination. In addition the method of approach of presently reported research work is given in Chapter 1 and covers:

- wall-to-liquid heat transfer in a fluidized bed
- fouling and scaling behaviour in a fluidized bed
- equilibrium and entrainment in the evaporator stage
- constructional and operational aspects of an MSF/FBE.

In Chapter 2 wall-to-liquid heat transfer measurements are described based on two measurements principles:

A. The heat flux to the liquid in the tube is determined by an imbedded thermocouple in the heat exchanger tube wall and by application of the Nusselt correlation for condensation heat transfer on the outside of the tube.

B. The heat flux to the liquid in the tube is determined by application of two imbedded thermocouples, positioned in the same cross section of the tube at different distances of the tube wall surface.

The latter method is introduced after failure of the first one to produce reliable values for the wall-to-liquid heat transfer.

The description of the test module is elaborated by an explanation of the method used with imbedded thermocouples and the computation of a fictive distance $x_f$ for the imbedded thermocouples and used in the calculations.

Since fouling and scaling of the heat transfer surface is a major factor in thermal desalination Chapter 3 is devoted to these subjects. After a review of known chemical scale control systems applied in conventional thermal desalination the features with respect to fouling and scale control of a fluidized bed are discussed.
Scale formation and scale growth mechanisms are discussed after which the absence of fouling in the fluidized bed application and influence of surface roughness on scale formation are clarified.

Scale control experiments are conducted with highly polluted brackish surface water, sea water and artificial sea water and aimed for information about the maximum permissible operating temperature without scale adherence to the heat exchanger surfaces. The surface water experiments and a part of the sea water experiments are dedicated to calcium carbonate and magnesium hydroxide scales whereas the remaining of the (high temperature) tests with natural and artificial sea water are focussed on calcium sulphate scales.

After a literature survey with respect to equilibration and model approach most of Chapter 4 is dedicated to experimental work concerned with analysing the equilibration and entrainment phenomena in an MSF/FBE stage. Three different basic geometries A, B and C and for each a number of alternative interior forms and dimensions have been used in the experimental work and measurements are performed for a range of salt concentrations, temperature levels and temperature drops per stage.

Analyses of droplet behaviour in the vapour channel resulted in geometry proposals for flash stages of an MSF/FBE. The allowable temperature loss as a result of friction and velocity changes in the vapour path has been analysed and a mathematical function for the allowable temperature loss is proposed by the author.

Chapter 5 deals with the operational aspects of a first generation MSF/FBE. Plant layout of the 50 m³/day unit has been described as well as the hydraulic behaviour of parallel fluidized beds and the applied throttling device. Constructional changes as a result of initial operation are reviewed.

A measurement feature to obtain temperature readings from within the fluidized bed heat exchanger tubes under operating conditions is clarified and the measures taken to improve the non-condensable venting system are presented.
Developments of the MSF/FBE technology after the first generation 50 m³/day unit are reviewed and comprise of:

- 500 m³/day experimental unit on the island of Texel
- 100 m³/day commercial units (city of Amsterdam)
- 4000 m³/day design.
1.1. General aspects.

"The importance of salt removal from ocean water or other saline water resources reaches far beyond its mere technological aspects, because the availability of fresh water has a decisive effect on the pattern of human development" [1.1].

In most developed countries water is looked upon as a natural right, and many industries use vast quantities of water in their operations. The world wide ever-increasing demand for drinking water and process water of high purity necessitates the development of techniques to provide mankind with fresh water.

The increasing use of fresh water is due to an increasing population, rising living standards, industrialization, expansion of irrigation agriculture and the fact that an increasing fraction of the population lives in the arid and semiarid parts of the world.

**FIGURE 1-1.**
Growth of installed desalination capacity, world wide.
The oceans represent the largest water reservoir of the earth. Ocean water, however, contains on the average 3.5% of dissolved salts, a concentration that makes the water unsuitable for many uses in household and industry and for the irrigation.

The technique of turning sea or brackish water into fresh water is called desalination. The major desalination processes available are reverse osmosis, electrodialysis and distillation. The range of applicability of one process over the other is determined primarily by the state of the art of the process, the salinity and composition of the feedwater.

Figure 1-1 shows the growth of installed desalination capacity in the world. Distillation processes are the most widely used, accounting for about 85% of total capacity. Membrane processes account for about 15% of the total capacity [1.2].

Table 1-1 compiled from [1.2] gives the enumeration of the different desalination processes as of June 1980, including plants under construction.

<table>
<thead>
<tr>
<th>type of process</th>
<th>number of plants</th>
<th>plant capacity in 1000 m³/day</th>
</tr>
</thead>
<tbody>
<tr>
<td>DISTILLATION</td>
<td></td>
<td></td>
</tr>
<tr>
<td>multi-stage flash evaporator</td>
<td>415</td>
<td>4893</td>
</tr>
<tr>
<td>vertical tube multiple effect evaporator</td>
<td>127</td>
<td>336</td>
</tr>
<tr>
<td>submerged tube</td>
<td>127</td>
<td>82</td>
</tr>
<tr>
<td>single-stage flash evaporator</td>
<td>77</td>
<td>43</td>
</tr>
<tr>
<td>vapour compression</td>
<td>168</td>
<td>84</td>
</tr>
<tr>
<td>horizontal tube multiple effect</td>
<td>51</td>
<td>86</td>
</tr>
<tr>
<td>subtotal:</td>
<td>965</td>
<td>5524</td>
</tr>
<tr>
<td>MEMBRANE</td>
<td></td>
<td></td>
</tr>
<tr>
<td>reverse osmosis</td>
<td>929</td>
<td>1478</td>
</tr>
<tr>
<td>electrodialysis</td>
<td>82</td>
<td>135</td>
</tr>
<tr>
<td>electrodialysis-reversing</td>
<td>228</td>
<td>139</td>
</tr>
<tr>
<td>subtotal:</td>
<td>1239</td>
<td>1752</td>
</tr>
<tr>
<td>total:</td>
<td>2204</td>
<td>7276</td>
</tr>
</tbody>
</table>

**TABLE 1-1.**

Land-based desalting plants in operation or under construction with capacities of 95 m³/day or more (June 1980).
As can be seen from figure 1-1 and table 1-1, the bulk of the installed capacity in desalination is distillation. Currently Multi-Stage Flash evaporation is the dominant conventional established process and can be seen as a reference for this thesis. Figure 1-2 represents a simple flow scheme of an MSF evaporator which can be described best as a counter-current heat exchanger.

**FIGURE 1-2.**

*Flow scheme of "once through" Multi-Stage Flash evaporator.*

The feed (f.i. sea water) is heated in the condensers of the heat recovery section. After obtaining a final heat supply in the brine heater, the flashing process will start. The brine flashes through all the chambers or stages where the flashing is initiated in the orifices in the intersection walls. A drop in saturation pressures in the stage occurs resulting in a partial evaporation of the brine in each flash stage. The vapour produced flows through the water/steam separators (demisters) and condenses on the outer surface of the condensers which are cooled by the incoming feed.

**FIGURE 1-3.**

*Flow scheme of "recirculation" Multi-Stage Flash evaporator.*
The distillate is collected in the trays under the condenser surface and cascades down, in the same way and direction as the brine, to the tray in the next stage. Pumps are used for the feed supply and to remove the brine and distillate from the coldest flash stage. A vacuum system removes dissolved gases and inleaking non-condensables.

The evaporator mentioned above is the once-through type where no recirculation of the brine is applied. Most widely used is the recirculation type where the brine is partially recycled (figure 1-3). This type is preferred because of advantages in chemical treatment, removal of dissolved gases and temperature control.

Figure 1-4 illustrates the temperature diagram for an MSF with recirculation of the brine.

**FIGURE 1-4.**
Temperature-heat transfer surface diagram for a Multi-Stage Flash evaporator with recirculation of brine.
1.1.1. New developments.

In addition to the increasing demand of desalted water there are a number of influences on the development and production of desalination processes and units.

Three major influences on development of desalination processes and units are:

- the ever existing driving force - based on scientific aspects and competition - to improve systems;
- the impact of increased fuel costs.
  The increased fuel costs due to the sudden price increase of oil posted by the OPEC-Countries since 1973 forced principal investigators, technologists and planners to take a critical look at the economics of desalination processes;
- the relative change in prices of copper-alloys and mild steel.
  Relative change in prices of heat transfer material as copper-alloys and construction material such as mild steel forced such scientists and technologists to take a critical look at ideas about heat transfer enhancement, steel weight reduction and corrosion/erosion aspects.

There is a large number of different new developments in all desalination processes and equipment carried out by many research and development institutes, manufacturers, and licencees. Some developments aim at better heat transfer, and some at improving many varied aspects of this wide technology.

Present developments are:

- Vertical Tube Foam Evaporator (VTFE)
- Vapour Compression evaporator (VC)
- Horizontal Aluminium Tube Multiple Effect evaporator (HATME)
- Falling Film Multi-Stage Flash evaporator (FFMSF)
- Direct Contact Condensation (DCC)
- Multi-Effect Stack evaporator (MES) and the present work on
- Multi-Stage Flash/Fluidized Bed Evaporator (MSF/FBE)

Before explaining the subject of this thesis in detail and outlining the method of approach for the investigations the developments as summarized above will be briefly reviewed for a broader perspective of developments in desalination by thermal processes.
1.1.1.1. Vertical Tube Foam Evaporation.

In an MSF evaporator liquid in the condenser tubes is heated throughout all the stages and after final heat supply in the brine heater the brine flashes in the stages. The produced vapour is then condensed on the outer surface of the condenser tubes. The Multiple Effect distillation process operates differently and in a series of identical vessels, called effects. In each effect on one side of the heat transfer surface evaporation takes place, whereas at the other side simultaneously condensation of produced vapour occurs.

Vertical Tube Foam Evaporation operates as a Multiple Effect Evaporator. SEPHTON [1.3] developed the vertical tube foam evaporator (VTFE) as it is briefly described below. Figure 1-5 illustrates this development.

**FIGURE 1-5.**
Flow scheme of a two effect Vertical Tube Foam Evaporator.

The introduction of a selected surfactant into sea water feed, followed by imposing foamy vapour-liquid flow over a heat transfer surface, provides a new approach for desalination. Vertical tube foam evaporation (VTFE) also known as interface enhancement for evaporation of liquids has been developed beyond the pilot plant stage.
Applied to the upflow mode of Vertical Tube Evaporation (VTE), it provides a desalination plant that to a certain extent is comparable with MSF in simplicity and a heat transfer performance that is enhanced to a level substantially higher than the downflow VTE. The surfactant, added to the feed in up to 25 ppm does not transfer into the vapour phase but is concentrated in the brine and is largely recyclable by a foam fractionation process.

1.1.1.2. Low temperature Vapour Compression.

This development is illustrated in figure 1-6. Reference [1.4] amongst others refers specifically to this subject.

![Diagram](image_url)

**FIGURE 1-6.**
Flow scheme of a mechanically driven Vapour Compression unit.

Vapour compression, which is based on a mechanical heat pump cycle is an efficient distillation process. In this process the incoming sea water is pretreated with a scale inhibiting additive and passed through a heat exchanger where the heat in the discharged brine and product stream is recovered.

The sea water is mixed with recirculated brine and sprayed on the outside of a bundle of horizontal heat transfer tubes at a rate just sufficient to create thin continuous liquid films.
At the inside of the tubes condensating steam is supplied by the compressor. The latent heat of condensating is transferred to the thin liquid film on the outside of the tubes. This results in a partial evaporating of the brine into vapour. After passing through a demister to remove droplets, the vapour is compressed to increase the vapour pressure and discharged to the inside of the tubes. There it condenses, supplying the latent heat required for the evaporation process. The condensate, which is the product and the rejected brine are pumped out after exchanging heat with the feed.

Non-condensable gases are first concentrated in an auxiliary condenser and then exhausted by means of a vacuum pump. Operating temperatures of this first generation units were up to 55 °C, minimizing corrosion and scale while using polyphosphate feed pretreatment. However, additional effort was given to the development of higher temperature plants. Considerable effort has gone into selecting the best configuration of liquid film thickness, vapour velocity, condensate drainage, non-condensable gas-purging and compactness.

As a result of development carried out in the conjunction with the operation of aluminium tubes it is now feasible to operate at temperatures up to 70 °C.

1.1.1.3. Horizontal Aluminium Tubes Multi-Effect evaporator.

WEINBERG [1.5] reports on the results of the development of the Horizontal Aluminium Tube Multi-Effect Evaporator (HATME). Figure 1-7 presents a simplified flow scheme of this Multi-Effect Evaporator.

The plant operates on the principle of multi-effect distillation, whereby a certain amount of steam is used for the production of several times its own weight of distillate by means of repeated stages of evaporation and condensation which are carried out simultaneously in a series of identical vessels called effects. This basic unit of the plant consists of a cylindrical shell maintained under vacuum which is provided internally with an array of many tubes through which water vapour or steam pass and are condensed.
FIGURE 1-7.
Flow scheme of Horizontal Tube Multi-Effect evaporator.

A number of spray nozzles distribute sea water or concentrated sea water over the outer surface of the tubes to cool them. In the compartments distillate, saline water and vapour are collected separately and passed on to the next vessel in the line. This type of unit is operated at a low top temperature (70 °C) to minimize scaling of the heat exchanger surface. The use of polyphosphate additive is sufficient to inhibit the build-up of scale.

The heat transfer surface is made entirely of aluminium, which allows design large heat transfer areas, thereby reducing thermal driving forces.

1.1.1.4. Falling Film Multi-Stage Flash evaporator.

The development of the Falling Film Multi-Stage Flash evaporator (FFMSF) is performed by STANDIFORD [1.6] and in principle shown in figure 1-8. This evaporator is an outgrowth of attempts to develop a high performance ratio desalting unit for very small capacities such as might be needed for small vessels and beach resorts.

The principal evaporator processes used for large desalination plants have many drawbacks when applied to small units. The falling film multiple-effect requires brine pumps for each effect, and since possible a dozen effects might be involved, the costs of pumps would be exorbitant. Other evaporator types could be used that do not need these pumps but when built in small sizes require level controllers for each effect which are almost as expensive as small pumps. Furthermore, a multiple-effect evaporator used for sea water and for most brackish waters requires means for alkaline scale prevention.
FIGURE 1-8.
Flow scheme of a Falling Film Multi-Stage Flash evaporator.

In an FFMSF evaporator, sea water is brought into the upper water box and flows down as a film inside the tubes of the vertical heat exchanger. Vapour evolution in the liquid film is avoided by leaving the core of the tube open to atmospheric pressure and by not heating the brine above the atmospheric boiling point.

Enriching the gas phase in carbondioxide in the core of the falling film tubes provides a simple means of avoiding scale on the heated surfaces. Part of the exhaust from the vacuum pump or ejectors is fed into the brine collecting sump below the falling film tubes to avoid scaling of the heat transfer surfaces during evaporation in the flash chambers.

The flash chamber itself required a separate development because the boiling liquid must be brought to the upper part. Its design employs a series of rotating impellers on a common shaft in a series of stationary cylindrical "cups". The impeller centrifuges the flashing brine into the walls of the cup, developing enough head to carry the residual liquid up to the next cup. At the top of each cup is the vapour outlet and a kind of brine orifice. Above this brine orifice are diffusers to convert rotary liquid motion into the head needed to carry the liquid up to the next cup.
The flasher requires limited vessel volume while still exposing a large liquid surface area for flashing of the brine.

1.1.1.5. Direct Contact Condensation Multi-Stage Flash distillation.

In the design of desalination plants based upon conventional distillation methods the metal tubing heat transfer surface represents a substantial component of the plant. The number of stages or effects and the total amount of tubing are determined by striking a balance between the plant running cost, which is determined by steam economy ratio, capital amortization and running costs. The capital amortization is among others influenced by the investment in heat transfer tubing.

Potentially there is an economic advantage to a distillation method that eliminates metal tubes for heat transfer by relying on vapour condensation by direct contact with a cold liquid stream.

In the conventional MSF distillation system a hot stream of brine proceeds through a series of flash evaporation stages in channel flow. The cooling water in the heat recovery and heat rejection streams is forced through the stage tube bundles and interconnecting water boxes by a recirculation pump.

In DCC distillation, developed by KOGAN [1.7] and shown in figure 1-9, the flashing brine stream and the distilled cooling water stream must both proceed in open channel flows in order to enable vapour flashed off the brine stream to reach the cooling water stream and to condense by direct contact with it. It is difficult to maintain the two streams in countercurrent flows through a large number of consecutive stages without recourse to a multitude of large capacity low pressure difference pumps.

This difficulty has been solved by harnessing the energy liberated during the vapour flashing process to lift the brine stream through the consecutive stages. When a small fraction of a saturated liquid stream is flashed into vapour upon entry into an evaporation stage, its volume undergoes a tremendous increase.

With a correct design of the geometry of passage between consecutive stages the mechanical energy associated with the increase of volume during change of phase can be used to lift the two-phase mixture of liquid and vapour against gravity.

Sea water enters into the first stage with high temperature and rises against gravity through the consecutive stages by flash evaporation. It leaves the last stage with low temperature. Vapour liberated by flash
evaporation condenses by direct contact on the relatively cold distillate. The distillate flow runs counter-current to the brine flow through a separate section behind the brine flow shown in figure 1-9. The temperature of the distillate rises as it moves towards the first stage.
A heat exchanger unit had to be incorporated into the system in order to recover the heat from the condensing water stream. A thin wall plastic tube heat exchanger (PVF) was developed for this purpose. The heat exchanger has a considerable length and consists of four passes. A lower cost plastic heat exchanger has been developed, and extruded biaxially oriented polypropylene tubes have been used in the present pilot plant.
1.1.1.6. Multi-Effect Stack type of distillation process.

TAKADA [1.8] reports on the development of the Multi-Effect Stack (MES) type distillation process as shown in figure 1-10 and described briefly below.

**FIGURE 1-10.**
Flow scheme of a Multi-Effect Stack evaporator.

Feed sea water is pumped through a series of preheaters incorporated in each effect and the warmed sea water is sprayed into the top of the first effect. The sea water descends as a thin film on the outside of the tubes. Heating steam from a boiler condenses inside the tubes, causing the external sea water film on the tubes to boil and produce new vapour at a lower temperature in the first effect. Only a fraction of the sea water film is evaporated. The new vapour is led through mist separators; it is used partially to preheat the feed water and finally condenses inside the tubes of the second effect to form the fresh water product.
The unevaporated portion of the sea water in the first effect falls to the floor and is sprayed into the top of the second effect. Due to the lower pressure, spraying is accompanied by flashing to a cooler temperature. The sea water again flows as a thin film over the outside of the tubes in the second effect and is partially evaporated by the heat transfer from the condensing vapour inside the tubes. The process is repeated from effect to effect down the stack. The vapour generated in each effect is a combination of flashing through the nozzles and thin-film boiling at the tube surface. The vapour from the next-to-last effect is condensed on a shell-and-tube cooler in the last effect and the process heat is finally carried away to the sea by the cooling water. Part of the cooling water discharge is used as feedwater to the stack. The concentrated brine is blown down from the next-to-last effect. The fresh water product from each effect is cascaded down the plant, flashing between effects and returning useful heat to the process before finally being withdrawn from the last effect at the lowest temperature.
1.2. The Multi-Stage Flash/Fluidized Bed Evaporator.

For any development a specific relationship between the capital costs and the running costs exists, hence there is a necessity to categorize specific items of a development in the following categories:

- Investment Costs
- Energy Consumption Costs
- Operation and Maintenance Costs.

![Diagram](attachment:image.png)

**FIGURE 1-11.**

*Relationship between energy costs and capital costs.*

Figure 1-11 shows the simplified and qualitative relationship between the costs of distilled water and the energy consumption of the system. Recognizing the high energy consumption and the limitations of the well-known conventional Multi-Stage Flash Evaporator, Delft University of Technology decided in 1968 to start attempts to develop a type of evaporator which overcomes major drawbacks of the conventional type of evaporator i.e.:

- limitation of the specific energy consumption
- large volume and weight of the flash evaporator
- limited range of turn down ratio of the product output (partial load).
Started as an attempt to improve a vertical upward flashing evaporator [1.9], however, the developments as described by KLAREN [1.10] and finally in this thesis demonstrate a considerable change in purpose of the investigations. The result is a completely new type of evaporator in which the vertical design the only feature is still resembling the earlier developments.

Major general aspects of the developments described in this thesis are:

- reduction of investment cost
- possibilities for reduction of the specific energy consumption
- reliable and easy operation of the unit.

For the sake of clearness a short description of the developed unit and its features will be given below.

The developed vertical evaporator is of the Multi-Stage Flash type and consists of an evaporation and condensation column including an integrated brine heater.

![Diagram of Multi-Stage Flash/Fluidized Bed Evaporator]

**FIGURE 1-12.**

*Flow scheme of a Multi-Stage Flash/Fluidized Bed Evaporator.*
Figure 1-12 shows a general flow scheme of a Multi-Stage Flash Evaporator with a Fluidized Bed Heat Exchanger. The condenser is of the "long tube" design and consists of a number of vertical parallel condenser tubes partly filled with solid particles, fluidized by the liquid flow upward through the tubes. The solid particles in fluidized state cause a breakdown of the laminar thermal boundary layer and a promotion of the turbulence of the liquid in the tube. This results in a high heat transfer coefficient at low values of the liquid velocities in the condenser tube and permits the use of a "long tube" condenser design with an extremely short stage height. Because of the soft mechanical abrasive action of the fluidized particles exerted on the tube wall, the fouling factor of the fluidized bed heat exchanger is drastically reduced in comparison with the heat exchangers of the conventional Multi-Stage Flash evaporators. In this vertical Multi-Stage Flash evaporator the brine enters the flash stage as a film and breaks up in thin filaments and small droplets. This causes high interfacial heat transfer. High heat transfer coefficients combined with drastically reduced fouling factors and good flash-off or approach to equilibrium result in much lower specific energy consumption of the Multi-Stage Flash/Fluidized Bed Evaporator (MSF/FBE) in comparison with the conventional evaporator (MSF). Steam/water separation without the application of wire-mesh demisters is achieved through application of centrifugal effects in the vapour space. A significant increase of the product output per unit volume for this vertical Multi-Stage Flash evaporator in comparison with conventional evaporators is achieved, resulting in reduction of investment costs. The earlier developments of the vertical flash evaporator, referred to as Multi-Stage Flash/Fluidized Bed Evaporator (MSF/FBE) have been described extensively by KLAREN [1.10]. After these early developments it was felt that different measurements and further investigations were necessary for a basic design of the MSF/FBE. The necessary investigations and measurements to serve that purpose can be summarized with:

- heat transfer in a liquid-solid fluidized bed
- de-scaling and de-fouling capabilities of the fluidized bed
- flash process and entrainment in a vertical flash stage.
1.3. Method of approach for the investigations.

The method of approach for the investigations is presented in the investigation flow scheme in figure 1-13.

**FIGURE 1-13.**

Subjects of investigations and relation between the chapters of this thesis.

The investigations and measurements are divided according to the major components of the Multi Stage Flash/Fluidized Bed Evaporator: The condensation column with the fluidized bed condenser sections and the evaporation column with the respective flash stages.

For the condensation part thorough knowledge of the features and the limitations of a heat exchanger with fluidized bed is necessary. The investigations needed for this knowledge and information origin in two main sources, the heat transfer properties in the liquid-solid fluidized bed and the de-scaling and de-fouling capabilities of the fluidized bed.
For a proper understanding and design of the evaporation part knowledge about the flash process, the limits of approach to equilibrium and detailed information about the entrainment aspects of a vertical evaporator stage in a minimum stage volume are necessary.

For an optimum design of an MSF/FBE it is required to study the behaviour and all related information obtained from the pilot plant. Figure 1-14 shows a simplified flow scheme of an MSF/FBE with an illustrative impression of the scope and location of the abovementioned aspects of investigation.

FIGURE 1-14.
Simplified scheme of a Multi-Stage Flash/Fluidized Bed Evaporator showing to what part of the evaporator the respective content of the chapters of this thesis have been dedicated to.
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Figure 2-0 presents a visual aid for the contents of this chapter.
2.1. Introduction.

A vertical Multi-Stage Flash/Fluidized Bed Evaporator (MSF/FBE) has a fluidized bed heat exchanger in the heat recovery section and the brine heater as can be seen in figure 2-1.

This fluidized bed heat exchanger is of the liquid-solid type. It consists of a large number of vertical parallel condenser tubes partly filled with spherical glass particles. The particles are brought into a homogeneous fluidization state by the liquid flowing upward through the tubes.

**FIGURE 2-1.**
Simplified flow scheme of a Multi-Stage Flash/Fluidized Bed Evaporator (MSF/FBE).
Because of the heat transfer and liquid velocity features as analyzed in this and the following chapters, long tubes condenser configuration is chosen. Long tube configuration means that the upflowing liquid through the condenser and the flashing brine in the stages are both vertical oriented but in opposite directions.

For a well defined design of a complete evaporator it is important to know the heat transfer relations for a fluidized bed heat exchanger as well as the limits thereof.

2.1.1. Heat transfer in an MSF/FBE.

In a Multi-Stage Flash evaporator the evaporation takes place in the flash chamber and the condensation in the heat exchanger. The brine is heated under pressure while flowing upward up to and including the brine heater. In the brine heater the external heat is supplied to the liquid in order to create positive temperature difference for the evaporation. After obtaining the final energy supply in the brine heater the brine is discharged through all stages each at a reduced saturation pressure in comparison with the previous stage. Evaporation takes place from the brine in each of the stages in an attempt to obtain equilibrium with the vapour condition prevailing in each individual stage. The evaporation process is elaborated in Chapter 4.

The feedwater is for a large part heated by recovering the latent heat of vaporization. The heat transfer takes place from the vapour space through the attendant condensate film on the outside of the condenser tube, via the tube wall and other resistances, to the feedwater in the heat exchanger tubes and as illustrated in figure 2-2.

![FIGURE 2-2. Qualitative temperature profile for heat transfer in a (fluidized bed) heat exchanger tube.](image-url)
2.1.2. Experimental approach.

The lack of suitable and reliable information on heat transfer from wall to liquid in a liquid-solid fluidized bed system requires experimental work. This chapter describes the test module built and experimental work performed in order to provide some of the necessary data for a further design of fluidized bed heat exchangers as essential part of the Multi-Stage Flash/Fluidized Bed Evaporator (MSF/FBE) under development.

A description of the test module is given in subchapter 2.2. The aim of the experimental work is to obtain information on local wall-to-liquid heat transfer coefficients. The measurements are carried out based on two different measurement principles:

- **method A**: (subchapters 2.3 and 2.4) using a thermocouple in the tube wall and a thermocouple probe in the bulk of the liquid.
  
  In the first part of the experimental work a measuring principle has been applied using a relatively thin heat exchanger tube wall. The heat flux to the tube is determined using the Nusselt correlation for condensing steam on the outside of a tube.
  
  To establish the wall temperature a thin thermocouple has been imbedded in a circumferential groove at the outside of the heat exchanger tube.
  
  The results of the measurements to obtain values for the local wall-to-liquid heat transfer coefficients were not satisfactory.

- **method B**: (subchapters 2.5 and 2.6) using two thermocouples in the tube wall.
  
  For the second part of the experimental work the test module has been modified to such an extent that the heat flux to the tube can be measured locally by applying two imbedded thermocouples at different distance from the wall in the same cross section of the tube.
  
  During both parts of the experimental work averaged heat transfer coefficients are determined using only one thermocouple in the tube wall and the thermocouple in the bulk of the liquid. Average heat transfer coefficients are measured based on various different lengths of the tube.

After description of measurement principles and discussion of the results obtained with the measurements, subchapters 2.7 and 2.8 deal with determination of errors whereas subchapter 2.9 is devoted to specific analysis of measuring of wall temperatures with imbedded thermocouples.
2.2. Test module for heat transfer measurements.

The test module for experimental determination of the wall to liquid heat transfer coefficient of liquid-solid fluidized bed is schematically shown in figure 2-3.

FIGURE 2-3.
Flow scheme of test module for heat transfer experiments.
The test module can be subdivided into three main circuits:

- **heater circuit**: in this circuit water is heated by means of electrical resistance elements and finally will produce saturated steam of desired pressure and temperature;

- **heat exchanger**: this circuit contains a single tube fluidized bed in which the circulating liquid is heated by the condensing steam from the heater circuit;

- **cooler circuit**: in this circuit and by means of a flexible system of a two stage cooling system the liquid temperature at the entrance of the fluidized bed will be lowered to the desired level.

The heater circuit and the heat exchanger circuit are of the closed loop type with only a make up system for leakage and losses from the respective circuits. Both systems are operated with demineralized and degasified distilled water. All circuits are made of coated steel, stainless steels and copper-nickel alloys in order to avoid disturbances in operation due to corrosion or corrosion products.

2.2.1. The heater circuit.

The heater circuit mainly consists of an electrical heater, a flash tank and a recycle pump. The water of the heater circuit is heated to an elevated temperature sufficient to start a flash type of boiling when it enters the flash tank. Supply of energy is fully adjustable from 0 - 100 kW. Fluctuations in the outlet temperature due to slight variation of the grid voltage are eliminated by a fast response thyristor controlled heater element. Water temperature at the outlet of the heater system should be kept within narrow limits as the measuring procedure is to be performed under stationary conditions. In the flash tank part of the circulated water will evaporate during the flash-off and after passage of the scrubber and a wire mesh demister the almost saturated steam enters the condensation chambers of the heat exchanger circuit.
FIGURE 2-4.
Composition drawing of the heat exchanger.
During operation some leakage of water will occur and some of the condensed steam will not return to the heater circuit. In order to keep the water content of the system constant a continuous make up is installed. The amount of make up water is controlled by the liquid level in the flash tank.

2.2.2. The heat exchanger circuit.

Heating of the continuous medium of the liquid-solid fluidized bed takes place in the heat exchanger while heat rejection is performed in the cooler section of the heat exchanger circuit. The water is pumped through the various parts of the circuit by a recycle pump and because of the influence on the porosity the flow is controlled by a parabolic type valve in combination with a high accuracy by-pass of low capacity over the recycle pump.

The latent heat of the condensing steam from the heater circuit heats up the liquid in the fluidized bed heat exchanger tube. This heat exchanger is the essential part of the apparatus where the heat transfer measurements are performed.

In the heat exchanger tube particles are fluidized by the upflowing liquid. The experiments covered a range of glass particles with diameters ranging from 1 to 4 mm. The volume flow is controlled by the described set of valves where the total fluidization length is inspected by means of a pyrex glass sight section at the top of the fluidized bed heat exchanger tube.

Figure 2-4 presents a composition drawing of the entire heat exchanger section with clarification of specific components.

The heat exchanger tube is surrounded by two concentric vapour spaces both in connection with the steam supply line. The inner vapour space serves the condensing of the steam required for the temperature elevation of the liquid through the fluidized bed. The outer vapour space creates an insulation of the inner space while the steam condensed in this annular space is separately discharged from the system.

The principle of the two concentric vapour spaces allows for accurate heat balance calculation as well as for accurate calculation of the Reynolds number of the condensate film along the heat exchanger tube.
Condensate is discharged from the respective vapour spaces and collected in hold up tanks enabling to operate without a condensate level in the heat exchanger. Continuous venting of non-condensable gases is performed by a vacuum pump, where the steam carried with the gases is condensed separately. The equipped instruments of the heat exchanger section is described in the respective subchapter.

An adjustable probe is furnished on top of the heat exchanger tube and can travel the entire length of the inner side of the tube in order to measure bulk liquid temperatures at the respective locations of tube wall mounted thermocouples.

2.2.3. The cooler circuit.

The water of the fluidized bed heat exchanger circuit is cooled in the heat reject section as shown in the flow scheme as presented in figure 2-3. The cooler of the reject section is of a two stage type where in an intermediate hold up and mix tank, desired temperature levels of the coolant can be achieved.

The intermediate cooling system was installed to enable experiments over a wide range of temperatures with sufficient large coolant flows for proper control purposes.

Detailed description of the above mentioned circuits, components and equipment can be found in references [2.2], [2.3], [2.4], [2.5] and [2.6].

2.2.4. Instrumentation of the test module.

The test module contains a large number of indicating and recording measuring instruments. Figures 2-3 and 2-5 indicate the major measuring points. Not shown are the level control loops for the flash tank and the condensate sample tanks.

2.2.4.1. Volume flows.

Flow meters of the turbine type are used for the measurement of the volume flow through the heat exchanger tube and the volume flow in the condensate return line from the inner condensation chamber around the heat exchanger to the heater circuit. The volume flow of the recirculating heating liquid in the heater circuit is measured by means of a magnetic flow transducer mounted in front of the electrical heater.
The volume flow in the heat reject section is controlled by a self acting constant flow valve. The actual volume flow is governed by a simple control loop activated from the main control panel.

2.2.4.2. Pressure indicators.

In the main circuits four high precision manometers of the Bourdon tube type are installed. Pressure readings are obtained from:

- the heat exchanger tube circuit, after the fluidized bed;
- the heater circuit in the main line to the flash tank;
- the steam line entering the condensation chamber of the heat exchanger circuit.
2.2.4.3. **Temperatures.**

For measurement of temperatures 30 temperature sensors are installed in the test module.
Seven resistance thermometers of the PT 100 (Platinum-Rhodium type) are installed for high accuracy measurements of various temperatures.
As can be seen from figure 2-5 resistance thermometers are applied at the following locations:

- steam to condensation chamber,
- inner condensation chamber (2x),
- entrance and outlet of liquid flow through the fluidized bed,
- condensate.

All other temperatures are measured with thermocouples (Chromel/Alumel) including the temperature measurements in the heat exchanger wall (12x) and the bulk of liquid in the fluidized bed. The latter is measured by a longitudinal movable probe which can be locked in pre-set positions corresponding with the set of thermocouples in a radial cross section of the heat exchanger tube wall.
The cold junctions of each of the mentioned thermocouples are placed in a triple point cell and kept continuously at 0 °C ± 0.01 °C.
The thermocouples installed in the heat exchanger tube wall have an outer diameter of 0.5 mm, all other thermocouples have an outer diameter of 1.5 mm.

2.2.5. **Data acquisition, storage and computation.**

The test module as described in this subchapter has a number of measuring instruments, which data are frequently to be monitored and interpreted for each of the experiments. Figure 2-5 indicates the measuring points incorporated in the data acquisition system.

Taken into consideration the number of different set points to cover the required span in porosities, particle diameters and temperature levels it was decided to develop a set of computer programs and routines compatible with the data acquisition system of the test module.
Figure 2-6 gives a simplified outline of computation procedures.
FIGURE 2-6.
Flow diagram of data acquisition and computation steps in the calculation of the heat transfer data.

Early results were obtained by means of off-line computations after the measuring data had been produced and stored on paper tape. Later on during the experiments the system was converted into a mode of operation were a maximum of in-line computations were performed by means of concentration of a process monitor (Doric digitrend 240 data acquisition system) and a table top mini computer (IBM series 5100). The latter introduction of the computational aids found its necessity in the need to perform in-line stability analyses of the experiments and to follow the state of the art in rational experimental and computational procedures.

Final computations of the results are performed with a developed computer program implemented in the basic system of the university's main computer (IBM 370/158 at the Mathematical Centre).
2.2.6. Calibration.

Before major test runs all instrumentation has been calibrated. Each of the volume flow meters are calibrated including the transducer, pulse converter and adaptor to the data acquisition system and including the complete wiring of the system. Calibration is performed on a constant volume flow-time basis with a defined total volume of liquid through the system.

Pressure indicators are of the precision type and are re-adjusted if necessary on a regular basis. Calibration is performed with high accuracy pressure indicators calibrated against master standards of the Netherlands Institute for Standards.

All temperature sensors, thermocouples and resistance thermometers are calibrated in a precision calibration bath (± 0.01 °C) and against a set of certified high resolutions, high accuracy thermometers of the Landsberger type (± 0.01 °C).

The thermocouples imbedded in the heat exchanger tube wall are also calibrated with the above described method and also inclusive all wiring connections and cold junctions. The results of the calibrations are, as coefficients of multi grade polynomials, introduced in the computer programs for the calculations necessary for the processing of the data. Extensive descriptions of calibration procedures can be found in the literature references as mentioned in subchapter 2.2.3.
2.3. Background and principles of the measuring method A.

2.3.1. The local heat transfer coefficient.

Assuming a heat transfer tube with a liquid-solid fluidized bed system at the inside and a warmer medium at the outside of the tube, the wall-to-liquid heat transfer coefficient ($\alpha_{w,1}$) for a fluidized bed system is defined as follows:

$$\dot{\phi} = \alpha_{w,1} A \left( T_{w,1} - T_{b,1} \right)$$  (2-1)

with:

$\dot{\phi}$ = heat flux to the liquid

$\alpha_{w,1}$ = wall-to-liquid heat transfer coefficient (average)

$A$ = area of heat transfer surface

$T_{w,1}$ = wall temperature liquid side

$T_{b,1}$ = liquid temperature in the bulk

It is obvious that directly measuring of $\alpha_{w,1}$ is not possible: $\dot{\phi}$, $A$, $(T_{w,1} - T_{b,1})$ must be known.

To provide a heat flux to the liquid, a system of condensation of saturated steam at the outside of a vertical tube (film condensation) has been chosen. The heat flux can be computed by means of the analytical correlation of NUSSELT [2.7] for such a system. Other systems of heating are found to be too expensive or in practice not executable. This means that we have to deal with a vertical heat transfer tube, with at the outside film condensation of saturated steam and at the inside a fluidized bed system with an upward flow direction.

As it is more interesting to know the "local" heat transfer coefficient $\alpha_{w,1,z}$ than an average $\alpha_{w,1}$, it is necessary to measure the bulk temperature and the temperature somewhere in the tube wall at a certain location $z$, as shown in figure 2-7.

The specific problems dealing with measuring the temperatures in the tube wall will be discussed in subchapter 2.9.

The following heat balances can now be formulated:

$$\dot{\phi}_{z} = \alpha_{c,z} \pi D \left( T_{s} - T_{w,c,z} \right)$$  (2-2)
FIGURE 2-7.

Location of the temperature measuring points applying the measuring method A.

\[ \phi_z = \frac{\lambda_{CN} \frac{2}{\pi}}{\ln \left( \frac{D_0/2}{D_0/2-x_f} \right)} \left( T_{w,c,z} - T_{x_f,z} \right) \] (2-3)

\[ \phi_z = \frac{\lambda_{CN} 2\pi}{\ln \left( \frac{D_0}{D_i} \right)} \left( T_{w,c,z} - T_{w,1,z} \right) \] (2-4)

\[ \phi_z = \alpha_{w,1,z} \pi D_i \left( T_{w,1,z} - T_{b,1,z} \right) \] (2-5)
with:

\( \phi_z \) = local heat flux to the liquid at location \( z \)

\( \alpha_{c,z} \) = condensation heat transfer coefficient at location \( z \)

\( D_o \) = outer diameter of the heat exchanger tube

\( T_s \) = saturation temperature at the surface of the condensate

\( T_{w,c,z} \) = wall temperature condensation side at location \( z \)

\( \lambda_{CN} \) = thermal conductivity of copper nickel material

\( x_f \) = fictive distance for temperature measurements

\( T_{f,z} \) = temperature of the centre of the thermocouple at location \( z \)

\( T_{b,l,z} \) = liquid temperature in the bulk at location \( z \)

\( D_i \) = inner diameter of the heat exchanger tube

\( \alpha_{w,l,z} \) = wall-to-liquid heat transfer coefficient at location \( z \)

\( T_{w,l,z} \) = wall temperature, liquid side at location \( z \)

from which \( \alpha_{w,l,z} \) is solvable.

Fouling factors of the outer and inner tube wall have been neglected as by earlier experiments of KLAREN [2.1] and VEENMAN [2.8] under almost the same conditions has been proven to be an acceptable assumption. After test runs (total approx. 750 hours) with the test module, surfaces of the heat transfer tube were found to be clean.

2.3.2. The Nusselt correlation.

NUSSELT [2.7] determined in 1916 an analytical correlation for laminar film condensation on vertical surfaces. This correlation reads for the local value of the heat transfer coefficient at a distance \( z \) from the start of condensation:

\[
\alpha_{c,z} = \left[ \frac{\lambda^3_c \rho_c^2 g_n h_{lg}}{4z \Delta T_c \eta_c} \right]^{0.25}
\]  

(2-6)

with:

\( \lambda_c \) = thermal conductivity of the condensate

\( \rho_c \) = density of the condensate

\( g_n \) = acceleration of gravity

\( h_{lg} \) = latent heat of evaporation

\( \Delta T_c \) = mean temperature difference over the condensate film

\( \eta_c \) = mean dynamic viscosity of the condensate

\( z \) = undisturbed length of the condensate film
When the thickness of the condensate film is much smaller than the radius of a tube, the correlation is also valid for laminar film condensation on a vertical tube. This is in most cases when the radius is greater than 2 mm.

The equations (2-2), (2-4) and (2-5) can now be solved in an iterative way. The VDI-Wärme-Atlas [2.9] gives the following conditions for the validity of correlation (2-6):

- saturated steam
- no non-condensable gases
- no shear stress at the interface vapour film, steam velocity \(< 5 \text{ m/s}
- \) laminar flow condensate film, Reynolds number \(< 350
- \) linear temperature profile
- negligible heat capacity in the condensate
- no inertia effects
- no interfacial resistance.

To fulfill the first four conditions the necessary requirements are being made in the test module. For low pressure steam it is common practice to neglect the last four conditions.

Surface tension, however, causes the flow to become unsteady by inducing a wavy motion which invalidates the model of steady laminar flow. And it is found by EMERSON [2.10] that the model is truly applicable only when the Reynolds number is less than about 30. As the error in \( \alpha_{w,1,z} \) increases with low values of the Reynolds number it is necessary to know the error made by using correlation (2-6) with Reynolds number greater than 30.

2.3.3. The calculation of the local heat transfer coefficient.

The calculation procedure as described in this subchapter is based on the use of Nusselt correlation for laminar film condensation.

Equations (2-2) and (2-3) lead to:

\[
\alpha_{c,z} = \frac{\lambda_{CN} \pi D_o}{D_o / 2} \left( \frac{T_s - T_{w,c,z}}{\ln \left( \frac{D_o / 2}{D_o / 2 - x_f} \right)} \right)
\]

The temperature difference over the condensate film must be solved before the local heat transfer coefficient \( \alpha_{c,z} \) can be found.
An equation to solve the temperature difference over the condensate film is found by combining the equations (2-6) and (2-7) where the resulting equation can be written as:

\[ AX^{0.75} + BX = C \]  
(2-8)

with:

\[ X = \left( T_S - T_{n, c, z} \right) = \Delta T_c \]  
(2-9)

where \( X \) is the temperature difference over the condensate film and:

\[ A = \left( \frac{\lambda_c}{c} \frac{D_c^2}{2g_n h_{lg}} \right)^{0.25} \frac{D_o}{2} \]  
(2-10)

\[ B = \frac{\lambda_{CN}}{\ln \left( \frac{D_o/2}{D_o/2-x_f} \right)} \]  
(2-11)

\[ C = B \left( T_S - T_{x_f, z} \right) \]  
(2-12)

Iterative calculation leads to a solution of equation (2-8).

With a known value for the temperature difference over the condensate film, the local value of the condensation heat transfer coefficient can be calculated with equation (2-6).

With the local value for the condensation heat transfer coefficient the overall heat transfer coefficient related to the outside of the tube is expressed:

\[ h_o = \frac{1}{\frac{1}{\alpha_{c, z}} + \frac{D_o}{2\lambda_{CN}} \ln \left( \frac{D_o/2}{D_o/2-x_f} \right)} \]  
(2-13)

The local heat flux to the tube can now be calculated and reads:

\[ \phi_z = h_o \pi D_o \left( T_S - T_{x_f, z} \right) \]  
(2-14)
Since the local heat flux to the tube has been found, the overall heat transfer coefficient related to the inside of the tube can now be calculated with a combination of the equations (2-3), (2-4) and (2-5) as follows:

\[ \dot{\phi}_z = h_i \pi D_i \left( T_{x_f,z} - T_{b,1,z} \right) \]  

(2-15)

Now the overall heat transfer coefficient can be expressed as:

\[ h_i = \frac{1}{\frac{1}{\alpha_{w,1,z}} + \frac{D_i}{2 \lambda_{CN}} \ln \left( \frac{D_i/2 - x_f}{D_i/2} \right)} \]

(2-16)

this finally leads to:

\[ \alpha_{w,1,z} = \frac{1}{\frac{1}{h_i} - \frac{D_i}{2 \lambda_{CN}} \ln \left( \frac{D_i/2 - x_f}{D_i/2} \right)} \]

(2-17)

2.3.4. The average heat transfer calculation over the entire tube length.

Calculation over the heat transfer coefficient in this subchapter is based on the average heat flux resulting from the total heat flux to the liquid in the heat exchanger tube.

The total heat flux to the liquid in the heat exchanger tube is expressed by:

\[ \dot{\phi}_{w,tot} = \dot{\phi}_m C_{p_l} \left( T_{l,out} - T_{l,in} \right) \]

(2-18)

with:

- \( \dot{\phi}_{w,tot} \) = total heat flux to the liquid in the heat exchanger tube
- \( \dot{\phi}_m \) = mass flow through the heat exchanger tube
- \( C_{p_l} \) = specific heat of the liquid
- \( T_{l,out} \) = liquid temperature at the exit of the heat exchanger tube
- \( T_{l,in} \) = liquid temperature at the entrance of the heat exchanger tube
with an effective heat exchanger tube length of \( l_{\text{tot}} \):

\[ \phi_{w,l} = \frac{\phi_{\text{tot}}}{l_{\text{tot}}} \]  

(2-19)

with:

\[ \phi_{w,l} = \text{heat flux to the liquid per unit of length (average)}. \]

Assuming a uniform distribution over the length of the heat exchanger tube:

\[ \phi_z = \phi_{w,l} \]  

(2-20)

The average heat transfer coefficient can be calculated.

Since the calculated heat transfer coefficient is based on an averaged heat flux over the length of the tube it is referred to as the "average" heat transfer coefficient.

The calculation of the average heat transfer coefficient is based on the equations (2-15), (2-16) and (2-17) in subchapter 2.3.3 and leads to:

\[ \bar{\alpha}_{w,l} = \frac{1}{h_i - \frac{D_i}{2\lambda_{\text{CN}}} \ln \left( \frac{D_o/2-x_l}{D_i/2} \right)} \]  

(2-21)
2.4. Results obtained with measuring method A.

The principle of the measuring method A has been elaborated in subchapter 2.3.

The geometry of the heat exchanger tube, the selected material, the thermocouple description and the range of independent variables covered in the experiments are presented in table 2-1.

The stationary conditions during the actual experiments and particular during the measurements of the data necessary for the computations of the heat transfer coefficients were monitored by continuous checking on heat and mass balances as well as by accurately monitoring temperatures.

The in-line data acquisition and computations by means of the installed mini computer performed the majority of these controlling functions.

<table>
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<td>external diameter of heat exchanger tube</td>
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<td>chromel-alumel</td>
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<tr>
<td>shield material</td>
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</tr>
<tr>
<td>shield diameter</td>
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<tr>
<td>computed &quot;fictive distance&quot;</td>
<td>$x_f = 0.325 \pm 0.075$ mm</td>
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</tbody>
</table>

<table>
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<th>covered range of independent variables</th>
<th></th>
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</thead>
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<td>0.003 and 0.005 m</td>
</tr>
<tr>
<td>particle density</td>
<td>2640 respectively 2625 kg/m$^3$</td>
</tr>
<tr>
<td>fluidized bed porosity</td>
<td>0.90 to 0.70</td>
</tr>
<tr>
<td>temperature range</td>
<td>30 to 120 $^\circ$C</td>
</tr>
<tr>
<td>temperature increase</td>
<td>17 to 23 $^\circ$C</td>
</tr>
<tr>
<td>(over total tube length)</td>
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</table>

TABEL 2-1.

Basic data of the test module and the covered range of experiments (method A).
2.4.1. Local heat transfer coefficients.

The measuring method describes the experimental determination of a local wall-to-liquid heat transfer coefficient with a limited temperature increase of the liquid through the fluidized bed. This temperature increase over the entire tube length has been limited to a maximum of 20 °C.

The limitation is governed by the expected optimum temperature increase per unit of tube length for a full size MSF/FBE with a large number of evaporation stages.

Computation of the measured data and a subsequent analysis of the results showed large errors in the wall-to-liquid heat transfer coefficient. These errors in the wall-to-liquid heat transfer coefficients were regarded as too high for the purpose of the heat transfer calculations for liquid-solid fluidized bed systems.

It was concluded that the followed principle of measuring did not lead to satisfactory results for the local wall-to-liquid heat transfer values. The main reasons for failure of this type of measurements can be found in the inaccurate values for the heat transfer coefficients at the condensing side of the heat exchanger tube and at the non-linear heat flux distribution over the tube length. As it can be analysed the application of the analytical correlation for laminar film condensation on vertical planes as developed by NUSSELT [2.7] yield incorrect values for the condensation heat transfer coefficient.

This may be introduced by the negligence of surface tension of the condensate film flowing down the tube surface. The prevailing conditions and the surface tension possibly leads to instability of the condensate film resulting in wave formation at the surface. Literature survey revealed that investigations performed by EMERSON [2.10] lead to the conclusion that the analytical correlation of Nusselt is accurate when the Reynolds number of the condensate film does not exceed the value of 8.

The above led to the conclusion that, in order to establish more accurate values for local wall-to-liquid heat transfer coefficients, the values for the heat flux through the condenser tube wall had to be established in a different way eliminating the necessity for the analytical correlation of Nusselt. The developed method is described in subchapter 2.5.
2.4.2. Average heat transfer coefficients.

For the average wall-to-liquid heat transfer measurements the heat flux is determined as an average value over the tube length. The temperature increase in the fluidized bed is limited to approx. 20 °C.

Measurements are carried out over the entire temperature range of interest for the specific application being the subject of this thesis and cover liquid temperatures from 30 °C up to 120 °C. Particle size has been varied from 3 mm to 5 mm whereas specific density of the particles ranges from 2640 kg/m³ to 2625 kg/m³.

Table 2-1 gives an outline of the geometrical conditions of the test module, the particles and indicates the covered range of independent variables.

Figures 2-8 through 2-11 reveal the results of the above mentioned experiments in terms of calculated average heat transfer coefficients. The porosity of a liquid-solid fluidized bed is defined as:

\[
\varepsilon = 1 - \frac{\text{volume of the particles}}{\text{volume of the empty tube over the fluidization height}}
\]

![Graph](image)

**FIGURE 2-8.**

Heat transfer coefficients as a function of liquid temperature for \(d_p = 0.003 \text{ m}\) and for different porosities.
FIGURE 2-9.
Heat transfer coefficients as a function of porosity for \( \varphi_p = 0.003 \text{ m} \) and for different temperatures.

FIGURE 2-10.
Heat transfer coefficients as a function of liquid temperature for \( \varphi_p = 0.005 \text{ m} \) and for different porosities.

FIGURE 2-11.
Heat transfer coefficients as a function of porosity for \( \varphi_p = 0.005 \text{ m} \) and for different temperatures.
2.5. Measuring method B.

The dependence on the Nusselt correlation for the local heat flux in the measuring method A did not lead to satisfactory results. For this reason an alternative measuring method B was used as described in this subchapter.

2.5.1. Principle of the measuring method B.

The measuring method B as described further in this subchapter applies a measuring system where two thermocouples are placed in radial cross section of the tube. For this reason the heat exchanger tube has a relative thick wall to enable two measuring thermocouples to be implemented in the radial cross section. Calculations as further described in subchapter 2.10 revealed that the axial heat flux appeared to be small in comparison with the radial heat flux, justifying the measuring method with relative thick tube walls.

The measuring principle is shown in figure 2-12.

**FIGURE 2-12.**

Location of the temperature measuring points applying the measuring method B.
The pair of thermocouples are inserted in radial machined grooves in the outside and on the inside of the tube. Application of each of the thermocouples pairs have been performed with high precision and great care, allowing the assumption of equal distances to the outside wall and inside wall respectively \((x_f)\).

Destructive testing after completion of the test validated the above assumption and proved that the physical distance to the wall was well within the applied tolerances.

2.5.2. The local heat transfer calculation.

The local heat flux can be calculated with the equation:

\[
\dot{\phi}_{w,z} = \frac{\lambda_w 2\pi}{\ln \left( \frac{D_o/2-\alpha_f}{D_i/2+\alpha_f} \right)} \left( T_{o,x_f,z} - T_{i,x_f,z} \right) \tag{2-22}
\]

with:

\[
\dot{\phi}_{w,z} \quad \text{local heat flux to the liquid at location } z
\]

\[
\lambda_w \quad \text{thermal conductivity of the heat exchanger tube material}
\]

\[
T_{o,x_f,z} \quad \text{temperature of the centre of thermocouple at the outer side of the tube wall}
\]

\[
T_{i,x_f,z} \quad \text{temperature of the centre of thermocouple at the inner side of the tube wall}
\]

The overall heat transfer coefficient can now be formulated as:

\[
h_i = \frac{\dot{\phi}_{w,z}}{\pi D_i \left( T_{i,x_f,z} - T_{b,1,z} \right)} \tag{2-23}
\]

This leads with:

\[
\alpha_{w,1} = \frac{1}{h_i} \cdot \frac{D_i}{2\lambda_w \ln \left( \frac{D_i/2+\alpha_f}{D_i/2} \right)} \tag{2-24}
\]
2.5.3. The heat transfer calculation average over a part of the tube length.

Values for average wall-to-liquid heat transfer coefficients over a limited temperature range are calculated by the measured values at two locations of the axial movable thermocouple in the centre of the heat exchanger tube \( T_{b, l, z_1} \) and \( T_{b, l, z_2} \) and the imbedded thermocouples on the inside of the tube wall \( T_{i, x_f, z_1} \) and \( T_{i, x_f, z_2} \). These measurements are also performed under stationary conditions.

The values for the heat transfer coefficient averaged between the location \( z_1 \) and \( z_2 \) are calculated as follows:

\[
\overline{\alpha}_{w, l} = \frac{1}{\frac{D_i \ln \left( \frac{D_i/2+x_f}{D_i/2} \right)}{h_i} - \frac{1}{2\lambda_w} \ln \left( \frac{D_i/2+x_f}{D_i/2} \right)}
\]  
\[\text{with:}\]

\[\overline{h_i} = \frac{\overline{\tau}}{\overline{\phi_w}} \frac{D_i}{\pi (z_1 - z_2) \overline{\Delta T}} \]  

The average temperature difference \( \overline{\Delta T} \) between the temperature measuring point in the tube wall and the bulk of the liquid is defined as:

\[
\overline{\Delta T} = \frac{(T_{i, x_f, z_1} - T_{b, l, z_1}) + (T_{i, x_f, z_2} - T_{b, l, z_2})}{2}
\]

whereas the average heat flux is expressed with:

\[
\overline{\phi_w} = \overline{\phi_m} \overline{c_p} (T_{b, l, z_1} - T_{b, l, z_2})
\]
with:

\[ C_P = \text{mean specific heat of the liquid} \]

\[ T_{b,1,z_1} = \text{liquid temperature in the bulk of the liquid at location } z_1 \]

\[ T_{b,1,z_2} = \text{liquid temperature in the bulk of the liquid at location } z_2 \]

The average mass flow through the heat exchanger tube:

\[ \tau = \frac{\pi D_i^2}{8} \left( \rho_{z_1} u_{l,s,z_1} + \rho_{z_2} u_{l,s,z_2} \right) \quad (2-30) \]

with:

\[ \rho_{z_1} = \text{density of the liquid at location } z_1 \]

\[ u_{l,s,z_1} = \text{superficial liquid velocity at location } z_1 \]

\[ \rho_{z_2} = \text{density of the liquid at location } z_2 \]

\[ u_{l,s,z_2} = \text{superficial liquid velocity at location } z_2 \]

and:

\[ \bar{C}_P = \frac{C_{P,z_1} + C_{P,z_2}}{2} \quad (2-31) \]
2.6. Results obtained with measuring method B.

It was decided to construct the heat exchanger tube in 8 separate segments to be able to calibrate the wall thermocouples with a high accuracy. The calibration was carried out in a constant temperature bath, thermocouples completely wired and aged and with calibrated precision thermometers.

The geometry of the heat exchanger tube, the selected material, the thermocouple description and the range of independent variables covered in the experiments are given in table 2-2.

The stationary conditions during the actual experiments and particular during the measurements of the data necessary for the computations of the heat transfer coefficients were monitored by continuous checking on heat and mass balances as well as by accurately monitoring of temperatures. The in-line data acquisition and computations by means of the installed mini computer performed the majority of these controlling functions.

<table>
<thead>
<tr>
<th>Geometry and selected material</th>
</tr>
</thead>
<tbody>
<tr>
<td>length of heat exchanger tube</td>
</tr>
<tr>
<td>internal diameter of heat exchanger tube</td>
</tr>
<tr>
<td>external diameter of heat exchanger tube</td>
</tr>
<tr>
<td>material of heat exchanger tube</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Thermocouples</th>
</tr>
</thead>
<tbody>
<tr>
<td>type</td>
</tr>
<tr>
<td>conductor material</td>
</tr>
<tr>
<td>shield material</td>
</tr>
<tr>
<td>shield diameter</td>
</tr>
<tr>
<td>computed &quot;fictive distance&quot;</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Covered range of independent variables</th>
</tr>
</thead>
<tbody>
<tr>
<td>particle diameter</td>
</tr>
<tr>
<td>particle density</td>
</tr>
<tr>
<td>fluidized bed porosity</td>
</tr>
<tr>
<td>temperature range</td>
</tr>
<tr>
<td>temperature increase</td>
</tr>
</tbody>
</table>

(over total tube length)

TABLE 2-2.
Basic data of the test module and the covered range of experiments (method B).
2.6.1. Local heat transfer coefficients.

In the series of measurements and calculations performed with the described method B, heat transfer coefficients at the location $z$ of a thermocouple pair are calculated. The main purpose of the investigation is to measure the local wall-to-liquid heat transfer coefficient as a function of the liquid bulk temperature and consequently the majority of the obtained results are concentrated on the "local" heat transfer coefficients. The calculated heat transfer coefficients are presented in the figures 2-13, 2-14 and 2-15.

2.6.2. Average heat transfer coefficients.

The calculation of "average" heat transfer coefficients is based on averaging the heat flux over a certain length of the heat exchanger tube. In these measurements the heat flux to the liquid in the fluidized bed is averaged on the basis of temperature readings of the thermocouple in the bulk of the liquid at different positions $z$ and the imbedded thermocouples on the inside of the tube wall.

As already mentioned the maximum temperature increase over the entire length of the heat exchanger tube has been limited to approx. 20 °C whereas the averaging mechanism as described above has been applied over temperature ranges of approx. 3 °C, 7 °C and 9 °C. The calculated average heat transfer coefficients are presented in the figures 2-16 and 2-17.

The results of the above measurements and calculations served the purpose of verifying the results as found with the measurements mentioned in subchapter 2.3.4.
FIGURE 2-13.
Heat transfer coefficients as a function of liquid temperature for $d_p = 0.001 \text{ m}$ and for different porosities.
FIGURE 2-14.
Heat transfer coefficients as a function of liquid temperature for $d_p = 0.002 \text{ m}$ and for different porosities.
FIGURE 2-15.

Heat transfer coefficients as a function of liquid temperature for \( d_p = 0.003 \text{ m} \) and for different porosities.
FIGURE 2-16.
Average heat transfer coefficients as a function of liquid temperature for $d_p = 0.002$ m and for different porosities.
FIGURE 2-17.
Average heat transfer coefficients as a function of liquid temperature for $d_p = 0.003 \text{ m}$ and for different porosities.
2.7. Calculation of error in the local heat transfer coefficient.

The calculation of the error in the measurements performed will be discussed in this chapter and is also based on the work performed by SCHÖN [2.2], DE GROOT [2.5], BRINK [2.6], SMIT [2.11] and POTZE [2.12].

Since the heat transfer coefficients will always be expressed as a function of the porosity of the fluidized bed system, the next subchapter is dedicated to the establishment of the error in the porosity of the fluidized bed.

The maximum error will be determined in dependence of the error in the readings of the instruments and possible manufacturing imperfections. From the manufacturing of the segments of the heat exchanger tube and the machining because of application of groove type of imbedded thermocouple the following measured tolerances are applicable:

\[
D_o/2 = 0.0225 \pm 5.10^{-5} \text{ m} \tag{2-32}
\]

\[
D_i/2 = 0.015 \pm 5.10^{-5} \text{ m} \tag{2-33}
\]

The error in the value for the fictive distance \(x_f\) results from analyses with the FIESTA computer program, reported by SMIT [2.11] and in subsection 2.9:

\[
x_f = 0.000325 \pm 7.5 \times 10^{-5} \text{ m} \tag{2-34}
\]

The calculation of the total error in the heat transfer coefficient is based on equation (2-24):

\[
\alpha_{w,1} = \frac{1}{\frac{1}{h_1} - \frac{1}{2\lambda_w} \ln \left( \frac{D_i/2 + x_f}{D_i/2} \right)} = \frac{1}{Q} \tag{2-35}
\]

The error in the heat transfer coefficient reads:

\[
\Delta\alpha_{w,1} = \frac{1}{Q^2} \Delta Q = \alpha_{w,1}^2 \Delta Q \tag{2-36}
\]
The error in \(Q\) can be expressed as:

\[
\Delta Q = \frac{1}{h_i} \Delta h_i + \frac{D_i}{2\lambda_w} \ln \left( \frac{D_i/2+x_f}{D_i/2} \right) \Delta \lambda_w + \left| \frac{1}{\lambda_w} \ln \left( \frac{D_i/2+x_f}{D_i/2} \right) - \frac{1}{\lambda_w} \right| \Delta D_i/2 + \\
\frac{D_i}{\lambda_w (D_i/2+x_f)} \Delta \left( D_i/2+x_f \right)
\]

(2-37)

The error in the overall heat transfer coefficient is derived from equation (2-23):

\[
\Delta h_i = \frac{h_i}{\phi_{w,z}} \Delta \phi_{w,z} + \frac{h_i}{D_i/2} \Delta (D_i/2) + \frac{h_i}{(T_i-x_{f,z} - T_{b,1,z})} \Delta (T_i-x_{f,z} - T_{b,1,z})
\]

(2-38)

and the error calculation in the heat flux reads is derived from equation (2-22):

\[
\Delta \phi_{w,z} = \frac{\phi_{w,z}}{\lambda_w} \Delta \lambda_w + \frac{\phi_{w,z}}{(T_o-x_{f,z} - T_i-x_{f,z})} \Delta (T_o-x_{f,z} - T_i-x_{f,z}) + \\
\frac{1}{(\ln \left( \frac{D_o/2-x_f}{D_i/2+x_f} \right))^2} \cdot \frac{2\pi \lambda_w (T_o-x_{f,z} - T_i-x_{f,z})}{(D_o/2-x_f)} \Delta (D_o/2-x_f) + \\
\frac{1}{(\ln \left( \frac{D_o/2-x_f}{D_i/2+x_f} \right))^2} \cdot \frac{2\pi \lambda_w (T_o-x_{f,z} - T_i-x_{f,z})}{(D_i/2+x_f)} \Delta (D_i/2+x_f)
\]

(2-39)

From the calibration results as described by SCHOEN [2.4] the following maximum errors in temperature measurements are:

\(T_{o,x_{f,z}} : \pm 0.16 \degree C\)

\(T_{i,x_{f,z}} : \pm 0.16 \degree C\)

\(T_{b,1,z} : \pm 0.18 \degree C\)
The conductivity of tube wall material $\lambda_w$ reads:

$$\lambda_w = 47.12 + 0.0514 \ T_w \ W/m \ K$$  \hspace{1cm} (2-40)

with:

$$T_w = \text{average wall temperature at location } z$$

Since the calculated total error is strongly dependent on the temperature difference over the tube wall two typical calculations are presented below in table 2-3 to indicate the maximum occurring error with two extreme temperature differences over the tube wall.

<table>
<thead>
<tr>
<th>measurement $\rho x_f, z$</th>
<th>unit</th>
<th>probe position $(T_{D,1,z})$</th>
<th>SC-8</th>
<th>SC-12</th>
<th>SC-13</th>
</tr>
</thead>
<tbody>
<tr>
<td>$T_{0,x_f,z}$ $^\circ C$</td>
<td>± 0.16</td>
<td>101.66</td>
<td>71.13</td>
<td>111.95</td>
<td></td>
</tr>
<tr>
<td>$T_{1,x_f,z}$ $^\circ C$</td>
<td>± 0.16</td>
<td>98.07</td>
<td>62.04</td>
<td>105.93</td>
<td></td>
</tr>
<tr>
<td>$T_{b,1,z}$ $^\circ C$</td>
<td>± 0.18</td>
<td>91.46</td>
<td>45.06</td>
<td>96.48</td>
<td></td>
</tr>
<tr>
<td>$T_{0,x_f,z} - T_{1,x_f,z}$ $^\circ C$</td>
<td>± 0.32</td>
<td>3.59</td>
<td>9.09</td>
<td>6.02</td>
<td></td>
</tr>
<tr>
<td>$T_{1,x_f,z} - T_{b,1,z}$ $^\circ C$</td>
<td>± 0.34</td>
<td>6.61</td>
<td>16.98</td>
<td>9.45</td>
<td></td>
</tr>
</tbody>
</table>

$\lambda(T_{D,1,z})$ W/m K $\pm 0.05$ 52.25 (99.86 $^\circ C$) 50.54 (66.58 $^\circ C$) 52.719 (108.94 $^\circ C$)

$D/2$ m $\pm 5.10^{-3}$ 0.0225 0.0225 0.0225

$D_{i}/2$ m $\pm 5.10^{-3}$ 0.015 0.015 0.015

$x_f$ m $\pm 7.5.10^{-5}$ 0.000325 0.000325 0.000325

<table>
<thead>
<tr>
<th>maximum error</th>
<th>maximum error</th>
<th>maximum error</th>
</tr>
</thead>
<tbody>
<tr>
<td>% abs.</td>
<td>% abs.</td>
<td>% abs.</td>
</tr>
<tr>
<td>$\dot{q}_{w,z}$ J/m s</td>
<td>9.50</td>
<td>303</td>
</tr>
<tr>
<td>$h_i$ J/m² s $^\circ C$</td>
<td>14.99</td>
<td>768</td>
</tr>
<tr>
<td>$a_{w,1}$ J/m² s $^\circ C$</td>
<td>17.21</td>
<td>911</td>
</tr>
</tbody>
</table>

TABLE 2-3.

Maximum error calculation in $\phi$, $h$ and $\alpha$ for measurements SC-8, SC-9 and SC-13.
2.8. **Determination of the error in the porosity.**

The error in the porosity consists of an error due to temperature difference over the length of the fluidized bed as well as an error due to tolerance in equipment dimensions and fluidized bed length. The porosity $\varepsilon$ is defined as:

$$\text{porosity } \varepsilon = 1 - \frac{\text{volume of the particles}}{\text{volume of the empty tube over the fluidization height}}$$

(2-41)

The porosity as applied in this chapter is an average value over the entire fluidization of the tube. Under stationary conditions of mass flow and temperature difference between outlet and inlet of the tube, the superficial velocity $u_{l,s}$ will increase with increasing bulk temperature. The porosity as a consequence will change over the height of the heat exchanger tube. Assuming constant porosity neglects this variation and is therefore estimated below.

The definition of the superficial velocity $u_{l,s}$ reads:

$$u_{l,s} = \frac{1}{\frac{\pi}{4} D_i^2 \rho_1}$$

(2-42)

or:

$$u_{l,s} = \frac{D}{\rho_1}$$

(2-43)

with:

$\rho_1 = \text{density of the liquid}$

and:

$$D = \frac{1}{\frac{\pi}{4} D_i^2 \rho_1}$$

(2-44)

To describe the superficial velocity $u_{l,s}$ as a function of the porosity $\varepsilon$ it requires the expression for the terminal velocity $u_\infty$ of one single sphere or particle in an infinite stationary volume.
The expression for the terminal velocity results from equating gravity and drag forces exerted on the particle and reads:

\[
    u_\infty = \left( \frac{4}{3} \frac{d_p}{\rho_1} \frac{g_n}{C_{d_\infty}} - \frac{4}{3} \frac{d_p}{\rho_1} \frac{g_n}{C_{d_\infty}} \right) = \left( \frac{B}{\rho_1} - A \right)
\]

(2-45)

with:

- \(d_p\) = particle diameter
- \(\rho_s\) = density of the particle material
- \(C_{d_\infty}\) = drag coefficient stagnant liquid

with:

\[
    A = \frac{4}{3} \frac{d_p}{C_{d_\infty}} g_n \]

(2-46)

and:

\[
    B = \frac{4}{3} \frac{d_p}{C_{d_\infty}} \frac{\rho_s}{g_n} \]

(2-47)

The drag coefficient is a function of the Reynolds number defined as:

\[
    Re_d = \frac{\rho_1 u_\infty}{\eta_1} \frac{d_p}{\eta_1} \]

(2-48)

For \(Re_d > 1000\) the value of the drag coefficient becomes a constant value: 0.44.

The drag coefficient will only become dependant on the Reynolds number for the particles if particles with a diameter of 1 mm are used and then only for the lower temperatures.

For almost the entire range of experiments and for stationary conditions the parameter A and B can be considered as constants.

Correction of the terminal velocity \(u_\infty\) for an environment of a tube wall for a single particle in the tube yields according to RICHARDSON and ZAKI [2.16]:

\[
    u_1 = u_\infty C
\]

(2-49)
with:

\[ C = 10 \frac{d}{D_i} \]  

(2-50)

The equation for the falling velocity for a swarm of particles in a stagnant liquid \( u_{sw} \) according to RICHARDSON and ZAKI [2.16] reads:

\[ u_{sw} = u_i \epsilon^n \]  

(2-51)

with:

\[ \epsilon = \text{porosity of a swarm of particles} \]

\[ n = \text{exponent} \]

The exponent \( n \) results from the expressions:

\[ n = 2.39 \quad , \quad \text{Re}_{d_{\infty}} > 500 \]  

(2-52)

\[ n = 4.45 \text{Re}_{d_{\infty}}^{-0.1} \quad , \quad 200 \leq \text{Re}_{d_{\infty}} < 500 \]  

(2-53)

\[ n = \left( 4.45 + 18 \frac{d}{D_i} \right) \text{Re}_{d_{\infty}}^{-0.1} \quad , \quad 1 \leq \text{Re}_{d_{\infty}} < \]  

(2-54)

Physically there is no difference between the falling velocity of a swarm of particles in a tube filled with a stagnant liquid and the superficial velocity of a liquid in a tube in which solid particles are fluidized, providing that for both systems the fluids, the particles and the porosities are equal. This means that the velocity resulting from equation (2-51) can also be used for the calculation of the mass flow through a tube in which fluidization of solid particles takes place.

With the above:

\[ u_{l,s} = u_{sw} \]  

(2-55)

the porosity can now also be written as:

\[ \epsilon = \left( \frac{u_{l,s}}{u_i} \right)^{1/n} \]  

(2-56)
The only temperature dependent parameter in equation (2-56) is now implicitly the liquid density \( \rho_1 \).

The error in the porosity as a consequence of heating of the liquid can be calculated as follows:

\[
\Delta \varepsilon = \left| \frac{\partial \varepsilon}{\partial u_1, s} \right| \Delta u_1, s + \left| \frac{\partial \varepsilon}{\partial u_1} \right| \Delta u_1
\]  
(2-57)

where:

\[
\frac{\partial \varepsilon}{\partial u_1, s} = \frac{u_1, s}{n u_1^{1/n}} (1/n-1)
\]  
(2-58)

and:

\[
\frac{\partial \varepsilon}{\partial u_1} = - \frac{u_1, s}{n u_1^{1/(n+1)}}
\]  
(2-59)

with equation (2-42) we can express \( \Delta u_1, s \) as:

\[
\Delta u_1, s = \left| \frac{\partial u_1, s}{\partial \rho_1} \right| \Delta \rho_1 = \frac{D}{\rho_1^2} \Delta \rho_1
\]  
(2-60)

Equation (2-45) and (2-49) yield:

\[
\Delta u_1 = \left| \frac{\partial u_1}{\partial \rho_1} \right| \Delta \rho_1 = \left| \frac{C}{2 \left( \frac{B}{\rho_1^2} - A \right)^{1/2}} \frac{B}{\rho_1^{1/2}} \right| \Delta \rho_1
\]  
(2-61)

or:

\[
\Delta u_1 = \frac{BC}{2\rho_1^2 \left( \frac{B}{\rho_1} - A \right)^{1/2}} \Delta \rho_1
\]  
(2-62)

Table 2-4 shows the actual values of the parameters, the calculated constants \( A, B, C \) and \( D \) and the maximum error in the porosity \( \varepsilon \) due to negligence of the temperature increase in the heat exchanger tube. The value of \( \Delta \rho_1 \) is the actual difference in the density of the liquid when heated from \( \theta_{in} \) to \( \theta_{out} \).
<table>
<thead>
<tr>
<th>parameter</th>
<th>unit</th>
<th>specific conditions and results</th>
</tr>
</thead>
<tbody>
<tr>
<td>( \Delta (\theta_{\text{out}} - \theta_{\text{in}}) )</td>
<td>°C</td>
<td>20</td>
</tr>
<tr>
<td>( d_p )</td>
<td>m</td>
<td>0.001</td>
</tr>
<tr>
<td>( \rho_s )</td>
<td>kg/m³</td>
<td>2470</td>
</tr>
<tr>
<td>( D_i )</td>
<td>m</td>
<td>0.030</td>
</tr>
<tr>
<td>( \rho_{li}(110 , ^°C) )</td>
<td>kg/m³</td>
<td>950</td>
</tr>
<tr>
<td>( \varepsilon )</td>
<td>-</td>
<td>0.85</td>
</tr>
<tr>
<td>( \text{Re}_{d_p} )</td>
<td>-</td>
<td>802</td>
</tr>
<tr>
<td>( C_{d_{\infty}} )</td>
<td>-</td>
<td>0.47</td>
</tr>
<tr>
<td>( u_i )</td>
<td>m/s</td>
<td>0.1954</td>
</tr>
<tr>
<td>( u_{l,s} )</td>
<td>m/s</td>
<td>0.1325</td>
</tr>
<tr>
<td>( \phi_m )</td>
<td>kg/s</td>
<td>0.0889</td>
</tr>
<tr>
<td>( A^* )</td>
<td>m²/s²</td>
<td>0.0278</td>
</tr>
<tr>
<td>( B^* )</td>
<td>kg/m s²</td>
<td>68.74</td>
</tr>
<tr>
<td>( C^* )</td>
<td>-</td>
<td>0.926</td>
</tr>
<tr>
<td>( D^* )</td>
<td>kg/m² s</td>
<td>125.80</td>
</tr>
</tbody>
</table>

**error in terms**

| \( \Delta u_{i,s} \) | * \(10^{-3} \) | 0.976 | 0.614 | 1.360 | 0.854 | 1.520 | 0.957 |
| \( \Delta u_i \)    | * \(10^{-3} \) | 1.170 | 1.170 | 1.595 | 1.595 | 1.803 | 1.803 |
| \( \frac{\partial E}{\partial u_{i,s}} \) | 2.6841 | 3.5158 | 1.9297 | 2.5272 | 1.7206 | 2.2543 |
| \( \frac{\partial E}{\partial u_i} \)   | 1.8200 | 1.4987 | 1.3080 | 1.0776 | 1.1668 | 0.9607 |

**calculated error in**

| abs | 0.00475 | 0.00391 | 0.00471 | 0.00388 | 0.00472 | 0.00389 |
| %   | 0.56    | 0.56    | 0.55    | 0.55    | 0.56    | 0.56    |

\[
A^* = \frac{4}{3} \frac{d_p}{C_{d_{\infty}}} \quad B^* = \frac{4}{3} \frac{d_p}{C_{d_{\infty}}} \quad C^* = 10 \quad D^* = \frac{\phi_m}{\frac{\pi}{4} D_i^2}
\]

**TABLE 2-4.**

Calculation of the error in the porosity for \( d_p = 0.001, 0.002 \) and \( 0.003 \) m at 110 °C.
<table>
<thead>
<tr>
<th>parameter</th>
<th>unit</th>
<th>specific conditions and results</th>
</tr>
</thead>
<tbody>
<tr>
<td>( \theta_{\text{out}} - \theta_{\text{in}} )</td>
<td>°C</td>
<td>120 - 100 120 - 100 50 - 30 50 - 30</td>
</tr>
<tr>
<td>( d_p )</td>
<td>m</td>
<td>0.001 0.001 0.001 0.001</td>
</tr>
<tr>
<td>( \rho_s )</td>
<td>kg/m³</td>
<td>2470 2470 2470 2470</td>
</tr>
<tr>
<td>( D_1 )</td>
<td>m</td>
<td>0.030 0.030 0.030 0.030</td>
</tr>
<tr>
<td>( \rho_{l0} )</td>
<td>kg/m³</td>
<td>950 950 992 992</td>
</tr>
<tr>
<td>( c )</td>
<td></td>
<td>0.85 0.70 0.85 0.70</td>
</tr>
<tr>
<td>( R_{\text{e}} )</td>
<td>-</td>
<td>802 802 247.5 247.5</td>
</tr>
<tr>
<td>( C_{\text{d}_c} )</td>
<td>-</td>
<td>0.47 0.47 0.74 0.74</td>
</tr>
<tr>
<td>( u_{l} )</td>
<td>m/s</td>
<td>0.1954 0.1954 0.1504 0.1504</td>
</tr>
<tr>
<td>( u_{z_s} )</td>
<td>m/s</td>
<td>0.1325 0.0833 0.0992 0.0603</td>
</tr>
<tr>
<td>( \phi_m )</td>
<td>kg/s</td>
<td>0.08893 0.05592 0.06658 0.04051</td>
</tr>
<tr>
<td>( A )</td>
<td>m²/s²</td>
<td>0.0278 0.0278 0.0177 0.0177</td>
</tr>
<tr>
<td>( B )</td>
<td>kg/m s²</td>
<td>68.74 68.74 43.71 43.71</td>
</tr>
<tr>
<td>( C )</td>
<td></td>
<td>0.926 0.926 0.926 0.926</td>
</tr>
<tr>
<td>( D )</td>
<td>kg/m² s</td>
<td>125.8 79.15 94.24 57.34</td>
</tr>
</tbody>
</table>

**error in terms**

\[
\Delta u_{\bar{z},s} = 10^{-3}
\]
\[
\Delta u_{\bar{v}} = 10^{-3}
\]
\[
\frac{2c}{\Delta u_{\bar{z},s}} = 2.6841 3.5158 3.5438 4.7338
\]
\[
\frac{2c}{\Delta u_{\bar{v}}} = 1.8200 1.4987 2.2076 1.8180
\]

**calculated error in porosity**

\[
\text{abs} = 0.00475 0.00391 0.00248 0.00203
\]
\[
\% = 0.56 0.56 0.29 0.29
\]

**TABLE 2-5.**

Calculation of the error in the porosity for \( d_p = 0.001 \) m and for different temperature levels.

\( A \) = \( \frac{4}{3} \frac{d_p \varphi_n}{C_{\text{d,c}}^{\infty}} \)

\( B \) = \( \frac{4}{3} \frac{d_p \varphi_n \rho_s}{C_{\text{d,c}}^{\infty}} \)

\( C \) = \( 10 \)

\( D \) = \( \frac{\phi_m}{d_1^2} \)

- 2.43 -
Table 2-5 indicates the influence of the temperature level on the porosity for the particle diameter \( d_p = 0.001 \text{ m} \). The error in the porosity resulting from the equipment dimensions and the error in the fluidized bed length is calculated as follows:

\[
\varepsilon = 1 - \frac{M_p}{\rho_s A_t L_{fl}}
\]  

(2-63)

where:

\( M_p \) = mass of particles

\( A_t \) = cross section area of the tube

\( L_{fl} \) = fluidization length

The error in equation (2-63) reads:

\[
\Delta P = \frac{\partial P}{\partial \rho_s} \Delta \rho_s + \frac{\partial P}{\partial A_t} \Delta A_t + \frac{\partial P}{\partial L_{fl}} \Delta L_{fl} + \frac{\partial P}{\partial M_p} \Delta M_p
\]  

(2-64)

with:

\[
P = \frac{M_p}{\rho_s A_t L_{fl}}
\]  

(2-65)

<table>
<thead>
<tr>
<th>parameter</th>
<th>unit</th>
<th>error</th>
<th>( d_p = 0.001 \text{ m} )</th>
<th>( d_p = 0.002 \text{ m} )</th>
<th>( d_p = 0.003 \text{ m} )</th>
</tr>
</thead>
<tbody>
<tr>
<td>( \rho_s )</td>
<td>kg/m³</td>
<td>±60</td>
<td>2470</td>
<td>2554</td>
<td>2520</td>
</tr>
<tr>
<td>( A_t )</td>
<td>m²</td>
<td>±3.10⁻⁶</td>
<td>707.10⁻⁶</td>
<td>707.10⁻⁶</td>
<td>707.10⁻⁶</td>
</tr>
<tr>
<td>( L_{fl} )</td>
<td>m</td>
<td>±0.032</td>
<td>2.208</td>
<td>2.208</td>
<td>2.208</td>
</tr>
<tr>
<td>( m_p )</td>
<td>kg</td>
<td>±0.0005</td>
<td>0.576</td>
<td>0.596</td>
<td>0.588</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>parameter</th>
<th>unit</th>
<th>error</th>
<th>( d_p = 0.001 \text{ m} )</th>
<th>( d_p = 0.002 \text{ m} )</th>
<th>( d_p = 0.003 \text{ m} )</th>
</tr>
</thead>
<tbody>
<tr>
<td>porosity ( \varepsilon = 0.85 )</td>
<td>-</td>
<td>abs.</td>
<td>±0.0066</td>
<td>±0.0064</td>
<td>±0.0065</td>
</tr>
<tr>
<td></td>
<td></td>
<td>%</td>
<td>0.77</td>
<td>0.76</td>
<td>0.76</td>
</tr>
<tr>
<td>porosity ( \varepsilon = 0.70 )</td>
<td>-</td>
<td>abs.</td>
<td>±0.0130</td>
<td>±0.0127</td>
<td>±0.01285</td>
</tr>
<tr>
<td></td>
<td></td>
<td>%</td>
<td>1.86</td>
<td>1.82</td>
<td>1.84</td>
</tr>
</tbody>
</table>

TABLE 2-6.
Calculation of the error in the porosity as a result of tolerances in dimensions and particle mass.

- 2.44 -
and:

$$\Delta \epsilon = \Delta p$$  \hspace{1cm} (2-66)

Table 2-6 presents actual values of the parameters and the calculated maximum error in the porosity as a result of equipment dimensions, particle mass and fluidization length. Combination of the calculated maximum errors in the porosity as shown in tables 2-4, 2-5 and 2-6 indicates that the maximum error in the porosity occurs in the application of particles of 1 mm and the lowest porosity of the fluidized bed (0.70) and amounts to 3 percent of the value for the porosity.

<table>
<thead>
<tr>
<th></th>
<th>$d_p = 0.001 \text{ m}$</th>
<th>$d_p = 0.002 \text{ m}$</th>
<th>$d_p = 0.003 \text{ m}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\epsilon = 0.85$</td>
<td>$\Delta \epsilon$</td>
<td>0.01135</td>
<td>0.01111</td>
</tr>
<tr>
<td></td>
<td>$\Delta \epsilon \text{ in } %$</td>
<td>1.33</td>
<td>1.30</td>
</tr>
<tr>
<td>$\epsilon = 0.70$</td>
<td>$\Delta \epsilon$</td>
<td>0.01691</td>
<td>0.01658</td>
</tr>
<tr>
<td></td>
<td>$\Delta \epsilon \text{ in } %$</td>
<td>2.42</td>
<td>2.37</td>
</tr>
</tbody>
</table>

**TABLE 2-7.**

Combination of partial calculated errors in the porosity and expressing the maximum error.
2.9. Measurements of temperatures with imbedded thermocouples.

Tube wall temperature measurement without disturbance of the condensate film on the outside and without an obstruction for the fluidized bed on the inside of tube can be performed by a system where wall temperature thermocouples have been applied which are imbedded in radial machined grooves in the heat exchanger tube, as can be seen in figure 2-18.

FIGURE 2-18.
Imbedded thermocouple in heat exchanger tube wall, actual thermocouple is shown left of A and after almost covering the full circumference the heat leaves the tube wall at the right side of E.

In the direction of the heat flux we encounter now a series of heat resistances caused by solder material and the various components of the thermocouple. The temperature distribution in the immediate surrounding of the thermocouple will therefore be disturbed, which results in a change in heat flux distribution.

2.9.1. FIESTA application for the determination of $x_f$.

The purpose of the work performed and described in this subchapter is to establish reliable values for a fictive distance $x_f$ from the outer or inner side of the tube wall corresponding with the position of the
isotherm as measured with the thermocouple in stationary condition but now in undisturbed tube wall material. The fictive distance $x_f$ and a typical isotherm are illustrated in figure 2-19.

**FIGURE 2-19.**
Isotherm through thermocouple into undisturbed material.

**FIGURE 2-20.**
Composition of insulated shielded type thermocouple.

Thermocouples used for temperature registration in the tube wall are of the insulated shielded type as shown in figure 2-20.

Computation of the temperature distribution in the surrounding area of the thermocouple is based on the application of the computer code FIESTA-II (Finite Elements Stress and Temperature Analysis) of the Energy Centre of the Netherlands. This program is based on the finite elements method for calculation of stationary temperature distribution in flat rotation symmetric and three-dimensional constructions.
The following boundary conditions are applied for the calculation of
temperatures within the observed construction and under stationary
conditions:

- fixed heat flux imposed on a side of an element or segment
- heat transfer coefficient to adjacent liquid
- temperature of the adjacent liquid.

Out of the possible elements of the program the following were applicable
for solving the present problem:

SEG 8 - three-dimensional annular segment
RING 4 - annular segment with square cross section
RING 3 - annular segment with triangular cross section
SEG 4 - two-dimensional annular segment.

First a geometry has to be defined by an orthogonal matrix of mesh lines.
This matrix also fixed the axes of the coordinate system. In order to
minimize the input the computer program FIESTA-II is equipped with an
advanced mesh generator.

FIGURE 2-21.
Definition of typical elements for FIESTA calculations (longitudinal/radial).

- 2.48 -
Elements can be defined per element or for those with the same boundary conditions per set of elements. A group of elements with the same boundary conditions is treated as a concatenated block of elements. The definition of the elements also needs the specific values of the boundary conditions and the tabulated material properties from a data bank.

The calculation of the temperature distribution over the observed construction can be made more precise within the accuracy of the boundary conditions, by a division into a larger number of elements.

In the output of the program the calculated temperatures will be presented at the nodal points given by the matrix of the mesh lines.

Figures 2-21 and 2-22 presents a typical element definition for the present problem of calculation of isothermals around the imbedded thermocouple.

**FIGURE 2-22.**
Definition of typical elements for FIESTA calculations (tangential/radial).
2.9.2. Determination of $x_f$.

Because of the application of thermocouples in radial machined grooves, where the heat flux encounters a series of heat resistances as explained in the general description, the temperature distribution compared with undisturbed wall material will change. These deviations will occur in radial, axial and tangential direction of the tube wall, creating a three-dimensional disturbance around each of the thermocouples.

For the performance of the three-dimensional temperature calculations, the element SEG 8 has been the basis for the division of the tube wall. An illustration of the tube wall division into SEG 8 elements is given in figure 2-23.

---

**FIGURE 2-23.**

*Illustration of a three-dimensional annular segment.*

A radial cross section through the thermocouple in the groove has only been made where axial disturbances only are calculated in one direction. This is used when the cross section ABM is symmetrical which is the case when both thermocouple conductors are situated in the cross section. The fictive distance $x_f$ can be determined if the isotherm through the measuring point is followed in radial direction up to the point where the influence of the radial disturbance disappeared.
In addition this isotherm will be followed in the axial cross section BCFG into the undisturbed tube wall material and up to the point where the isotherm in relation to the tube wall remains in a constant position.

The distance between the isotherm at that point and the outer side of the tube wall is the arithmetic value for the fictive distance $x_f$ to be used in the heat transfer calculations.

For the six limiting surfaces of the three-dimensional annular segment SEG 8 the various boundary conditions as heat flux $\phi$, heat transfer coefficient $a$ and temperature $T$ have to be determined and imposed upon as constraints. For the inner side and the outer side of the annular segment the boundary conditions for heat transfer coefficient and temperature are given and can be varied to compute various cases. For the cross section ABCD is a heat flux equal to zero in force because of symmetrical conditions.

The cross section EFGH is chosen in the area where the axial disturbance disappeared. In order to establish values for cylindrical length of the annular segment, i.e. the length between FG and BC, computations are performed with the defined annular segments with square cross section, RING 4 and annular segments with triangular cross section, RING 3.

The cross section ADHE is chosen to be consistent with the cross section A-D as shown in figure 2-18. In this cross section the influence of radial disturbance at the spot where the thermocouple conductors axially leave the groove and are guided outside the tube wall material, is assumed to be the same as at the actual place of the thermocouple. As a consequence the heat flux through the cross section ADHE equals zero. The other axial boundary cross section of the three-dimensional annular segment SEG 8, the cross section BCGF, is chosen where the influence of the disturbance by the thermocouple in a radial way disappeared. As a consequence also the heat flux through the cross section BCGF equals zero. In order to fulfil the requirements of this assumption radial temperature computations are performed to establish the radial length of the annular segment i.e. the length between BC and AD. The radial temperature calculations are performed with the two-dimensional annular segment SEG 4.

The division into SEG 4 elements can be seen in figure 2-22.
All computations to establish accurate values for the fictive distance \( x_f \) have been performed for different values for heat flux, heat transfer coefficients and temperature levels. Minimum and maximum values of these quantities covered the entire range of the actual heat transfer data as used with or resulted from the experiments as described in this chapter. As the actual position of the thermocouple conductors could only be established with destructive testing afterwards, a series of computations have been performed for different positioning of the conductors within the thermocouple shield. Material properties are given in table 2-8.

<table>
<thead>
<tr>
<th>item</th>
<th>material</th>
<th>thermal conductivity at 20 °C in W/m °K</th>
</tr>
</thead>
<tbody>
<tr>
<td>tube wall material</td>
<td>copper nickel 90/10</td>
<td>48.3</td>
</tr>
<tr>
<td>solder material</td>
<td>castolin (40% Ag)</td>
<td>108.9</td>
</tr>
<tr>
<td>thermocouple conductor A</td>
<td>chromel</td>
<td>19.2</td>
</tr>
<tr>
<td>thermocouple conductor B</td>
<td>alumel</td>
<td>29.7</td>
</tr>
<tr>
<td>thermocouple insulation material</td>
<td>magnesium oxide (MgO)</td>
<td>3.0</td>
</tr>
<tr>
<td>thermocouple shield</td>
<td>stainless steel AISI 321</td>
<td>21.4</td>
</tr>
</tbody>
</table>

**TABLE 2-8.**
Material properties of tube wall, solder material and thermocouple elements.

Finally the computed fictive distances \( x_f \) for a range of heat flux and an average temperature of 75 °C are presented in table 2-9.

<table>
<thead>
<tr>
<th>heat flux in W/m² (tube 0.045/0.030 m)</th>
<th>fictive distance ( x_f ) in m × 10^{-3}</th>
</tr>
</thead>
<tbody>
<tr>
<td>3000</td>
<td>0.320</td>
</tr>
<tr>
<td>4000</td>
<td>0.321</td>
</tr>
<tr>
<td>5000</td>
<td>0.323</td>
</tr>
<tr>
<td>6000</td>
<td>0.325</td>
</tr>
<tr>
<td>7000</td>
<td>0.326</td>
</tr>
<tr>
<td>8000</td>
<td>0.328</td>
</tr>
<tr>
<td>9000</td>
<td>0.330</td>
</tr>
</tbody>
</table>

**TABLE 2-9.**
Calculated fictive distances \( x_f \) for different values of heat flux.
2.9.3. Analysis of the error in $x_f$.

Analysis made by SMIT [2.11] reveal that the maximum error in a constant fictive distance $x_f$ consists of three major components:

a) Influence of the heat flux.
Variation of the heat flux from 3000 W/m and based on the outer diameter of the heat exchanger tube of 0.045 m resulted in a maximum error in the fictive distance $x_f$ equals $0.325 \pm 0.0005$ mm.

b) Error due to unknown positioning of thermocouple conductors.
Calculations of the fictive distance $x_f$ based on radial and axial positioning of the thermocouple conductors as well as intermediate positions showed a maximum error in $x_f$ equalling $0.325 \pm 0.025$ mm.

c) Error due to mechanical imperfections.
Analysis of the mechanical, dimensional and application imperfections resulted in the maximum error in $x_f$ of $0.325 \pm 0.040$ mm.

All calculations for the heat transfer coefficients as described in this chapter are based on a fixed value for the fictive distance with a value and a maximum total error:

$$x_f = 0.325 \pm 0.075 \text{ mm}$$

2.10. Remarks and conclusions.

Observation of the results of the measurements in respect of tube wall temperatures with method B revealed the following:

- tube wall temperatures measured on the outside indicate a non-uniform temperature profile over the length of the heat exchanger tube. The observed temperature profile has the shape of a "wave" at higher temperature levels;

- tube wall temperatures measured on the inside show a smoothed profile.

As observed by POTZE [2.12] and studies by BENJAMIN [2.13] and MARSCHAL [2.14] the condensate running down the vertical heat transfer tube will show a curved surface. The waves, or ripples on the surface are caused by the influences of surface tension and gravity forces exercised on the free surface of the condensate film.
The observed almost stationary waves lead to a non-uniform distribution of the heat flux to the heat exchanger tube and introduce a divergence in the calculated heat transfer coefficients. Figures 2-24 and 2-25 show the measured temperature at the outside and the inside of the heat exchanger tube for measurements with porosity of 0.85 and particle size of 3 mm.

**FIGURE 2-24.**

Temperature profiles at the outside of the heat exchanger tube wall for various measurements (SC-1-1 through SC-1-7).

From the measured temperatures at the inner side of the heat exchanger an observation can be made that through the mechanism of axial heat flux in the tube wall smoothing of the temperature profile occurs. With the applied measuring methods as described in this chapter it is of importance to verify the extend of axial heat flux through the tube because of its influence in the calculated local heat transfer coefficient from the measured temperatures in a radial cross section of the tube. In order to investigate the influence of the axial heat flux in this three-dimensional conductivity mechanism, a finite element approach has been followed.
FIGURE 2-25.

Temperature profiles at the inside of the heat exchanger tube wall for various measurements (SC-1-1 through SC-1-7).

The calculations are based on the tube temperatures on the outside of the wall as found from the highest temperature measurement with particles of 3 mm and a porosity of 0.85. Under the assumption of a smooth wall temperature profile at the outside, this profile has been used as the boundary condition for the heat conductivity calculation. Computations are performed with the MARC [2.15] computer program. Figure 2-26 shows the temperature profile.

A heat flux with a value based on the temperature difference between the outside of the tube wall and the saturated steam temperature in the vapour space has been applied as heat flux constraint.

The calculations are based on the assumption of constant distribution of the heat flux on the fluidized bed side of the heat exchanger tube wall. Temperature and heat flux are assumed to be axial symmetric.
FIGURE 2-26.
Temperature profile (measurement SC-1-7) at the outside of the tube wall and as boundary conditions for the MARC calculations.

The heat exchanger tube is divided in 24 annular, axial symmetrical elements.
Figure 2-27 shows the defined elements and the imposed heat flux per element.

FIGURE 2-27.
Element definition for the finite elements calculation with uniform heat flux absorption in the fluidized bed.
The results of the finite elements computations are presented in figure 2-28.

**FIGURE 2-28.**

Presentation of calculation and measured temperatures at the inside of the heat exchanger tube.

As can be seen from figure 2-28 the above mentioned calculations with the MARC computer code reveals roughly the same tendency as the temperature profile originating from the measurements of the inside of the heat exchanger wall. Basic conclusions of the above calculations are that the axial conductivity through the tube wall can explain some smoothing of the temperature profiles and that the assumption of uniform distribution of the heat flux on the fluidized bed side is reasonable. The latter is of importance for further investigations on the wall-to-liquid heat transfer coefficients in a fluidized bed. Further improvements of the above mentioned analysis can be made by enhancing the used model with respect to introduction considerations of heat transfer at the condensing steam side.
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scale is defined as a deposit formed by precipitation. It is generally referred to as a "precipitation scale." It is formed by the reaction of calcium and magnesium ions with carbonate ions to form calcium carbonate and magnesium carbonate, respectively. This scale formation can be prevented by using inhibitors to prevent the formation of scale. The reaction involved is as follows:

\[ \text{Ca}^{2+} + \text{CO}_3^{2-} \rightarrow \text{CaCO}_3 \]
CHAPTER 3 SCALE AND FOULING BEHAVIOUR OF A LIQUID-SOLID FLUIDIZED BED HEAT EXCHANGER

Figure 3-0 gives an outline of this chapter on scaling and fouling behaviour of a liquid-solid fluidized bed heat exchanger.

3. SCALE AND FOULING BEHAVIOUR

3.1. Introduction

3.2. Scale control
  - Polyphosphate compound control 3.2.1.
  - Acidity control 3.2.2.
  - Polymer additives control 3.2.3.
  - Calcium sulphate scale control 3.2.4.

3.3. Fluidized bed heat exchanger

3.4. Mechanisms of crystallization
  - Nucleation 3.4.1.
  - Crystal growth 3.4.2.

3.5. Sludge/fouling influence

3.6. Surface condition

3.7. Particle velocity, etc.
  - Particle velocity model approach 3.7.1.
  - Empirical relations 3.7.2.

3.8. Scale control experiments
  - Canal water 3.8.1.
  - Sea water 3.8.2.
  - Artificial sea water 3.8.3.

3.9. Concluding remarks
3.1. Introduction.

Scale is defined as a deposit formed by precipitation from solution of a substance with a solubility which decreases as the temperature increases, generally referred to as a substance with an inverse solubility curve. In a heat exchanger scale will mostly occur on the heated surface and form an insulating layer on this surface which greatly impedes the flow of heat. This scale formation on the heat exchanger surfaces has to be averted in order to keep the specific energy consumption well within reasonable limits. Despite of this commonly used definition, not all scale is formed by substances with an inverse solubility curve.

For example, two of the three most important scale formers encountered in sea water distillation are calcium carbonate (CaCO$_3$) and magnesium hydroxide (Mg(OH)$_2$). Although these materials do have inverse solubility curves, their deposition is also affected by a chemical reaction. Carbon dioxide is present in sea water primarily as the bicarbonate ion; when sea water is heated, part of the carbon dioxide is released as gas and the remainder is transformed to carbonate.

The reaction involved is as follows:

$$2 \text{HCO}_3^- + \text{CO}_3^{2-} + \text{CO}_2 \uparrow + \text{H}_2\text{O} \quad (3-1)$$

The carbonate ion reacts with calcium sea in the water to form calcium carbonate, which is relatively insoluble. Carbon dioxide is released primarily at the heating surface and at this point the insoluble calcium carbonate is formed as scale according to:

$$\text{Ca}^{2+} + \text{CO}_3^{2-} + \text{CaCO}_3 \quad (3-2)$$

Further decomposition of the carbonate may occur, described by the reaction:

$$\text{CO}_3^{2-} + \text{H}_2\text{O} \rightarrow \text{CO}_2 \uparrow + 2 \text{OH}^- \quad (3-3)$$

Subsequently the hydroxide ions may then react with magnesium ions to form a precipitation of magnesium hydroxide, governed by the following reaction:

$$\text{Mg}^{2+} + 2 \text{OH}^- + \text{Mg(OH)}_2 \uparrow \quad (3-4)$$
Whether calcium carbonate or magnesium hydroxide will be deposited depends primarily on the temperature. At temperatures from 65 to 85 °C, calcium carbonate predominates, while magnesium hydroxide can be formed at higher temperatures either directly as mentioned above or through the hydro-magnesite state into brucite according to:

$$3 \text{MgCO}_3 \cdot \text{Mg(OH)}_2 \cdot 3 \text{H}_2\text{O} \rightarrow 4 \text{Mg(OH)}_2 + 3 \text{CO}_2$$

(3-5)

One type of scale which is truly caused by the inverse solubility of a substance in solution and which is the third important scaling component encountered in sea water distillation, is calcium sulphate which is formed according to the reaction:

![Graph showing solubility curves of calcium sulphate scales for sea water.](image)

**FIGURE 3-1.**

The solubility curves of the calcium sulphate scales for sea water.
$$\text{Ca}^{2+} + \text{SO}_4^{2-} + x\text{H}_2\text{O} \rightarrow \text{CaSO}_4 \cdot x\text{H}_2\text{O} \quad (3-6)$$

where:

$$x = 0, \frac{1}{2} \text{ and } 2$$

Most encountered form of calcium sulphate is hemihydrate type ($\text{CaSO}_4 \cdot \frac{1}{2} \text{H}_2\text{O}$) while the other sulphate scales are of less concern. For example gypsum ($\text{CaSO}_4 \cdot 2\text{H}_2\text{O}$) becomes insoluble in sea water regardless of temperature when the concentration factor exceeds approximately three times the standard sea water concentration. However, such high concentration factors are not attractive in flash distillation.

<table>
<thead>
<tr>
<th>Ions</th>
<th>g/kg</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\text{SO}_4^{2-}$</td>
<td>2.7017</td>
</tr>
<tr>
<td>$\text{Cl}^-$</td>
<td>19.3605</td>
</tr>
<tr>
<td>$\text{Ca}^{2+}$</td>
<td>0.4081</td>
</tr>
<tr>
<td>$\text{Mg}^{2+}$</td>
<td>1.2975</td>
</tr>
<tr>
<td>$\text{Na}^+$</td>
<td>10.7678</td>
</tr>
<tr>
<td>$\text{K}^+$</td>
<td>0.3876</td>
</tr>
<tr>
<td>$\text{Sr}^{2+}$</td>
<td>0.0136</td>
</tr>
<tr>
<td>$\text{HCO}_3^-$</td>
<td>0.1425</td>
</tr>
<tr>
<td>$\text{Br}^-$</td>
<td>0.0659</td>
</tr>
<tr>
<td>$\text{F}^-$</td>
<td>0.0013</td>
</tr>
<tr>
<td>$\text{H}_2\text{BO}_3$</td>
<td>0.0265</td>
</tr>
<tr>
<td>$\text{I}^-$</td>
<td>0.00005</td>
</tr>
<tr>
<td>$\text{Si}^{4+}$</td>
<td>0.00002 to 0.004</td>
</tr>
<tr>
<td>others</td>
<td>0.0013</td>
</tr>
<tr>
<td><strong>Total Dissolved Solids (TDS)</strong></td>
<td>35.1745</td>
</tr>
<tr>
<td><strong>water</strong></td>
<td>964.8255</td>
</tr>
<tr>
<td><strong>Total</strong></td>
<td>1000.0000</td>
</tr>
<tr>
<td><strong>chlorinity</strong></td>
<td>19.381</td>
</tr>
<tr>
<td><strong>chlorosity</strong></td>
<td>19.862</td>
</tr>
<tr>
<td><strong>salinity</strong></td>
<td>35.010</td>
</tr>
<tr>
<td><strong>density (20 °C) in kg/m$^3$</strong></td>
<td>1024.8</td>
</tr>
</tbody>
</table>

**TABLE 3-1.**

_Ionic composition of standard sea water._
3.2. Scale control.

Chemical treatment is widely used in industry to remove ions that might otherwise cause scale formation. The most obvious method is water softening, by which calcium and magnesium ions are removed either by ion exchange resins or by treatment with soda ash and lime or caustic soda or sodium phosphate. Sulphate may also be removed by treatment with barium chloride.

However, there are more calcium, magnesium and sulphate ions present than will be involved in the formation of scale. The consumption of chemicals for sea water softening by chemical treatment is so high that the costs of these type of treatment become prohibitive.

Currently three different methods for control of calcium carbonate and magnesium hydroxide scale also referred to as alkaline scales are applied for commercial flash distillation.

The methods can be categorized:

a. low temperature - polyphosphate compound control
b. high temperature - acidity control
c. high temperature - polymer additive control.

3.2.1. Polyphosphate compound control.

This low temperature scale control (maximum 90 °C) uses a polyphosphate compound threshold treatment to promote the formation of a soft sludge of calcium carbonate and magnesium hydroxide instead of adherent scale. The mechanism by which these polyphosphate compound treatment prevents scale deposition is not clearly understood. It might be that the polyphosphate chain enters the crystal lattice while it is at an embryo stage and prevents further bound formation which results in a finely dispersed micro sludge.

As the temperature of the sea water or brine is increased, the polyphosphate molecules become hydrolysed. The subsequent hydrolysis reaction limits the use of polyphosphate compounds to a temperature of about 90 °C.

3.2.2. Acidity control.

Acid treatment was introduced in the 1960's as a mean of raising top brine temperature above 90 °C to enable the plant production capacity to be increased. The pH control method consists of supplying hydrogen ions to break down the bicarbonate ions according to the following equation:
\[
\text{HCO}_3^- + \text{H}^+ \rightarrow \text{H}_2\text{O} + \text{CO}_2^- 
\] (3-7)

Carbon dioxide formed in this reaction is usually released to the atmosphere in a degassing tower followed by a vacuum deaerator to reduce the oxygen content to a low level (< 0.1 mg/l). Main drawbacks of the acid dosing scale control method are the risks of corrosion in carbon steel evaporator, sometimes costs and availability and the hazard to personnel handling the acid.

3.2.3. Polymer additives control.

In order to extend the range of the additive dosing methods to higher than brine temperatures of 90 °C, very extensive research and development work has been carried out. Results of this work are different low molecular weight polymeric carbolic acids with a low polymerization degree. These polymers may act either as threshold inhibitors or by means of distortion of the lattice structure of the crystals or both, so as to reduce adhesion to other crystals or to metal surfaces.

Operations has shown that it is possible to control scale formation at brine temperature up to 110 °C.

3.2.4. Calcium sulphate scale control.

At the moment there is no reliable and simple method available to pass the calcium sulphate hemihydrate solubility limits. This means for conventional flash distillation plants in which recirculation of the brine is applied that a maximum brine temperature for safe continuous operation should be limited to 115 °C at a concentration of the brine of approximately 1.8 times that of standard sea water. Although the hemihydrate solubility line seems to allow for a higher operating temperature there are two reasons for limiting the maximum brine temperature to 115 °C at a concentration factor of 1.8.

The necessary temperature difference between the bulk of the liquid and the wall of the heat exchanger for a positive heat flux to the brine results in tube wall temperatures higher than the bulk temperature. Although the solubility line for the hemihydrate form of calcium sulphate scale may not yet been passed. A nucleation of scale based on anhydrous calciumsulphate scale as a result of the relative anhydrous supersaturation may occur. This also will lower the maximum operating temperature for scale-free operation.
3.3. The fluidized bed heat exchanger.

Early experiments showed that when applying a liquid-solid fluidized bed in the heat exchanger section, the solubility limits of substances with an inverse solubility curve could be passed to a certain extent without the forming of scale deposits on the heated surfaces. The polishing action of the particles of the fluidized bed removed the initially formed and deposited scale crystals from the heat exchanger surfaces.

The possibility of inhibiting and or removing of scale by the polishing action of the fluidized particles was in the process of developing the vertical evaporator noticed by KLAREN [3.2] in an experimental heat exchanger for the determination of the overall heat transfer coefficient of a liquid-solid fluidized bed system. During these experiments a solution was circulated through a test module and became supersaturated with calcium sulphate above a certain temperature. Operation for a period of six hours without fluidized particles resulted in a clearly visible scale on the heated surface of the tubes while a reduction of the overall heat transfer coefficient of approximately 20% could be observed. When operated as a fluidized bed heat exchanger over the same period of time, at the same temperature and with initially an identical solution, no scale was observed and the overall heat transfer coefficient kept its initial value.

Considering the importance of these phenomena on the development of the vertical flash evaporator it was decided to investigate fouling, scale formation and its related influences.

In order to analyse the complex system of scale formation and the non-scaling capabilities of the liquid-solid fluidized bed a distinction was made for the following parameters:

a. sludge and fouling influence on scaling
b. surface condition of the heat exchanger
c. momentum transfer of the fluidized particles.

As these parameters influence the scaling behaviour and the formation of adherent scale to the heat exchanger wall an analysis of the specific influence was made and described in the subchapters 3.5, 3.6 and 3.7. However, it was felt necessary first to describe the mechanism of crystallization.

- 3.6 -
3.4. Mechanism of crystallization.

The deposition of a solid crystalline phase from liquid can only occur if some degree of supersaturation has first been achieved in the system. The attainment of the supersaturated state is essential for any crystallization operation and the degree of supersaturation or deviation from the equilibrium saturated condition is the prime factor controlling the deposition process.

Any crystallization operation can be considered to comprise three basic steps:

1. achievement of supersaturation
2. formation of crystal nuclei
3. growth of the crystals.

The supersaturation of a system may be achieved by heating or is a result of a chemical reaction. The supersaturation has already been discussed in the introduction of this subchapter. The other two factors, nucleation and growth will be considered in this subchapter.

3.4.1. Nucleation.

The condition of supersaturation alone is not sufficient for a system to start crystallizing. Before crystals can grow there must exist in the solution or at a metal surface a number of minute solid bodies known as centres of crystallization or nuclei. Nucleation may occur spontaneously or it may be induced artificially; these two cases are frequently referred to as homogeneous and heterogeneous nucleation respectively.

It is not always possible, however, to decide whether a system has nucleated of its own accord of whether it has done so under the influence of some external stimulus.

Exactly how a crystal nucleus is formed within a homogeneous fluid system is not known with any degree of certainty.

The formation of crystal nuclei is a difficult process. The constituent molecules have not only to coagulate, resisting the tendency to redissolve, but they also have to become orientated into a fixed lattice.

The construction process, which occurs very rapidly, can only occur in local regions of high supersaturation and many of these sub-nuclei fail to achieve maturity; they simple redissolve because they are extremely unstable. If, however, the nucleus grows beyond a certain critical size
it becomes stable under the average conditions of supersaturation existing in the bulk of the fluid.

The reason for becoming stable beyond a certain critical size can be explained by the overall excess free energy relation as deduced by GIBBS [3.3].

The transition from the liquid to the solid state represents a stepwise decrease in the degree of molecular mobility and likewise a decrease in the free energy of the system.

The total quantity of work, \( w \), required to form a stable crystal nucleus is equal to the sum of the work required to form the surface, \( W_s \) (a positive quantity) and the work required to form the bulk of the particle, \( W_v \) (a negative quantity).

The free energy changes associated with the process of homogeneous nucleation may be considered as follows. The overall excess free energy \( \Delta G \), between a small solid particle of solute and the solute in solution is equal to the sum of the surface excess free energy, \( \Delta G_s \) (i.e. the excess free energy between the surface of the particle and the bulk of the particle) and the volume excess free energy, \( \Delta G_v \) (i.e. the excess free energy between a very large particle (\( r=\infty \)) and the solute in solution).

\( \Delta G_s \) is a positive quantity, the magnitude of which is proportional to \( r^2 \). In a supersaturated solution \( G_v \) is a negative quantity proportional to \( r^3 \). Thus:

\[
\Delta G = \Delta G_s + \Delta G_v = 4\pi r^2 \sigma + \frac{4}{3} \pi r^3 \Delta G_v
\]  

(3-8)

where:

\( \sigma \) = surface energy

\( \Delta G_v \) = free energy change of the transformation per unit volume.

The free energy of formation \( \Delta G \), passes through a maximum (figure 3-2).

**FIGURE 3-2.**

Free energy as a function of the crystal nucleus.
This maximum value, $\Delta G_{\text{crit}}$, corresponds to the critical nucleus $r_c$ and for a spherical cluster is obtained by maximising equation (3-8):

$$\frac{d\Delta G}{dr} = 8\pi r \sigma + 4\pi r^2 \Delta G_v = 0$$

Therefore:

$$r_c = -\frac{2\sigma}{\Delta G_v}$$

where $\Delta G_v$ is a negative quantity. From equation (3-8) and (3-10) we find:

$$\Delta G_{\text{crit}} = \frac{16\pi \sigma r^3}{3(\Delta G_v)^2} = \frac{4\pi \sigma r^2}{3}$$

The behaviour of a newly created crystalline lattice structure in a supersaturated solution depends on its size; it can either grow or redissolve, but the process which it undergoes should result in the decrease of free energy of the particle.

The critical size $r_c$, therefore, represents the minimum size of a stable nucleus. Particles smaller than $r_c$ will redissolve, because only in this way the particle can achieve a reduction in its free energy. Similarly, particles larger than $r_c$ will continue to grow.

The amount of energy $\Delta G_{\text{crit}}$, necessary to form a stable nucleus is found in the temperature rise of the liquid towards the heat exchanger wall compared with the bulk in first instance and secondly in the overall temperature rise of the liquid in longitudinal direction. It is not uncommon to find some particular spot on the heat exchanger wall acting as a crystallisation centre. The most reasonable explanation of this phenomenon is that minute cracks, manufacturing grooves and crevices in the surface retain tiny crystals which seed the system when it becomes supersaturated. It is possible of course, for a metal surface to be in a condition in which it acts as a catalyst for nucleation.

As the presence of a suitable foreign body or "sympathetic" surface can induce nucleation at degrees of supersaturation lower than those required for spontaneous nucleation, the overall free energy change associated with the formation of a critical nucleus under heterogeneous conditions, $\Delta G_{\text{crit}}^{1}$, must be less than the corresponding free energy change, $\Delta G_{\text{crit}}$, associated with homogeneous nucleation, i.e.:
\[ \Delta G^{1}_{\text{crit}} = \phi \Delta G_{\text{crit}} \quad (3-12) \]

where:

\[ \phi < 1 \]

3.4.2. Crystal growth.

As soon as stable nuclei, i.e. particles larger than the critical size, have been formed in a supersaturated system they begin to grow into crystals of visible size. Many attempts have been made to explain the mechanism and rate of crystal growth and these may be broadly classified under the three general headings of: "surface energy", "adsorption layer" and "diffusion" theories. The surface energy theories are based on the postulation of GIBBS [3.3] and CURIE [3.4] that the shape a growing crystal assumes, has a minimum surface energy.

More frequent applied is the theory of diffusion originated by NOYES and WHITNEY [3.5] and NERNST [3.6]. This diffusion theory presumes that the matter is deposited continuously on a crystal face at a rate proportional to the difference in concentration between the point of deposition and the bulk of the solution. The mathematical analysis is similar to the one used for diffusional and mass transfer processes.

VOLMER [3.7] suggested that crystal growth is a discontinuous process, taking place by adsorption layer by layer. Volmer's theory is based on thermodynamic reasoning. When units of the crystallizing substance arrive at the crystal face they are not immediately integrated into the lattice, but are free to migrate over the crystal face (surface diffusion).

Sparingly soluble substances as \( \text{CaCO}_3 \), \( \text{Mg(OH)}_2 \) and \( \text{CaSO}_4 \cdot \text{xH}_2\text{O} \) are characterized by growth kinetics dependent on the surface rather than dependence of volume diffusion.

Crystal growth for the sparingly soluble substances, subject of this study, is determined by the complex of diffusion to the surface, migration over the crystal face and integration at the surface.

If the liquid velocity is sufficient high, the overall growth rate would be determined by the rate of integration of the molecules into the crystal lattice. If the crystal is grown in stagnant solution then the rate of the diffusion step will be at a minimum. Overall growth rate has a complex dependence on temperature supersaturation, crystal size, crystal habit and turbulence in the system.
The presence of impurities in a system can have a profound effect on the growth of a crystal. Some impurities can suppress growth to a large extend; some may enhance growth. Some impurities can exert an influence at very low concentrations, less than 1 part per million, whereas others need to be present in fairly large amounts.

A general equation for crystallization based on the overall driving force can be written as:

$$\frac{dm}{dt} = K_G A (c-c^*)^n \quad (3-13)$$

where:

- $m$ = mass of solids deposited in time $t$
- $A$ = surface area of the crystal
- $K_G$ = overall crystal growth coefficient
- $n$ = overall growth rate order
- $c$ = solute concentrate in the solution
- $c^*$ = equilibrium saturation concentration.

The diffusion step is generally considered to be linearly dependent on the concentration driving force and reads:

$$\frac{dm}{dt} = k_d A (c-c_i) \frac{dc}{dx} \quad (3-14)$$

with:

- $k_d$ = coefficient of mass transfer by diffusion
- $c_i$ = solute concentration in the solution at the crystal solution interface.

The migration and integration at the surface of the crystal i.e. the surface reaction is expressed by:

$$\frac{dm}{dt} = k_r A (c_i-c^*)^z \quad (3-15)$$

with:

- $k_r$ = rate constant for surface reaction
- $z$ = surface reaction exponent.
Despite of an abundance of literature on the subject of crystallization, the relationship between the coefficients $K_G$, $k_d$, and $k_r$ remains obscure. With the influences of impurities, actual liquid velocity along the crystal (turbulence) it requires additional development of the crystallization models as well as concentrated experimental work to verify the models.
3.5. Sludge and fouling influence.

An analysis of scale, formed in a commercial Multi Stage Flash distillation unit, using energy dispersive spectroscopy indicated that the periphery of the scale contained the elements Ca and S [3.8]. Close to the wall a high concentration of other deposits and sludge formation was found. IZUMI et al. [3.9], [3.10], [3.11] concluded from their work that scale formation and deposition is largely influenced by a number of parameters as: fouling, liquid velocity and surface roughness. YORK and SCHORLE [3.12] also indicated that the nucleation site and the formed sludge or sediment plays an important role in the scale formation process.

It is usually assumed that the energy redistribution needed to form a crystal is reduced by the presence of any foreign solid, even if the solid is not of the same crystalline or chemical constitution.

**FIGURE 3-3.**
Illustration of excessive scale initiated on deposits and corrosion products.
Figure 3-3 shows a cross section of condenser tubes from a conventional flash evaporator after excessive scale formation. The author obtained the tubes for further analysis from an acid dosed conventional evaporator after re-tubing of the brine heater was necessary. Analysis of the scale revealed that the outer ring of scale contains a high percentage of corrosion products and sediment of a composition different from the scale components, also indicating that the actual scale nucleation was related to these foreign components. The importance of this subchapter is the understanding of the influence of sludge and fouling on the scale formation and the particular differences in this respect when applying a fluidized bed system. The moving particles of the liquid/solid fluidized bed will avoid the deposition of sludge and general fouling on the heat exchanger walls and consequently to some extend inhibit the formation of scale. It has been demonstrated by the author that a fluidized bed system exposed to a liquid which might cause fouling because of its suspended matter, results in the absence of fouling on the heat exchanger walls.

Figure 3-4 shows the test rig for fouling experiments with various suspended matter with and without a fluidized bed. In these experiments the maximum liquid temperature was kept well below the theoretical scaling temperature of the liquid in order to demonstrate the non-fouling capability of the fluidized bed system.

**FIGURE 3-4.**
Test rig for tube fouling experiments.
Table 3-2 summarizes the results obtained with these experiments, clearly
demonstrating the non-fouling capability of liquid-solid fluidized systems.

<table>
<thead>
<tr>
<th>testrun</th>
<th>type of suspended matter</th>
<th>approximate scaling temperature °C</th>
<th>minimum applied liquid temperature °C</th>
<th>steam condensation temperature °C</th>
<th>fluidized bed system</th>
<th>liquid velocity in heat exchanger tube in m/s</th>
<th>operation period in hours</th>
<th>results</th>
</tr>
</thead>
<tbody>
<tr>
<td>F - 4</td>
<td>clay</td>
<td>80</td>
<td>55</td>
<td>70 - 75</td>
<td>no</td>
<td>1.5</td>
<td>168</td>
<td>clearly visible sludge deposition</td>
</tr>
<tr>
<td>F - 5</td>
<td>clay</td>
<td>80</td>
<td>55</td>
<td>75</td>
<td>yes, lmm, c=0.8</td>
<td>0.14</td>
<td>168</td>
<td>clean, bright surface</td>
</tr>
<tr>
<td>F - 7</td>
<td>clay and sand</td>
<td>80</td>
<td>55</td>
<td>75</td>
<td>no</td>
<td>1.5</td>
<td>168</td>
<td>just visible sludge deposition</td>
</tr>
<tr>
<td>F - 8</td>
<td>clay and sand</td>
<td>80</td>
<td>55</td>
<td>75</td>
<td>yes, lmm, c=0.8</td>
<td>0.14</td>
<td>168</td>
<td>clean, bright surface</td>
</tr>
<tr>
<td>F - 11</td>
<td>clay</td>
<td>85</td>
<td>55</td>
<td>70</td>
<td>no</td>
<td>1.5</td>
<td>168</td>
<td>clearly visible sludge deposition</td>
</tr>
<tr>
<td>F - 12</td>
<td>clay</td>
<td>85</td>
<td>55</td>
<td>70</td>
<td>yes, lmm, c=0.8</td>
<td>0.09</td>
<td>168</td>
<td>clean, bright surface</td>
</tr>
</tbody>
</table>

**TABLE 3-2.**

Fouling experiments with and without a fluidized bed present.
3.6. **Surface condition.**

From the work of KARNAUKHOV et al. [3.13], SATO et al. [3.14] and YORK et al. [3.12] can be concluded that the condition of the surface has a significant influence on the formation of scale adherent to the heat exchanger wall. In this respect surface roughness, minor cracks and manufacturing grooves are to be distinguished in the surface condition. As a fluidized bed system has an influence on the surface condition it was necessary to investigate this matter.

It was decided to investigate this effect for different material compositions. In order to determine the influence of particles of a liquid-solid fluidization system on the surface roughness relatively simple test rigs were built.

The test rig as already shown in figure 3-4 served this purpose, as tests can be performed with different tube materials and fluidized beds. This test rig was also used for duration tests.

The figures 3-5 through 3-8 serve as an illustration for the heat exchanger tube material CuNi 90/10 and are obtained from Scanning Electron Microscopy (20 kV). The initial surface condition of the CuNi 90/10 test tube is shown in figure 3-5 and figures 3-6, 3-7 and 3-8 show the surface of the same test tube after 24, 72 and 336 hours of continuous operation with a fluidized bed system with glass particles. Diameter of the particles is 4 mm, density of the glass is 2420 kg per m³, average porosity of the fluidized bed is 0.8 and an average liquid temperature of 30 °C.
FIGURE 3-5.
Initial surface condition of CuNi 90/10 heat exchanger tube material (300x).

FIGURE 3-6.
Surface condition of CuNi 90/10 after 24 hours operation with 4 mm fluidized bed and porosity of 0.80 at a temperature of 30 °C (300x).
FIGURE 3-7.
Surface condition of CuNi 90/10 after 72 hours operation with 4 mm fluidized bed and porosity of 0.80 at a temperature of 30 °C (300x).

FIGURE 3-8.
Surface condition of CuNi 90/10 after 336 hours operation with 4 mm fluidized bed and porosity of 0.80 at a temperature of 30 °C (300x).
Table 3-3 gives the material specifications of the respective heat exchanger tubes as applied in these test and as they are commercially available.

<table>
<thead>
<tr>
<th>nominal composition in %</th>
<th>aluminium brass</th>
<th>cupronickel 90/10</th>
<th>cupronickel 70/30</th>
</tr>
</thead>
<tbody>
<tr>
<td>Cu</td>
<td>76</td>
<td>balance</td>
<td>balance</td>
</tr>
<tr>
<td>Al</td>
<td>2</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>Ni</td>
<td>-</td>
<td>10</td>
<td>30</td>
</tr>
<tr>
<td>Zn</td>
<td>balance</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>As</td>
<td>0.03</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>Mn</td>
<td>-</td>
<td>0.5</td>
<td>0.8</td>
</tr>
<tr>
<td>Fe</td>
<td>-</td>
<td>1.25</td>
<td>0.5</td>
</tr>
<tr>
<td>standard</td>
<td>ASTM B 111</td>
<td>ASTM B 111</td>
<td>ASTM B 111</td>
</tr>
<tr>
<td>tensile strength N/mm²</td>
<td>BS 1464-378-287</td>
<td>BS 378-2571</td>
<td>BS 1464-378-2571</td>
</tr>
<tr>
<td>specific weight kg/m³</td>
<td>355-410</td>
<td>285-335</td>
<td>355-430</td>
</tr>
<tr>
<td>thermal conductivity J/m²·s·°C·m</td>
<td>100.46</td>
<td>46.05</td>
<td>29.30</td>
</tr>
</tbody>
</table>

**TABLE 3-3.**

Material specification of three usual applied condenser tube materials for evaporator purposes.

Figure 3-9 elucidates the empirical relations found for the various tube materials, glass particles with an average density of 2340 kg/m³ and a porosity of 0.8 for the particle diameter and the operation time needed to reach a stabilization of the surface roughness.

**FIGURE 3-9.**

Operation time to reach stabilization in surface roughness in relation with particle diameter and for different materials.
Figures 3-10 and 3-11 represent the surface roughness, measured in micro-inch (ru), decrease with time in longitudinal and radial direction respectively for different particles sizes and for constant porosity.

**FIGURE 3-10.**
Surface roughness decrease with time in a radial direction on the tube surface.

**FIGURE 3-11.**
Surface roughness decrease with time in a longitudinal direction on the tube surface.
3.7. Particle velocity in liquid-solid fluidized bed.

The inhibition of scale adherent to the heat exchanger wall with application of a liquid-solid fluidized bed system is amongst others based on mechanical forces and momentum transfer. In this subchapter particle velocity model approach will be briefly reviewed as well as some experimental work in order to establish a basis for future investigations.

3.7.1. Particle velocity model approach.

The influence of the solid particle bouncing against the heat exchanger wall and rolling over the surface is evident in the fouling and scale inhibition mechanism. The inhibition of a layer of scale adherent to the heat exchanger wall will be influenced by the mechanisms of a fluidized bed particle colliding with a crystal nucleus. Based upon an existing cooperation it was decided that the Chemistry Department of the Delft University of Technology would further investigate momentum transfer and impact erosion of crystals by fluidized bed particles. The results of these investigations are published in the thesis of MEIJER [3.15].

3.7.2. Empirical relations.

BORDELT et al. [3.16] and [3.17] measured the mean transferred perpendicular kinetic energy of fluidized bed particles on a wall of a tube and found for the mean transferred perpendicular component of the kinetic energy of the particles:

$$ E = 9.13 \times 10^{-6} \left( \rho_s - \rho_l \right)^{0.38} \left( \frac{u_{r,s} d_p}{e} \right)^{0.66} \eta_1^{0.28} \tag{3-16} $$

Based upon similarities in the velocity distribution as expressed in the Maxwell-Boltzmann kinetic theory of gases the mean perpendicular component of the kinetic energy $E_x$ is obtained by:

$$ E_x = \frac{1}{2} m_p \tilde{v}_1^2 \tag{3-17} $$

with:

$$ \tilde{v}_1 = \text{mean impact velocity perpendicular to the wall} $$

$$ m_p = \text{solid particle mass}. $$
MEIJER [3.15] found the mean perpendicular kinetic energy calculated with equations (3-16) and (3-17) considerably deviating from observations. CARLOS and RICHARDSON [3.20] also concluded a necessity of modification of the Maxwell distribution to allow for an apparent anisotropy. As the model is derived for single fluidized particle movement in the vertical axis we have to establish the rate of anisotropy in the particle motion.

Here the author found the work of HANDLEY [3.18], HANRATTY [3.19] and CARLOS and RICHARDSON [3.20] of value to solve some of the problems. Hanratty relates the liquid turbulence to the radial mixing, expressed as the Peclet number, considering a fluidized bed as a physical extension of a fixed bed. The Peclet number is defined as:

\[
Pe = \frac{d_P u^2_{rms}}{\varepsilon E}
\]  

(3-18)

with \(E\) as the diffusion coefficient; the root mean square displacement in radial direction with time.

As the Peclet number should decrease continuously from the fixed state, Hanratty found a minimum for all observed fluidization systems applying on the average the same particle sizes and porosity as in the present work. The minimum exists for \(\varepsilon \approx 0.7\) and is explained by Hanratty as follows: the distance a fluid element must side step in order to pass around a fluidized bed particle increases with an increase of the porosity. This results in an increase of the turbulent diffusivity. At a porosity of approximately 0.70 fluid elements begin to flow past solid particles without the necessity at each passing to flow laterally in order to evade the particle. Beyond the critical porosity of 0.70 in more dilute fluidized beds the turbulence is generated by the particles and the turbulent diffusivity and is then a direct function of the decreasing particle population.

HANDLEY [3.18] demonstrated that the ratio of the vertical and horizontal particle turbulent velocity component for comparable systems is approximately constant at:

\[
\frac{\frac{\nu^2}{\nu}}{\frac{\nu^2}{h}} = 2.5
\]  

(3-19)
FIGURE 3-12.
Particle velocity components as a function of the fluidized bed porosity and for particle diameters of 2 and 3 mm.

Figure 3-12 shows the variation of the root mean square of the resultant turbulent particle velocity $\sqrt{v_{t}^2}$ and the vertical turbulent particle velocity component, $\sqrt{v_{y}^2}$, with voidage $\varepsilon$ based on the results of Handley and approximated for 2 and 3 mm particles with an average density of 2340 kg/m$^3$.

Handley also observed that the fluidized bed particle velocity components appear to exceed the turbulent fluid velocity components by a factor of about two and it therefore seems unlikely that the vertical particle motion a simple reflection is of the fluid turbulence in a uniform homogeneous field of flow. Handley's explanation is that a large proportion of the fluid is passing through the fluidized bed via a large number of small scale continuously fluctuating channels.

Particles may be elevated by these fluid streams until they drop out in regions of low fluid velocity and fall in the wake of other particles.
HANRATTY [3.19] and also CARLOS and RICHARDSON [3.20] investigated the relation between the tangential and radial components of the horizontal particle velocity. Although their results are only verified for porosities up to 0.70 it is assumed that over the entire range of porosity the components relate:

\[ v_h = \left( v_{\text{tan}}^2 + v_r^2 \right)^{\frac{1}{2}} \]  

\[ v_t = \left( v_h^2 + v_v^2 \right)^{\frac{1}{2}} \]  

(3-20)  

(3-21)

and as shown in figure 3-13 Carlos and Richardson found the best approximation of the velocity distribution in a probability distribution function in analogy with the Maxwell distribution for molecular velocities, but modified to allow for the existing anisotropy.

GORDON [3.21] and HANDLEY [3.18] from empirical data found the particle velocity to vary almost linear with the liquid velocities for porosities lower than 0.70.

**FIGURE 3-13.**

Particle velocity distribution as a function of the fluidized bed porosity.

CARLOS and RICHARDSON [3.20], however, found that for lower porosities as also can be seen from figure 3-13 the mean square of the particle velocities is linear dependent on the liquid velocity. Further it can be concluded that detail investigation beyond the scope of this work is felt necessary in order to derive more accurate equation for the particle movement over a sufficient large range of the porosity.

The momentum transfer from the particles to the wall with respect to the de-scaling capabilities therefore only can be given as tendencies because of the complex nature of the various mechanisms.
The relation between the superficial velocity of the fluidized bed system, i.e. the velocity of the liquid in the tube based on the empty cross section of the tube and the fluidized bed porosity are shown in figure 3-14. The calculation of the superficial velocities is based on the equations as given by Richardson and Zaki (Chapter 2.8).

Experimental results for the horizontal particle velocity as obtained by CARLOS and RICHARDSON [3.20] are in general agreement with the experimental results of HANDLEY [3.19] for the lower porosities of the bed. Experimental results of Handley for higher values of porosity show a sharp decline of particle velocity with increasing superficial velocity.

In order to establish data for the de-scaling capabilities of the liquid-solid fluidized bed over a range of the relevant parameters as found in the above analysis it was decided to conduct a series of de-scaling capability experiments. The empirical values to be obtained from these experiments will form the basis of a future fundamental approach in order to establish a comprehensive de-scaling model in relation with liquid fluidized bed application. Apart from fluid mechanics and specific particle velocity, attention will have to be paid to the relation between momentum transfer to the surrounding wall and the mechanical strength of scale crystals.

**FIGURE 3-14.**
Superficial velocity of fluidized bed system as a function of the porosity and for different particle sizes and liquid temperatures.
3.8. Scale control experiments.

Three types of experiments were carried out:

- with highly polluted surface water of the Schie canal
- with sea water on the island of Texel
- with artificial sea water.

3.8.1. Schie canal water

Experiments with water of the Schie canal give only qualitative information about the scale removing capability of the fluidized bed heat exchanger. These experiments are described in reference [3.2] and will be only briefly reviewed and commented upon.

![Diagram](image)

**FIGURE 3-15.**
Test module for the determination of the scale removing capacity of a fluidized bed.

Canal water was heated in a fluidized bed system as shown in figure 3-15. Chemical analysis showed that the scaling substance in the canal water was primarily calcium carbonate as a result of the decomposition of the carbonate ions at increasing water temperature in the test rig. It was proven that a flash evaporator with a fluidized bed heat exchanger is able to operate at a higher maximum temperature than a conventional flash evaporator, both operated without the dosing of chemicals to prevent scaling in the heat exchangers.
Table 3-4 gives an analysis of the canal water. The derivations by KLAREN [3.2] from these experiments for the maximum possible operation temperature, however, are based on a simple mechanical model neglecting the crystallization kinetics, the influence of aeration and the recirculation of the brine, the influence of foreign particles serving as nuclei and the state of the heat exchanger surface. The validity of the derivations is consequently highly questionable at least for those circumstances where recirculation of the liquid was applied.

Therefore the author decided to perform a different set of experiments.

3.8.2. Sea water.

As the principle application of the Multi Stage Flash/Fluidized Bed Evaporator (MSF/FBE) is in the desalination of sea water it was decided to move the test rig to a site with a sea water supply. The test rig was modified to a steam heated system in order to avoid the unfavourable influence of local hot spots caused by the electric heating system.

The modified test rig is shown in figure 3-16.

![Diagram of the modified test module for determination of scale removing capacity (steam heated).](image)
The main purpose of the experiments described in this subchapter 3.8.2. was to establish values for maximum operating temperature without an adhered scale layer on the heat exchanger walls. These experiments were carried out with values for the different parameters comparable with the "once through" mode of an MSF/FBE. The results therefore have to be categorized as qualitative overall results obtained from experiments and these experiments implicit take into account the various influences encountered like, some organic fouling, a certain tube material and roughness, degasing of the heated feed and for some experimental runs a certain recirculation of the liquid.

<table>
<thead>
<tr>
<th>Conductivity</th>
<th>[μS]</th>
<th>31000</th>
</tr>
</thead>
<tbody>
<tr>
<td>PH</td>
<td></td>
<td>8.0</td>
</tr>
<tr>
<td>TDS</td>
<td>ppm</td>
<td>25000</td>
</tr>
<tr>
<td>HCO₃⁻</td>
<td>ppm</td>
<td>140</td>
</tr>
<tr>
<td>Ca²⁺</td>
<td>ppm</td>
<td>325</td>
</tr>
<tr>
<td>Na⁺</td>
<td>ppm</td>
<td>7500</td>
</tr>
<tr>
<td>Mg²⁺</td>
<td>ppm</td>
<td>900</td>
</tr>
<tr>
<td>Cl⁻</td>
<td>ppm</td>
<td>13000</td>
</tr>
</tbody>
</table>

**TABLE 3-5.**

*Composition of sea water at the island of Texel.*

Sea water without any chemical treatment was heated and the experiments were performed with spherical glass particles of 2 and 3 mm diameter. As the composition of the feed, particularly the bicarbonate concentration and the acidity, directly influences the precipitation of calcium carbonates and also magnesium hydroxides, samples of the feed were analyzed regularly. Although the decomposition of bicarbonate ions at higher temperatures produces scale, this scale may not permanently adhere to the wall of the heat exchanger section as long as the particles of the fluidized bed are able to remove scale as fast as it precipitates directly on to wall. In that case the removing power of the particles meets the rate of precipitation at the wall.

Once the crystals are removed from the wall of the heat input sections by the removing power of the fluidized bed particles, these crystals do not redissolve because of their relative poor solubility and the small driving force for dissolution. These crystals will leave the system as suspended
particles with the discharge flow. During the experiments the brine temperature was raised very gradual to a maximum of 105 °C. With none of the test runs a scale adhered to the wall was found, while operating in the "once-through" mode, without recirculation.

<table>
<thead>
<tr>
<th>test run</th>
<th>feedwater temperature °C</th>
<th>maximum brine temperature °C</th>
<th>recirculation of the brine</th>
<th>particle diameter mm</th>
<th>average porosity input section</th>
<th>number of operation hours</th>
<th>visible scale at inspection</th>
</tr>
</thead>
<tbody>
<tr>
<td>2 - 16</td>
<td>14</td>
<td>60</td>
<td>0</td>
<td>2</td>
<td>0.8</td>
<td>24</td>
<td>none</td>
</tr>
<tr>
<td>2 - 18</td>
<td>13</td>
<td>65</td>
<td>0</td>
<td>2</td>
<td>0.8</td>
<td>24</td>
<td>none</td>
</tr>
<tr>
<td>2 - 21</td>
<td>14</td>
<td>70</td>
<td>0</td>
<td>2</td>
<td>0.8</td>
<td>20</td>
<td>none</td>
</tr>
<tr>
<td>2 - 22</td>
<td>14</td>
<td>75</td>
<td>0</td>
<td>2</td>
<td>0.8</td>
<td>26</td>
<td>none</td>
</tr>
<tr>
<td>2 - 23</td>
<td>15</td>
<td>80</td>
<td>0</td>
<td>2</td>
<td>0.8</td>
<td>24</td>
<td>none</td>
</tr>
<tr>
<td>2 - 25</td>
<td>16</td>
<td>85</td>
<td>0</td>
<td>2</td>
<td>0.8</td>
<td>36</td>
<td>none</td>
</tr>
<tr>
<td>2 - 26</td>
<td>16</td>
<td>90</td>
<td>0</td>
<td>2</td>
<td>0.8</td>
<td>36</td>
<td>none</td>
</tr>
<tr>
<td>2 - 28</td>
<td>15</td>
<td>95</td>
<td>0</td>
<td>2</td>
<td>0.8</td>
<td>48</td>
<td>none</td>
</tr>
<tr>
<td>2 - 29</td>
<td>16</td>
<td>100</td>
<td>0</td>
<td>2</td>
<td>0.7</td>
<td>72</td>
<td>none</td>
</tr>
<tr>
<td>2 - 31</td>
<td>15</td>
<td>100</td>
<td>0</td>
<td>2</td>
<td>0.8</td>
<td>96</td>
<td>none</td>
</tr>
<tr>
<td>2 - 33</td>
<td>16</td>
<td>105</td>
<td>0</td>
<td>2</td>
<td>0.7</td>
<td>72</td>
<td>white appearance in minor crevices</td>
</tr>
<tr>
<td>2 - 34</td>
<td>15</td>
<td>105</td>
<td>0</td>
<td>2</td>
<td>0.8</td>
<td>96</td>
<td>none</td>
</tr>
</tbody>
</table>

**TABLE 3-6.**

Results of scale-experiments with a particle diameter of 2 mm.

Tables 3-6 and 3-7 present a condensation of the described test runs. In order to be able to continue this empirical determination of the maximum operating temperature without scale adhered to the heat exchanger surfaces, a new test rig had to be built. It was decided to include also features for simulation of the recirculation of brine also with respect to release of carbon dioxide from decomposing bicarbonates.

<table>
<thead>
<tr>
<th>test run</th>
<th>feedwater temperature °C</th>
<th>maximum brine temperature °C</th>
<th>recirculation of the brine</th>
<th>particle diameter mm</th>
<th>average porosity input section</th>
<th>number of operation hours</th>
<th>visible scale at inspection</th>
</tr>
</thead>
<tbody>
<tr>
<td>3 - 1</td>
<td>12</td>
<td>60</td>
<td>0</td>
<td>3</td>
<td>0.8</td>
<td>24</td>
<td>none</td>
</tr>
<tr>
<td>3 - 3</td>
<td>12</td>
<td>70</td>
<td>0</td>
<td>3</td>
<td>0.8</td>
<td>24</td>
<td>none</td>
</tr>
<tr>
<td>3 - 4</td>
<td>13</td>
<td>80</td>
<td>0</td>
<td>3</td>
<td>0.8</td>
<td>20</td>
<td>none</td>
</tr>
<tr>
<td>3 - 6</td>
<td>12</td>
<td>85</td>
<td>0</td>
<td>3</td>
<td>0.8</td>
<td>28</td>
<td>none</td>
</tr>
<tr>
<td>3 - 7</td>
<td>12</td>
<td>90</td>
<td>0</td>
<td>3</td>
<td>0.8</td>
<td>36</td>
<td>none</td>
</tr>
<tr>
<td>3 - 8</td>
<td>13</td>
<td>95</td>
<td>0</td>
<td>3</td>
<td>0.7</td>
<td>48</td>
<td>none</td>
</tr>
<tr>
<td>3 - 10</td>
<td>13</td>
<td>95</td>
<td>0</td>
<td>3</td>
<td>0.8</td>
<td>48</td>
<td>none</td>
</tr>
<tr>
<td>3 - 11</td>
<td>13</td>
<td>100</td>
<td>0</td>
<td>3</td>
<td>0.7</td>
<td>48</td>
<td>none</td>
</tr>
<tr>
<td>3 - 12</td>
<td>13</td>
<td>100</td>
<td>0</td>
<td>3</td>
<td>0.8</td>
<td>72</td>
<td>none</td>
</tr>
<tr>
<td>3 - 14</td>
<td>14</td>
<td>105</td>
<td>0</td>
<td>3</td>
<td>0.7</td>
<td>48</td>
<td>none</td>
</tr>
<tr>
<td>3 - 15</td>
<td>13</td>
<td>105</td>
<td>0</td>
<td>3</td>
<td>0.8</td>
<td>96</td>
<td>none</td>
</tr>
</tbody>
</table>

**TABLE 3-7.**

Results of scale-experiments with a particle diameter of 3 mm.
Figure 3-17 shows the flow scheme of test rig. This experimental fluidized bed heat exchanger consists of two fluidized bed heat exchanger tubes in series and is designed in such a way that the heat flux at 120 °C top temperature is comparable with current MSF/FBE design heat flux values. In this system a flash tank is accommodated for partial evaporation of the heated fluid. This is to simulate a multi-stage flash system with a decarbonator for the removal of the carbondioxide from decomposition of bicarbonates. With this test module operation with the same carbondioxide equilibrium as in a multi-stage flash evaporator with recirculation is possible.

Figure 3-18 elucidates the results of the scale experiments while table 3-8 condenses the specific experiments.

---

**FIGURE 3-17.** Modified test module for determination of maximum temperature without scale formation and with the possibility for recirculation of brine.

**FIGURE 3-18.** Condensation of results obtained with the scale experiments.
<table>
<thead>
<tr>
<th>test run</th>
<th>feedwater temperature °C</th>
<th>maximum brine temperature °C</th>
<th>recirculation of the brine</th>
<th>particle diameter mm</th>
<th>average porosity input section</th>
<th>number of operation hours</th>
<th>visible scale at inspection</th>
</tr>
</thead>
<tbody>
<tr>
<td>4 - 5</td>
<td>17</td>
<td>105</td>
<td>0</td>
<td>3</td>
<td>0.8</td>
<td>12</td>
<td>none</td>
</tr>
<tr>
<td>4 - 7</td>
<td>16</td>
<td>115</td>
<td>0</td>
<td>3</td>
<td>0.8</td>
<td>12</td>
<td>none *)</td>
</tr>
<tr>
<td>4 - 9</td>
<td>17</td>
<td>125</td>
<td>0</td>
<td>3</td>
<td>0.8</td>
<td>12</td>
<td>none *)</td>
</tr>
<tr>
<td>4 - 10</td>
<td>17</td>
<td>135</td>
<td>0</td>
<td>3</td>
<td>0.8</td>
<td>24</td>
<td>none *)</td>
</tr>
<tr>
<td>4 - 11</td>
<td>16</td>
<td>140</td>
<td>0</td>
<td>3</td>
<td>0.8</td>
<td>48</td>
<td>just visible</td>
</tr>
<tr>
<td>4 - 12</td>
<td>16</td>
<td>145</td>
<td>0</td>
<td>3</td>
<td>0.8</td>
<td>48</td>
<td>just visible</td>
</tr>
<tr>
<td>4 - 14</td>
<td>16</td>
<td>150</td>
<td>0</td>
<td>3</td>
<td>0.8</td>
<td>72</td>
<td>thin layer &lt; 0.1 mm</td>
</tr>
<tr>
<td>5 - 2</td>
<td>17</td>
<td>105</td>
<td>0</td>
<td>2</td>
<td>0.8</td>
<td>12</td>
<td>none *)</td>
</tr>
<tr>
<td>5 - 3</td>
<td>17</td>
<td>115</td>
<td>0</td>
<td>2</td>
<td>0.8</td>
<td>12</td>
<td>none *)</td>
</tr>
<tr>
<td>5 - 4</td>
<td>17</td>
<td>125</td>
<td>0</td>
<td>2</td>
<td>0.8</td>
<td>24</td>
<td>none *)</td>
</tr>
<tr>
<td>5 - 5</td>
<td>17</td>
<td>130</td>
<td>0</td>
<td>2</td>
<td>0.8</td>
<td>24</td>
<td>none *)</td>
</tr>
<tr>
<td>5 - 6</td>
<td>16</td>
<td>135</td>
<td>0</td>
<td>2</td>
<td>0.8</td>
<td>48</td>
<td>none *)</td>
</tr>
<tr>
<td>5 - 7</td>
<td>16</td>
<td>140</td>
<td>0</td>
<td>2</td>
<td>0.8</td>
<td>48</td>
<td>just visible</td>
</tr>
<tr>
<td>5 - 9</td>
<td>16</td>
<td>145</td>
<td>0</td>
<td>2</td>
<td>0.8</td>
<td>72</td>
<td>thin layer &lt; 0.1 mm</td>
</tr>
<tr>
<td>6 - 3</td>
<td>17</td>
<td>125</td>
<td>30</td>
<td>3</td>
<td>0.8</td>
<td>24</td>
<td>none *)</td>
</tr>
<tr>
<td>6 - 4</td>
<td>17</td>
<td>130</td>
<td>30</td>
<td>3</td>
<td>0.8</td>
<td>48</td>
<td>just visible</td>
</tr>
<tr>
<td>6 - 5</td>
<td>16</td>
<td>135</td>
<td>30</td>
<td>3</td>
<td>0.8</td>
<td>72</td>
<td>thin layer &lt; 0.1 mm</td>
</tr>
<tr>
<td>7 - 1</td>
<td>16</td>
<td>125</td>
<td>30</td>
<td>2</td>
<td>0.8</td>
<td>48</td>
<td>none *)</td>
</tr>
<tr>
<td>7 - 2</td>
<td>16</td>
<td>130</td>
<td>30</td>
<td>2</td>
<td>0.8</td>
<td>72</td>
<td>thin layer &lt; 0.1 mm</td>
</tr>
</tbody>
</table>

*) Except for in small overview.

TABLE 3-8.

Results of scale experiments at elevated temperatures, with different particle sizes and recirculation option.

If at an inspection a visible scale appeared on the heat exchanger surface, the surface was cleaned by operating at ambient temperature for sufficient time and with the fluidized bed present over the entire length of the heat exchanger. The cleaning was usually performed in less than six hours of operation.

3.8.3. Artificial sea water calcium sulphate experiments.

The experimental results described in subchapter 3.8.1 and 3.8.2 confirmed that these types of fluidized beds prevent the adherence of calcium-carbonate and magnesium hydroxide scales on the heat exchanger surfaces when applied for canal water with the indicated composition and for sea water at the island of Texel. All experiments were carried out using feedwater without any chemical treatment.

From the above mentioned experiments it seemed likely that the limiting factor, which determines the maximum operating temperature without adhered scale on the heat exchanger surface, depends on the development of calcium sulphate scale.

To investigate the influence of the fluidized bed on the deposition of calcium sulphate and to determine the maximum operating temperature a new test module was built. The design of the test module aimed to perform...
experiments to determine maximum operating temperature with values for liquid velocities, fluidized bed porosities, particle size, tube length, tube diameter and heat flux, comparable with present commercial MSF/FBE design [3.22].

Since the interest was mainly focussed on the formation of calcium sulphate scale the experiments were performed with a liquid consisting of calcium sulphate, sodium and chloride ions in water at the same ratio as in standard sea water. In order to establish the most valuable results concentration recirculation, aeration and fluidized bed parameters in addition to the temperature can be varied.

In addition to these parameters measures were taken once scale adhered to the wall to examine the modification of the deposited scale as well as the habit of the crystals situated at the heat transfer surface or at a certain distance from the surface.

**FIGURE 3-19.**
Test module for artificial sea water experiments and aimed at calcium sulphate deposition.
heat exchanger type  | concentric, countercurrent
length             | 4.5 m
tube size         | 15.88 mm O.D.
wall thickness    | 1.2 mm
tube material     | Yorcalbro (76% Cu, 2% Al, balance Zn)
particle diameter | 2 resp. 3 mm
particle density  | 2580 kg/m³
porosity (average)| 0.80
superficial velocity | approximate 0.11 resp. 0.13 m/s
heating fluid     | hot oil
hydrocyclone performance | 12 μm (50% catchment)
cartridge filter  | 10 μm
dissolving tanks  | 275.10⁻³ m³ each
surface roughness Ra | 0.11 μm

**TABLE 3-9.**

Basic data of test module for calcium sulphate experiments.

Figure 3-19 shows the test module while table 3-9 outlines the basic features of the test module. Under scaling conditions, nuclei will be formed in the brine, due to the fracturing of the crystals by the glass particles. Removal of the nuclei is possible by the shown combination of hydrocyclone, three dissolving tanks and a filter.

To dissolve the crystals which have passed the hydrocyclone three tanks in series have been installed. Reference [3.23] elucidates the residence time and the dissolution calculations for small particles. Gases dissolved in the brine will be removed by the deaerator, when in operation. However, the same equipment part can also be used to aerate the circulating brine.

**3.8.3.1. Experiments.**

The brine has been prepared by dissolving NaCl, CaCl₂·6H₂O and Na₂SO₄·10H₂O in water, keeping the ratio of calcium and sulphate ions exactly the same as in standard sea water, denoted as standard sea water by the Hydrographic Laboratories of Copenhagen, Denmark [3.1]. It is obvious that dealing with only three salts and restricted by a fixed \( \text{Ca}^{2+}/\text{SO}^{\text{2-}} \) ratio, the chloride ion concentration becomes about 12% lower and the sodium ion concentration about 12% higher than in standard sea water.
If the calcium and the sulphate molar ion concentration are equal to those in standard sea water, the molar concentration factor is defined to be one, and the brine is composed of:

- 2.4091 g/l CaCl₂ 6H₂O (99.0% pure)
- 9.3330 g/l Na₂SO₄ 10H₂O (99.5% pure)
- 29.1925 g/l NaCl (99.7% pure)

The pH of the brine for all experiments was equal to seven. So apart from the possibility to change the concentration factor of the brine and the diameter of the glass particles, the test conditions can be varied in four ways:

- a. exclusion of nuclei and degasing of the brine
- b. recirculation of nuclei and degasing of the brine
- c. exclusion of nuclei and aeration of the brine
- d. recirculation of nuclei and aeration of the brine.

During the experiments the temperature of the brine leaving the heat exchanger tube is kept at a constant level and a few degrees below the expected maximum operating temperature. If no scaling is observed within three hours, the temperature is raised for about two degrees centigrade and kept constant again.

If scale formation occurs it is deposited at the upper part of the heat exchanger tube. The scale can be detected due to the fact that the glass particles, visible in the lower part of the glass sight section, slowly lose their transparency.

In case scale deposition is observed the temperature is maintained at that level for about six hours. In addition the brine concentration factor is checked by EDTA titration of the calcium ions. After finishing the experiments the heat exchanger tube is emptied within half an hour and the deposited scale layer at the inside of the tube is inspected.

Scale samples are taken for examination by means of X-ray diffraction and infrared spectroscopy. In some cases the scale layer is examined in detail by Scanning Electron Microscopy (SEM). For this purpose a small piece of the upper part of the heat exchanger tube, covered with a thin layer of scale is removed. Firstly SEM pictures (15 kV) are taken of the surface layer of the scale. To examine the scale deposited deeper in the layer and especially at the scale-metal interface the following method, developed by VAN WIJK [3.24 ] has been chosen to disclose these surfaces.
A two component resin is applied as a liquid on the scale layer. Separating the hardened resin from the heat exchanger tube the fracture will appear at the weakest spot which is at the scale-metal interface. Both the metal surface with the scale remnants and the resin with the attached layer of scale can be studied now.

3.8.3.2. Results and discussion.

Seven runs of experiments were performed without recirculation of nuclei and with degasing of the brine. In four of these experiments 2 mm glass particles were used, while the remaining three had 1 mm glass particles. Three runs of experiments were performed for indication of the influence of recirculation of nuclei and of aeration of the brine. It appeared that in all cases except one the deposited scale consisted of calcium sulphate anhydrite. Only once calcium sulphate hemihydrate was deposited. The method outlined by MARSHALL and SLUSHER [3.25] is applied to calculate the solubility lines of anhydrite and hemihydrate for the applied brine at concentration factors of interest in the experiments.

**FIGURE 3-20.**

Calculated solubility curves and results of experiments with a fluidized bed, clearly showing the ability to pass the hemihydrate line with application of fluidized beds.

Figure 3-20 shows the calculated solubility lines, together with the experimentally determined maximum operating temperatures without scale adherence of the heat exchanger tube with fluidized bed.

It is noticeable that scale deposition only occurs when the solubility line of the meta-stable hemihydrate is passed, although the brine is highly supersaturated relative to the anhydrite phase. Apparently anhydrite nucleates only slowly but develops rather fast if it can grow epitaxially on hemihydrate nuclei. So anhydrite crystals were developed when the
hemihydrate line was passed instead of hemihydrate crystals, since the
supersaturation for hemihydrate is very low in comparison with the one
for anhydrite. Only when the hemihydrate line is passed largely the super-
saturation and hence the growth rate for hemihydrate is large enough and
this fase will be formed instead of anhydrite.
How far the hemihydrate line can be passed without deposition of a
detectable scale layer using a fluidized bed depends among other things on
the temperature. The rate of nucleation and the growth rate increase as
the temperature is raised, whereas the polishing action of the fluidized
bed remains about the same. Therefore it is not possible to pass the
hemihydrate solubility line to the same extent at higher temperatures as
at lower temperatures.
Figure 3-20 shows that 2 mm glass particles inhibit scale deposition
more effectively than 1 mm glass particles, resulting in a top temperature
rise of 5 - 6 °C. Recirculation of nuclei lowers the maximum temperature
without scale adherence by about 8 °C.
In these experiments no nuclei are added to the brine. There are, however,
always nuclei created as soon as the solubility line for hemihydrate is
passed. These nuclei can be caught in micro cracks, grooves, corrosion
pits and crevices in the metal surface of the tube and have always a
prolonged residence time in the laminar boundary layer, where they can
help to start the deposition of a scale layer.
Aeration of the brine lowers the maximum temperature by about 6 °C, due to
bubble formation of the released dissolved gases. Since the solubility of
most gases decreases with higher temperature, bubbles tend to form when
the temperature is raised. These bubbles are formed preferentially at
sites such as scratches, the edges of corroding pits, crevices, oxide
films, etc., on heat transfer surfaces.
When bubbles are formed in a slightly subsaturated brine, the brine becomes
supersaturated at the gas-liquid-solid interfaces, resulting in scale
formation. If the bulk of the brine is subsaturated the deposited scale
will dissolve as the grown bubble is removed from the heat exchanger
surface. At higher temperatures, however, the dissolution rate of the scale
is much lower due to the decreased subsaturation, so new bubbles are formed
before the old deposit has been dissolved. Moreover, the new bubbles are
formed preferentially on the old scale deposits, which act as nucleation
sites for the bubbles and the bubble formation rate increases at higher
temperature.
Thirdly, during their stay at the heat transfer surface bubbles obstruct heat transfer, thus creating localized micro hot spots. These spots cause local supersaturation and as a result deposition of scale.

Using a fluidized bed heat exchanger the bubble growth effect of aeration is diminished to a large extent because the lifetime of the bubbles at the heat exchanger surface is reduced drastically. This results in only slightly reduced maximum operating temperature without scale adherence to the heat transfer surface.

Combination of recirculation of nuclei and aeration of the brine lowers the maximum operating temperature about 10 °C only, which is slightly lower than the maximum operating temperature in case of the recirculation of nuclei only.

Most likely, the "driving force" dominating the growth of the crystal layer is the force due to the supersaturation of anhydrite instead of the one for hemihydrate, since in both cases the limitations imposed by nucleation are no longer applicable.

Both recirculation of nuclei and aeration of the brine have a noticeable effect on the crystal size of the deposited scale. Each parameter reduces this size largely and to about the same degree, while combination of both parameters clearly leads to the smallest, crystal size. This could be expected because when only a few nuclei are present, these nuclei have the opportunity to grow further. When more nuclei are present, due to recirculation or aeration, the nuclei will not become so large. When many crystals are present, due to recirculation and aeration, the crystal size will be the smallest.

The very poor crystal habit of the first crystal layer situated closely to the wall indicates the existence of a higher supersaturation during the first nucleation, compared to the supersaturation needed for the further growth of the crystals. As soon as the first crystal layer is deposited, the habit is improved greatly and well shaped crystals are formed as can be seen on figure 3-21 (anhydrite) and figure 3-22 (hemihydrate).

During this further growth the crystals are situated with their length direction perpendicular to the surface but fracture due to collision with the glass particles. As a result most crystals are situated parallel to the metal surface as shown in figure 3-23.
FIGURE 3.21.
Scanning Electron Microscopy (SEM) picture of calcium sulphate-anhydrite scale. Well shaped crystal on top of a first layer of crystals with a poor crystal habit.
FIGURE 3-22.

SEM picture of calcium sulphate-hemihydrate scale. This hemihydrate scale also shows a well shaped crystal habit almost directly on the surface of the heat exchanger tube.
FIGURE 3-23.

SEM picture crystal habit further removed from the heat exchanger surface. The initially perpendicular growth direction has been fractured by the collision with glass particles of the fluidized bed.
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CHAPTER 4  FLASH STAGE GEOMETRY

Figure 4-0 presents a visual aid for the contents of this chapter of flash stage geometry aspects.

4. FLASH STAGE GEOMETRY

4.1. Introduction

4.2. Entrainment

4.3. Equilibration

Literature survey 4.3.1.
Equilibration approach model 4.3.2.

4.4. Experimental apparatus

Description of the test module 4.4.1.
Operation of the test module 4.4.2.
Calculation procedure 4.4.3.
Pressure measurements 4.4.4.
Temperature measurements 4.4.5.
Salt concentration measurements 4.4.6.
Brine level measurements 4.4.7.
Mass flow measurements 4.4.8.

4.5. Experimental results

Introductory remarks 4.5.1.
Entrainment 4.5.2.
Basic geometry A
Basic geometry B
Basic geometry C
Equilibration 4.5.3.

4.6. Some design criteria and concluding remarks

Entrainment separation 4.6.1.
Temperature loss 4.6.2.
Equilibration 4.6.3.
4.1. Introduction.

In order to develop optimum stage geometries for various types of Multi-Stage Flash/Fluidized Bed Evaporators it is of importance to understand the vapour release phenomena and the allied entrainment characteristics of the brine. Entrainment is understood to be the disengaging of brine droplets carried away with the vapour flow before the vapour is condensed. As virtually no relevant literature is available describing these phenomena under comparable circumstances this chapter reviews the theoretical and empirical studies undertaken aiming for a better knowledge of the mentioned phenomena for Multi-Stage Flash/Fluidized Bed Evaporators and development of tools for design of flash stages for this type of evaporator. Desirable, a stage should obtain adiabatically an equilibrium between all of the brine discharged to the next stage and all of the vapour discharged to the condenser section of this stage. The latent heat of the vapour should equal the change in sensible heat of the brine in cooling through the flash temperature range and there should be no mist or droplets in the vapour stream.

For any possible linear brine velocity and any practical width of the stream in the conventional horizontal MSF arrangement the brine pool will have a considerable depth and consequently agitation and turbulence of the whole depth of the brine stream is required to allow a maximum of approach to equilibrium. Bubbles of vapour are released with violence from the turbulent surface carrying droplets of entrained brine. To remove these droplets, carrying salt water to the distillate, wire mesh or other demisters must be used.

FIGURE 4-1.
Pool boiling system in a conventional MSF stage releasing secondary droplets when vapour breaks through the surface.
Vapour release in an MSF/FBE is different compared to the release of vapour in a conventional MSF. While the boiling process in a conventional MSF can be described as a pool or nucleate boiling system, the boiling process of an MSF/FBE can be characterized as a vapour space boiling system. Figure 4-1 visualizes the boiling process in a conventional MSF stage while figure 4-2 clarifies the boiling in an MSF/FBE stage.

**FIGURE 4-2.**
Vapour space boiling system in an MSF/FBE stage with downward oriented droplets.

In the MSF/FBE the brine passes through slits from higher stages to lower ones after which it diverts and is intercepted by the wall at the far end of the stage or by an installed diversion plate. Because of the metastable state of the brine when entering the lower stage, the brine is torn apart and passes through the upper part of the stage as thin film liquid filaments and relatively large droplets. Afterwards it flows down as a turbulent liquid film.

Evaporation and condensation at the interface of steam and water are reciprocal phenomena with very large numbers of water molecules and large masses of water always passing from either phase to the other in a dynamic equilibrium. The difference of the number in molecules which evaporates over those that condense is small compared to the total number changing phase in each direction. However, the nett result of molecules that evaporate is large for an extremely small difference of the temperature on each side of the interface. This temperature difference amounts only hundreds of a degree for any rate of evaporation as observed by OTHMER [4.1].

For efficient evaporation, the superheated brine should pass through the appropriate part of the stage in intimate contact with the vapours formed under near-equilibrium conditions.

In the MSF/FBE stage breakdown of the brine into thin filaments and the subsequent formation of turbulent liquid both contribute to a high interface mass and heat transfer which causes a good approach to equilibrium in a relatively short flash length and in a relatively small stage volume.
4.2. Entrainment.

Literature on entrainment and its separation for systems where super heated liquid is flashed in the vapour space is very limited. ROE and OTTHMER [4.2] and ALI EL SAI [4.3] claim that flashing processes in a vapour space boiling system result in a reduced pollution of the product water by entrainment of brine droplets. However, these references give no details on the stage geometry and the parameters of influence. It is possible to obtain some information from studies carried out in relation with Multi-Stage Evaporation. TOYAMA [4.4] and TOYAMA et al. [4.5] observed the entrainment and for a conventional type of evaporator stage with a given geometry they describe the particle motion. TOYAMA [4.4] derived a simplified model where the fractional separation efficiency controlled by the particle motion is a function of a gravitational settling parameter and an inertia parameter. For a specific set of experiments at approximately 65 °C and a temperature drop of 1.9 °C, a concentration of mist in the order of 0.03 kg/m³ was observed while the particle size was concluded with 95% of the total entrained mist weight smaller than 200 μm and approximately 50% of all mist had a particle size between 80 and 120 μm. The number of experiments as well as the range of temperature and temperature drop of the experiments was far too small for a comparison with the simplified model. GOLUB and ILUSCHENKO [4.6] investigated entrainment in a flash evaporator of the conventional MSF type where they defined the entrainment coefficient as the quotient of the salt concentration in the distillate and in the initial solution. Variation of the entrainment coefficient ω with the initial super heat ΔTᵦ is described in logarithmic coordinates by a relation of ω ~ ΔTᵦⁿ.

Entrainment was found slightly depending on boiling temperature and very sharply on super heat of the liquid. The specific flow rate was found to effect the entrainment to a much lesser extent than the super heat. Finally the entrainment coefficient ω is described as:

\[
ω₀ = C \frac{h_{lg}}{c_p L} \left( \frac{δ_{ob}}{δ_0} \right)^2 \ e^{-0.23 \frac{H}{D}}
\]

(4-1)
with:

\[ C = \text{constant} \]
\[ h_{lg} = \text{latent heat of vaporization} \]
\[ c_p = \text{specific heat of the liquid} \]
\[ \Delta T_o = \text{initial super heat of the liquid} \]
\[ \delta_{ob} = \text{observed particle diameter} \]
\[ \delta_a = \text{average particle diameter} \]
\[ H = \text{evaporator stage height} \]
\[ D = \text{equivalent hydraulic diameter of evaporator stage} \]

The above relation (4-1) indicates that the entrainment for a conventional evaporator stage is related to the temperature drop from one stage to the next stage and to the dimensions of the evaporator stage. As the entrainment behaviour and the entrainment separation mechanisms in the reviewed literature differ in the case of MSF/FBE processes and because of lack of solid background for the existing model approaches it was decided by the author not to use the above mentioned models. As the entrainment and entrainment separation is strongly influenced by the stage geometry it was decided to follow an experimental approach for the entrainment itself and entrainment separation behaviour. Experimental results and design criteria are reported in subchapters 4.5 and 4.6 respectively.
4.3. **Equilibration.**

The parameters effecting the equilibration achieved in a flash stage are the object of investigations with the aim of improving the design of stages and orifices.

4.3.1. **Literature survey.**

Publications of studies on conventional horizontal evaporators, related to the investigation of equilibration, have followed two lines. Firstly: the overall behaviour of stages has been studied, resulting in a correlation for approach to equilibrium at the exit of the respective stage in terms of the independent variables, flow rate, flow dept, stage length, initial super heat and saturation condition; RICHARDSON and WESTGARTH [4.7], WILLIAMSON and HEFLER [4.1], LIOR [4.9] and EL-DESSOUKY and MAYINGER [4.10] followed this approach. Secondly: internal details, temperature and velocity distributions have been measured and the results compared with models for submerged jets. This detail-approach was followed by COOGAN and BREWSTER [4.11], COURY, DERONZIER and HUYGHE [4.12], PORTEOUS and MUNCASTER [4.13] and TORMANTSEV, ILUSCHENKO and CHERNOZUBOV [4.14].

**COOGAN and BREWSTER [4.11].**

The authors derived an analytical model of a flash evaporator stage. The model is based on a submerged jet at the entrance, consisting of a wall jet near the bottom and an expanding jet above. They assumed that the evaporation mostly occurs near the free surface. Consequently the basic equations are derived on the assumption of absence of evaporation in the liquid body and amended to include bulk evaporation effects. The model is based on the mechanism of turbulent thermal conductivity. This model is not used in this thesis as the coefficient of turbulent thermal conductivity is experimentally determined on only one geometry of the stage entrance design and one complexion of the jet, whereas the geometry of the stage entrance differs from the present study.

**COURY, DERONZIER and HUYGHE [4.12].**

The authors describe the model along the same lines as COOGAN and BREWSTER [4.11] applying the turbulent thermal conductivity in the general energy equation but, however, explain the importance of vapour bubbles in the interstage orifice.
Finally the equilibration fraction is expressed as a function of the mass flow, the length of the stage, the back flow on top of the submerged jet and the turbulent thermal conductivity. The derivation of these authors, however, is not applicable to the boiling process in the vertical flash stage as the equations have implicit geometry features.

**POURTOUS and MUNCASTER [4.13].**

The authors developed a model that is based on the theory of equilibration during evaporation from a plane interface in open channel flow. They considered a fluid entering a duct in which corresponding to the pressure $p$ the saturation temperature is $T_{v_{sat}}$ and assumed the convective heat transfer to some area $A$, representing the total interface surface per kg of fluid. The temperature of the vapour in the bubble is taken as constant; the saturation value $T_{v_{sat}}$. The driving force for evaporation across an element of the stream is taken as $(T_b - T_{v_{sat}})$.

The rate of change of the brine temperature with time is:

$$c_{pl} \frac{dT_b}{dt} = -hA \left( T_b - T_{v_{sat}} \right)$$

or:

$$\frac{dT_b}{T_b - T_{v_{sat}}} = - \frac{hA}{c_{pl}} \frac{dt}{dt}$$

(4-3)

with:

$c_{pl}$ = specific heat capacity of the liquid
$h$ = heat transfer coefficient, which embodies a term for the internal convection in the fluid and an interface coefficient for evaporation
$T_b$ = brine temperature
$A$ = total interfacial surface per unit mass of liquid

The fraction of equilibration is defined as:

$$\beta = \frac{T_{b_{en}} - T_b}{T_{b_{en}} - T_{v_{sat}}}$$

(4-4)
with:

\[ T_{b\text{en}} \] = brine temperature at stage entrance  
\[ T_b \] = brine temperature

The time span follows from:

\[ dt = \frac{dL}{\bar{v}} \]  \hspace{1cm} (4-5)

with:

\[ L \] = flash stage length, where brine temperature \( T_b \) is reached  
\[ \bar{v} \] = mean bulk fluid velocity

Substitution of equations (4-4) and (4-5) for constant \( T_{v\text{sat}} \) in equation (4-3) results in:

\[ \frac{d\beta}{1 - \beta} = \frac{h A}{c_p \bar{v}} dL \]  \hspace{1cm} (4-6)

The initial bubble population is defined by the authors as \( N \) per kg of fluid. Flashing down the stage, bubbles will escape through the free surface.

The authors define the decrease of bubble population as:

\[ N = \frac{a}{L} \]  \hspace{1cm} (4-7)

with:

\( a \) = nucleation constant

The value of the nucleation constant will depend on the stage temperature, the initial superheat and the properties of the liquid.

The interface surface of the bubbles per kg of fluid under the assumption of spherical bubbles with a mean radius amounts to:

\[ A = 4\pi R_L^2 N \]  \hspace{1cm} (4-8)
For the growth of a vapour bubble in a superheated liquid the authors have used the equation derived by FOSTER and ZUBER [4.19]. This bubble growth relation reads:

\[ R_b^2 = \bar{c}^2 t = \frac{\bar{c}^2 L}{v} \]  

(4-9)

With the bubble growth coefficient \( \bar{c} \) expressed as:

\[ \bar{c} = \frac{\Delta T_o \sqrt{\pi \bar{c} \rho_1 \lambda_1}}{h_{lg} \rho_v} \]  

(4-10)

with:

- \( \Delta T_o \) = initial superheat of the fluid \((T_{b\;en} - T_{v\;sat})\)
- \( \lambda_1 \) = liquid conductivity

From the equations (4-7), (4-8) and (4-9) it follows:

\[ A = 4\pi \bar{c}^2 \frac{a}{v} \]  

(4-11)

The authors assumed the interface surface constant with stage length and integrated equation (4-6) to yield:

\[ \beta = 1 - e^{-n L} \]  

(4-12)

with:

- \( n \) = evaporation index

The evaporation index is a lumped parameter describing the flashing process of a bubbly flow in open channels. The evaporation index is presented by Porteous and Muncaster as:

\[ n \approx 4\pi^2 \rho_1 \lambda_1 \frac{a h (\Delta T_o)^2}{v^2 h_{lg}^2 \rho_v^2} \]  

(4-13)
Based upon experiments of RICHARDSON and WESTGARTH [4.7] the authors allowed for an initiation length effect by modifying the equation (4-12) to:

$$\beta = 1 - e^{-n(L-L_0)}$$  \hspace{1cm} (4-14)

The flashing process is now described by an initiation length $L_0$ and a characteristic evaporator index. The authors concluded that the flashing rate is determined by the evaporation index whereas the initiation length is a function of the inlet geometry.

TOKMANTSEV, ILUSCHENKO and CHERNOZUBOV [4.14].

The authors developed a model on the basis of vaporization in a jet of non-equilibrium superheated water freely flowing into a vapour space. The authors divide the evaporation process in a length where vapour formed is consumed by the growth of nuclei present in the liquid at the entrance of the vapour space and a length of dispersed jet movement where the amount of thermal energy, consumed for evaporation, is defined by the heat transfer through a constant interphase surface. The number of bubbles is decreasing with the stage length.

The fraction of equilibration $\beta$ is by the authors also defined as the quotient of the consumed heat and the available heat, expressed as:

$$\frac{\dot{Q}}{\dot{Q}_0} = \frac{\dot{m}_b \frac{c_p}{\rho} \Delta T}{\dot{m}_b \frac{c_p}{\rho} \Delta T_0}$$  \hspace{1cm} (4-15)

with $\dot{m}_b$ = mass flow of superheated liquid entering the stage,

or:

$$\beta = \frac{T_{v, sat} - T_B}{T_{v, sat} - T_{v, en}}$$  \hspace{1cm} (4-16)

In the first area the authors define $N$ bubble per unit of mass of liquid with an average radius $R_b$. 

- 4.9 -
Energy consumed for vaporization neglecting the nucleus radius reads:

\[
\frac{c_i}{p_1} \Delta T = \frac{4}{3} \pi R^3_0 \rho_v h_{lg} N
\]  
(4-17)

Bubble radius with time is defined by:

\[
R_b = \frac{c_2}{c_p} \Delta T_o t^2
\]  
(4-18)

While the equation for the flash length versus time reads:

\[
dt = \frac{dL}{\bar{v}}
\]  
(4-19)

The mean bulk fluid velocity \( \bar{v} \) is assumed to be constant.

With the equations (4-15), (4-16), (4-17), (4-18) and (4-19) the fraction of equilibration at the start of the dispersed jet area reads:

\[
\beta_1 = \frac{4\pi N \rho_v h_{lg} c_2^3}{3 L_1^2 \Delta T_o^2} \left( \frac{3}{3v^2 c_p} \right)
\]  
(4-20)

In the dispersed jet area the derivation of the model follows the approach of \textsc{Porteous} and \textsc{Muncaster} [4.13] and the equilibration factor for this area with:

\[
\beta_2 = \frac{T_{bL_1} - T_b}{T_{b\text{en}} - T_{b\text{sat}}} = (1-\beta_1) \frac{T_{bL_1} - T_b}{T_{bL_1} - T_{b\text{sat}}}
\]  
(4-21)

reads:

\[
\beta_2 = (1-\beta_1) \left[ 1 - e^{-n(L-L_1)} \right]
\]  
(4-22)

The total fraction of equilibration is also:

\[
\beta = \beta_1 + \beta_2
\]  
(4-23)

Experimental work by the authors resulted in a few values for the number of nuclei entering the stage (N) and for \( \frac{N c_2^3}{c_p} \).
Although the model presented by the authors is of interest because it attempt to describe vaporizing of superheated water freely flowing into a vapour space, similar to the design of an MSF/FBE stage.

4.3.2. Equilibration approach model.

As can be seen from the literature surveyed many attempts have already been made to describe the equilibration process in an evaporator stage. So far none of the derived models has shown sufficient broad applicability for the evaporation process in a stage of a vertical evaporator (MSF/FBE). The model as presented by Porteous and Muncaster has been well defined and experiments are provided to support the model. However, this model has been derived for evaporation from a plane interface in open channel flow and presents the typical pool boiling process of a conventional evaporator stage.

The attempts to formulate a model as presented by Tokmantsev, Iluschenko and Chernozubov should be more applicable to the design of stages for an MSF/FBE since its basis is a superheated water flow freely flowing into a vapour space.

Limited validity of the derived equations and the lack of experimental verification data makes it difficult to adopt the presented model. An abundance of available literature such as [4.15], [4.16], [4.17], [4.18] and [4.19] should facilitate the further development of an equilibrium approach model which was considered to be beyond the scope of this thesis.

From the available literature it is understood that the vapour bubble growth is governed by different mechanisms as described below.

The evaporation process in an MSF/FBE starts when the liquid passes the orifice in the separating interstage wall. When the vapour pressure in the liquid exceeds the external pressure it becomes possible for a vapour bubble to grow from a small existing nucleus in the liquid. This nucleus is a region of non-liquid phase and presumably consists of a gas or vapour phase stabilized on a solid particle.

The rate of growth of a vapour bubble, once formed, is determined by the liquid inertia, the surface tension and the difference between the pressure within the bubble and the external pressure. In the very initial stage of bubble expansion the growth is slow but accelerated with increase in bubble size, because of the reduction in surface tension.
When the rate of bubble growth becomes appreciable, the temperature and, hence, the pressure within the bubble will drop and the rate of growth is decreased.

One might therefore expect a maximum in the velocity of bubble wall. The reduction of the temperature within the bubble is a consequence of the latent heat requirement of the evaporation. This evaporation takes place at the vapour-liquid interface as the bubble grows. The thermal equilibrium requires that there be no temperature difference between the vapour in the bubble and the liquid surrounding it.

The dynamics on cavitation and vapour bubbles for the process of bubble growth in superheated liquids for the zone in which the number of bubbles is constant can be described in physical terms as follows.

When the equilibrium of a bubble or nucleus is disturbed the bubble starts to grow slowly under the restraining effects of surface tension.

It is expected that for MSF/FBE applications the growth velocity may reach the asymptotic value for a cavitation bubble as formulated by RAYLEIGH [4.16] before the rate of vapour in-flow is large enough to produce a cooling of the surrounding liquid.

At this point both inertial and thermal effects limit the growth rate. Inertial effects become less and less important until the radius has grown so large that the growth process is limited by the rate of heat supplied to the bubble wall.

The probable physical process of growth is illustrated in figure 4-3. Calculation of data is based on the work performed by VAN STRALEN [4.18].

**FIGURE 4-3.**

*Impression of the growth rate of a vapour bubble after entering the evaporator stage.*
It was decided to obtain maximum experimental data from the available test module upon which data future modeling for the approach to equilibrium can be compared.

The following subchapters described the test module and the experiments performed.
4.4. Experimental apparatus.

Experiments carried out by KLAREN [4.20] in respect of dynamic behaviour of a downward flashing evaporator already indicated that flash-off in a vertical evaporator can be reached within smaller evaporator stage volume than with conventional horizontal evaporators.

As already mentioned in the preceding subchapters it was decided to perform experiments to increase knowledge of flash-off and entrainment separation in a evaporator stage of a Multi-Stage Flash/Fluidized Bed Evaporator.

This knowledge is of importance in order to be able to design optimum Multi-Stage Flash/Fluidized Bed Evaporators with required performance and minimum capital investment.

**FIGURE 4-4.**
Flow scheme of the three stage test module with entrainment measuring equipment provided on the second stage.
Initial experiments for determination of basic aspects of flash-off and entrainment separation are performed and reported by Klaren. These experiments were performed with vapour space loading factors less than two, i.e. two kg of distillate produced per hour and per litre of vapour space. Recognizing that the evaporator stage could be optimized further after obtaining results of detailed experiments, it was decided to modify the existing test module.

Modifications were directed to more accurate determination of entrainment separation and flash-off over a much larger range of independent variables including the vapour space loading factor. Modifications also included for different geometries of the evaporator stage.

4.4.1. Description of the test module.

The experimental apparatus used for the experiments as described in this chapter is a modified version of the three stage test module used by KLAREN [4.20] for the experimental determination of the process properties of a flash-down stage. A simplified flow scheme of the test module is presented in figure 4-4.

Modifications have been made in such a way that the test module can be used for the determination of the maximum allowable vapour space loading for various geometries of the evaporator stage.

Basic features of the test module are in essence the same as for an MSF or an MSF/FBE and it will behave as part of a Multi-Stage Flash Evaporator. It is designed to withstand full vacuum and 3 bar and has a maximum distillate production of 250 kg/h and a brine mass flow rate of maximum 16000 kg/h.

The distillate production of 250 kg/h is based on total flash range of 10 °C and a maximum heat input of 112 kW.

For a better understanding of the operating principles of the test module a short description of the process will be given.
Figure 4-5 shows a flow scheme with equipment clarification of the test module.

**FIGURE 4-5.**
Flow scheme of test module with equipment clarification.

1. heat input section
2. evaporator stage
3. entrainment separation and measuring chamber
4. condenser section
5. distillate collecting tank
6. reject heat exchanger
7. intermediate cooling water reservoir
8. cooling water supply tank
9. brine recirculation pump
10. distillate pump
11. non-condensables venting pump
12. steam trap
13. conductivity measurement
14. flow measurement
15. sampling line
Figure 4-6 shows the major measuring points of this modified test module. As mentioned before the test module operates more or less as a three stage Multi-Stage Flash Evaporator which means that liquid is heated in the condenser section by obtaining heat of condensation from the evaporated and subsequently condensed part of the brine flashing down the evaporator stages. After the condenser section the liquid obtains final temperature increase in the heat input section or brine heater and flows down the evaporator stages where in each of the stages a part of the brine is evaporated in an attempt to reach its equilibrium with the saturated pressure in the respective condenser section.

**FIGURE 4-6.**
Flow scheme of test module with indication of major process control and measuring points.
In the last evaporator stage the brine is collected and after heat reject in a cooler recirculated to the condenser section by the brine recirculation pump. Each of the condensers is connected to a central vacuum line for removal of non-condensable gases while the vacuum is maintained by a liquid-sealed rotary pump. Distillate produced in each of the condensers is collected in a tank and transported by the distillate pump. The distillate produced in the first stage flows via a steam trap to the third condenser, while for measuring purposes the distillate of the second and measuring stage is isolated, monitored and separately fed to the distillate collecting tank where it is blended with the distillate from the first and third condenser.

The test module operates in closed loop mode and therefore both brine and distillate are returned to the entrance of the condenser section. Approximately the quantity of heat added to the brine flow in the heat supply section is rejected in the heat exchanger installed between the last evaporator stage and the brine recirculation pump. Temperature of the brine entering the condenser section is mainly determined by the mass flow and the temperature level of coolant through the heat reject section.

In order to obtain possibilities to conduct measurements over a wide range of temperatures and mass flows through the evaporator section, an intermediate special cooling system with the possibility of operation with an elevated cooling water temperature was installed. This cooling system is also shown in figure 4-6. As the quantity of the cooling water and the temperature level can be varied, the experiments can be carried out at maximum brine temperatures ranging from 40 °C to as high as 120 °C.

In order to avoid unfavourable influences due to typical entrance and outlet phenomena it was decided that for the entrainment measurements including the influences caused by geometry change, only the second evaporator stage was to be observed. The measuring principle with isolation of distillate produced in the second evaporator stage and separate collection of the entrained brine is shown in figure 4-7. By separating the entrained brine and the distillate of the second stage blending with the distillate of the other stages before measuring is avoided and interior changes for optimization only have to be performed in the second stage.
FIGURE 4-7.
Detail of entrainment separation chamber provided on the second evaporator stage.

4.4.2. Operation of the test module.

Temperature level for the experiments and also the temperature drop per stage can be adjusted by the amount of heat supplied in the electrical heated brine heater and the temperature and quantity of the cooling water flow through the heat reject section.

The brine level in the second evaporator stage can be changed under stationary thermal conditions by adjustment of the variable orifice between the second and the third evaporator stage. This orifice is shown in figure 4-8.

FIGURE 4-8.
Calibrated orifice adjustment in the interstage wall.
The entrained brine from the evaporator stage is collected in the separation chamber. This chamber and the wire mesh demister pads installed are designed to catch more than 99.6% of the entrained brine. Calculations of the entrainment separation efficiency are based on the work of CARPENTER [4.21] and accordance with the efficiency calculations presented in reference [4.22]. The collected entrained brine is periodically drained in a calibrated measuring glass while the collecting time is measured with a chronometer. After stationary conditions for a certain set of adjustments are entered, a number of complete series of measurements are performed enabling to establish average values of all parameters.

<table>
<thead>
<tr>
<th>DATA ACQUISITION SYSTEM</th>
<th>CALCULATION</th>
</tr>
</thead>
<tbody>
<tr>
<td>geometry, experiment and other fixed data</td>
<td>atmospheric pressure</td>
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<tr>
<td>mercury barometer data</td>
<td>saturation pressure stage 1</td>
</tr>
<tr>
<td>1st stage pressure data</td>
<td>thermodynamic properties stage 1</td>
</tr>
<tr>
<td>IFC data</td>
<td>distillate production stage 1</td>
</tr>
<tr>
<td>brine temperature and mass flow stage 1</td>
<td>brine mass flow to stage 2</td>
</tr>
<tr>
<td>brine level stage 1</td>
<td>saturation pressure stage 2</td>
</tr>
<tr>
<td>2nd stage pressure data</td>
<td>thermodynamic properties stage 2</td>
</tr>
<tr>
<td>IFC data</td>
<td>vapour flow produced in stage 2</td>
</tr>
<tr>
<td>brine temperature stage 2</td>
<td>vertical velocity of vapour flow</td>
</tr>
<tr>
<td>brine level stage 2</td>
<td>horizontal velocity of vapour flow</td>
</tr>
<tr>
<td>differential pressure data</td>
<td>vapour space loading</td>
</tr>
<tr>
<td>sample data: quantity, time and concentration</td>
<td>pressure loss over baffle and outlet</td>
</tr>
<tr>
<td></td>
<td>salt concentration in distillate of stage 2</td>
</tr>
</tbody>
</table>

FIGURE 4-9.
Calculation procedure for the vapour space loading and entrainment data.
4.4.3. **Calculation procedure.**

The calculation procedure necessary to obtain the required data for entrainment and vapour space loading factors is visualized in figure 4-9.

4.4.4. **Pressure measurements.**

The pressure measuring system for the determination of the pressure in the evaporator stages is shown in figure 4-10 and consists mainly of a precise U-tube manometer system with a slide gauge to measure the deflection of the mercury column. Accuracy of the slide gauge reading amounts to ± 0.05 mm.

The measured deflection is then transformed to the vapour space pressure and actually represents the saturation temperature in a stage. Measuring of the saturation pressure and also the brine temperature enables a determination of the approach to equilibrium.

**FIGURE 4-10.**

_U-tube manometer connection to the second stage. At the left side typical deflections are shown for atmospheric measurement calibration._
The pressure measuring system is connected to the stages by a wide elbow to allow for mass transport from and to the constant head tank without blocking the cross section which might cause a slight pressure difference over the system. Each of the measurements is preceded by an atmospheric measurement to determine the reference level of the mercury system and the distance between the constant head tank and the average level as shown in figure 4-10.

Equal pressure on both of the legs of the precise U-tube manometer yield the following equation:

\[ P + \left( L_{w, a} - \frac{D_{m, b}}{2} \right) \rho_w g_n + D_{m, b} \rho_m g_n = P_a + L_{w, c} \rho_w g_n \]  
(4-24)

Under the assumption that no temperature difference occurs between the two mercury legs of the U-tube manometer the following equation for the absolute pressure in the stage n can be derived:

\[ P_n = P_a + g_n \left\{ 0.5 \left( D_{m, a} + D_{m, b} \right) \rho_w - \left( D_{m, a} + D_{m, b} \right) \rho_m \right\} \]  
(4-25)

In these equations the values for density are determined at the appropriate temperatures.

The atmospheric pressure at the time of the measurements for the establishing of the atmospheric values is obtained from a precise mercury barometer.

Applicable corrections for capillary depression, linear expansion differences and latitude are expressed as polynomials because of computerized data acquisition and calculations.

The total maximum error in the evaporator stage pressure \( P_n \) was found by adding the subtotals of the analyses of:

- the error on the atmospheric pressure \( \Delta P_a \)
- the error in different manometer slide gauge reading \( \Delta P_{\text{man}} \)
- the error due to temperature differences along the manometer legs \( \Delta P_{\text{temp}} \).
The maximum error in the stage pressure $P_n$ calculated according to:

$$\Delta P_n = \Delta P_a + \Delta P_{man} + \Delta P_{temp}$$  \hspace{1cm} (4-26)

The maximum error for all of the experiments was found to be less than 104 N/m$^2$.

For the purpose of the investigations reported in this chapter it is also important to obtain information about the pressure loss over the interior components. It is obvious that good entrainment and high vapour space loading should be pursued but at minimum pressure loss between the evaporator stage and the respective condenser section.

**FIGURE 4-11.**
Inverse U-tube manometer for measurement of pressure loss over the interior components.

For this purpose a differential pressure manometer as shown in figure 4-11 was installed to measure the pressure loss over the baffle plate and outlet opening. Before each measurement the connecting tubes were adequately filled with liquid of the same temperature. Maximum error in the measured pressure difference is calculated to be less than 1 mm liquid column or less than approx. 10 N/m$^2$. 

- 4.23 -
4.4.5. Temperature measurements.

Temperatures are measured with Chromel-Alumel thermocouples of the shielded type. The conductors of the thermocouples are shielded by a stainless steel sheath and are insulated by magnesium oxide. Thermocouples with a sheath outer diameter of 1 mm are used for the brine temperatures in the stages while for other temperature measurements thermocouples with a sheath outer diameter of 1.6 mm are used. Temperatures are measured with reference to a cold junction kept at a constant temperature close to the triple-point of pure water. The discrepancy between the reference and the triple-point of pure water is caused by the conductive heat flux through the reference cold junction thermocouples and amounts to 0.01 °C for all measurements. Induced voltage is measured by an integral digital voltmeter with an accuracy of ± 1 µV.

The thermocouples for the temperature measurements in the evaporator stages are calibrated before and after the measurements. Calibration has been carried out with a precise constant temperature bath containing demineralized water for calibrations of temperatures not exceeding 95 °C. The calibration reference is established by series of high accuracy mercury glass thermometers with a maximum error after calibration and certification of ± 0.02 °C.

Analysing the calibration data by computer and applying the least square method resulted in power series for all of the thermocouples. This best approximation in the least squares sense i.e. the one for which the sum of the residuals squared over the whole domain is least, resulted in a maximum residual of 0.05 °C.

4.4.6. Salt concentration measurements.

For the aims of the geometry related investigation it is important to know the salt concentration of the circulated brine and the produced distillate. In order to monitor the conductivity, temperature corrected four-points conductivity cells have been installed in the feed line to the first stage and in the distillate line after the condenser of the observed stage.

For the actual measurements, however, samples have been taken at regular intervals. These samples were analysed with respect to salt content. As only sodium chloride is used to establish a certain salinity of the brine...
it was found that the titration technique applying the quantitative method of Mohr as described by KOLTHOFF and SANDELL [4.23] yielded the most accurate results.

Measurements of sodium chloride contents are reported by VEEKNMAN [4.24]. The maximum error in the distillate salt concentration \( c_{d,2} \) is calculated to be less than six percent while the maximum error in the brine concentration \( c_{b,2} \) is less than two percent according to VAN DER SANDEEN [4.25].

4.4.7 Brine level measurements.

The evaporator stages of the test module were designed to be able to vary the brine level over a wide range. In order to reach the higher values of vapour space loading most of the evaporator stage volume will be occupied by brine in a near equilibrium state. As the brine level subsequently will be rather close to the baffle plate and recognizing that the remaining aperture is a vital parameter, accurate brine height measurement was needed. The test module is equipped with a water level gauge and a pneumatic pressure difference transmitter on each stage. The system for measuring the brine height in the evaporator stage is shown in figure 4-12. Measurements are possible with a maximum error of 1 mm liquid column or approximately 10 N/m².

![Diagram of brine level measurement](image)

**FIGURE 4-12.**

*Conversion of brine level measurement to electrical signal for data acquisition.*

The mass flow data required for the calculations to be made are:

- the brine mass flow entering the observed evaporator stage
- the brine mass flow leaving the same stage
- the mass flow to the condenser section
- the entrained mass flow.

Direct and accurate measurements of these mass flows is extremely
difficult and therefore some related mass flows are measured and the
above stated mass flows will have to be derived.

The brine mass flow at the entrance of the first evaporator stage has been
measured with an in-line calibrated magnetic-hydrodynamic flow meter with
an accuracy of ± 1%.

The amount of vapour condensed in the second evaporator stage has been
measured with a calibrated flowrator with an accuracy of ± 3%.

The amount of entrained water collected in the separation chamber has been
determined by measuring a quantity of entrained water in a measuring glass.

The time required to sample this quantity has been measured with a chrono-
meter. Maximum error in the quantity entrained water is less than ± 1.3%.

Apart from the most important mass flows the respective cooling water
flows have been measured by orifices and liquid filled differential head
manometers, while the total distillate production has been measured by a
flow rator.

The evaporation process is considered to be stationary and adiabatic.

Application of the energy and mass conservation laws for the control
volume as shown in figure 4-13 yield the following equations:

\[
\dot{\phi}_{m_b,n-1} = \dot{\phi}_{m_b,n} + \dot{\phi}_{m_d,n} \quad \text{(4-27)}
\]

\[
\dot{\phi}_{m_b,n-1} = \dot{\phi}_{m_b,n} + \dot{\phi}_{m_d,n} \quad \text{(4-28)}
\]

This results in the equation for the mass flow of distillate produced
in stage n:

\[
\dot{\phi}_{m_d,n} = \frac{\dot{\phi}_{m_b,n-1} (h_{b,n-1} - h_{b,n})}{(h_{d,n} - h_{b,n})} \quad \text{(4-29)}
\]
The difference in enthalpy can be approximated by:
\[
h_{b,n-1} - h_{b,n} \approx c_p l (T_{b,n-1} - T_{b,n})
\]  
(4-30)

The distillate mass flow can now be expressed as:
\[
\frac{\dot{m}_{d,n}}{\dot{m}_{b,n-1}} = \frac{c_p l (T_{b,n-1} - T_{b,n})}{(h_{d,n} - h_{b,n})}
\]  
(4-31)

FIGURE 4-13.
Control volume for energy and mass conservation laws.
NOTE: figure 4-14 is part of subchapter 4.5.1.

**FIGURE 4-14.**
Basic geometry A of MSF/FBE stage.
4.5. Experimental results.

4.5.1. Introductory remarks.

Experiments were carried out on three different basic geometries of an MSF/FBE stage. The basic geometry A is shown in figure 4-14. To reach higher vapour space loading factor, the interstage wall between stage 1 and stage 2 has been lowered and the evaporator stage length can be reduced in basic geometry B (figure 4-15).

![Diagram of geometry B](image)

**FIGURE 4-15.**
Basic geometry B of MSF/FBE stage.

For a further increase of vapour space loading factors and additional investigations in respect of flash-off, the brine diversion plate has been brought closer to the interstage wall and the length of the evaporator stage can be reduced in basic geometry C (figure 4-16). The brine temperatures for the experiments ranged from 40 to 65 °C and salt concentration of the brine varied between 3500 mg sodium-chloride (NaCl) per dm³ and 76000 mg sodiumchloride per dm³.

As can be seen from figure 4-14 three different baffle lengths $L_B^n$ were investigated in basic geometry A, whilst for basic geometry B stage depth was varied. Finally in an attempt to reach the highest vapour space loading values basic geometry C was tested which also included a stage depth variation.
FIGURE 4-16.
Basic geometry C of MSF/FBE stage.

The influence of salt concentration in the brine on the quality of the distillate was investigated in basic geometries A and B. As entrainment within a given geometry merely is influenced by the vapour velocities, the salt concentration of the brine, the flow patterns and the brine distribution, these parameters have been subject of investigation.

FIGURE 4-17.
Clarification of dimensions, volumes and velocities.
Figure 4-17 gives an elucidation of the horizontal vapour velocity $v_h'$, the vapour aperture $h'$, vertical vapour velocity behind the baffle plate $v_v$ and the vapour space of an evaporator stage. Vapour space loading is defined as the distillate production in kilograms per hour per dm$^3$ vapour space and is an important value for the design of evaporator stages. As shown in this drawing the vapour aperture $h'$ can be expressed as:

$$h' = H_s - L_b - H_1$$  \hspace{1cm} (4-32)

Obvious there is a close relationship between the horizontal velocity of the vapour under the baffle and the aperture $h'$. Neglecting the influence of the changing boiling point elevation, the volume flow of vapour is determined by the temperature drop over the stage $\Delta T$, the brine mass flow and the saturation temperature of the vapour produced. The horizontal vapour velocity over the brine surface under the baffle plate is regarded as one of the main parameters in the studies with respect to entrainment separation.

Because of the recognition of the importance of a minimum value for the aperture $h'$ as further described in subchapter 4.5.2.1 it was decided that the relation between the quality of the produced distillate and the horizontal vapour velocity under the baffle plate and the aperture under the baffle plate should be presented separately. The actual values of both the horizontal velocity of the vapour under the baffle and the aperture $h'$ are calculated for each of the measurements from the data obtained.

The aim of the investigations with respect to entrainment separation and flash-off is to optimize the evaporator stage design of a Multi-Stage Flash/Fluidized Bed Evaporator. Since the vapour space loading factor is regarded as a proper instrument in the reduction of the total evaporator volume and consequently to an extent of the capital investment in the MSF/FBE it was decided to present the measured and calculated values for the distillate quality against the vapour space loading.

4.5.2. Entrainment

The entrainment measurements are carried out for the above mentioned three basic geometries A, B and C:

- For each geometry series of measurements are carried out for different lengths of baffle plates and/or stage depths.
For each length of baffle plate or stage depth the following parameters have been varied independently:

- salt concentration between 3500 and 76000 ppm
- temperature level between 40 and 65 °C
- temperature drop between 1.75 and 3.00 °C

For each of the above defined measurements series the brine level in the stage 2 has been varied with steps from 0.05 to 0.01 m in order to change the horizontal velocity under the baffle plate, the aperture under the baffle plate and the vapour space loading factor.

Measurements as carried out will be reviewed below and the final results will be commented upon in the subchapter on the design criteria (4.6).

4.5.2.1. Basic geometry A.

Some typical measured and calculated values of the experiments with basic geometry A and the shortest baffle plate length of 120 mm are tabulated serving to illustrate and are shown in table 4-1.

<table>
<thead>
<tr>
<th>measurement identification</th>
<th>A-48-0440</th>
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<th>A-50-0302</th>
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<tr>
<td>salt concentration of brine ppm NaCl</td>
<td>39325</td>
<td>76000</td>
<td>70000</td>
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<td>temperature of brine °C</td>
<td>61.75</td>
<td>61.58</td>
<td>48.01</td>
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<tr>
<td>temperature drop °C</td>
<td>2.84</td>
<td>2.69</td>
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</tr>
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<td>brine mass flow stage 2 kg/s</td>
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<td>3.55</td>
<td>3.15</td>
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<tr>
<td>distillate mass flow stage 2 kg/s</td>
<td>16.3.10^-3</td>
<td>15.5.10^-3</td>
<td>9.6.10^-3</td>
</tr>
<tr>
<td>available vapour space m^3</td>
<td>15.0.10^-3</td>
<td>14.0.10^-3</td>
<td>12.7.10^-3</td>
</tr>
<tr>
<td>horizontal vapour velocity m/s</td>
<td>18.73</td>
<td>25.30</td>
<td>56.05</td>
</tr>
<tr>
<td>vertical vapour velocity m/s</td>
<td>14.90</td>
<td>14.37</td>
<td>15.92</td>
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<tr>
<td>pressure drop over interior N/m²</td>
<td>120</td>
<td>120</td>
<td>105</td>
</tr>
<tr>
<td>vapour space loading factor kg/hr dm³</td>
<td>3.91</td>
<td>3.98</td>
<td>2.72</td>
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<td>0.975</td>
<td>0.985</td>
<td>0.970</td>
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<tr>
<td>quality of distillate ppm NaCl</td>
<td>3.1</td>
<td>2.8</td>
<td>1.1</td>
</tr>
</tbody>
</table>

TABLE 4-1.
Representative measured and calculated values obtained with stage geometry A.
Some of the typical measurement series as defined above are presented as curves in the figures 4-18, 4-19 and 4-20.

The influence of the vapour space loading factor on the quality of distillate produced is presented in figure 4-18.

Figure 4-19 indicates the relation between the horizontal velocity of the vapour stream towards the stage outlet determined at the vapour aperture $h'$ and the quality of distillate produced.

**FIGURE 4-18.**
Quality of the product versus the vapour space loading factor for three representative measurements using stage geometry A (see also table 4-1).

**FIGURE 4-19.**
Quality of the product versus the horizontal velocity under the baffle plate for geometry A (see also table 4-1).
Finally figure 4-20 shows the influence of the aperture \( h' \) on the distillate quality.

**FIGURE 4-20.**
Quality of the product versus the aperture under the baffle plate for geometry A (see also table 4-1).

Clearly can be seen in the figures with reference to the vapour space loading factors and the horizontal velocity of the vapour that deterioration of the distillate produced for different baffle lengths occurs at the different values for these two parameters. As can be seen from other measurements in this chapter higher velocities than 30 meter per seconds can be achieved without a sharp increase in salt concentration of the distillate. The reduction of the aperture \( h' \) shows an asymptotic deterioration of the distillate at an aperture of 25 mm. At all these measurements and different baffle lengths the approach to equilibrium was within acceptable limits (\( B > 0.960 \)).

As can be seen from figure 4-20 the salt content of the distillate increases with an increase of the aperture \( h' \). This is caused by a reduced separation action in the first part of the vapour channel due to lower horizontal velocities and will be further explained in subchapter 4.6.2.
It became clear that for the measurement with basic geometry A the limiting factor with all three baffle lengths has been the critical aperture $h'$. This can be explained by the phenomenon of wave-building on the surface due to shear stresses exerted on the water surface [4.24]. A small build-up of pressure upstream the baffle plate intensifies the waves in this area and once a certain minimum aperture is reached the horizontal velocity of the vapour causes a blow away of the upper part of these waves and consequently deterioration of distillate occurs. This process is visualized in figure 4-21.

![Wave building under the baffle plate at critical value for the aperture.](image)

FIGURE 4-21.

Wave building under the baffle plate at critical value for the aperture.

A theoretical explanation of this wave building phenomenon based on wave theories is beyond the scope of work as presented in this thesis. Experimental investigations and results clearly demonstrate the minimum possible aperture without deterioration of the product. The experiments made it clear that the maximum load of the evaporator stage is governed by the critical aperture between baffle plate and brine surface. This also proves that neither the horizontal velocity nor the vapour space loading were determining factors in the measurements with geometry A.

4.5.2.2. Basic geometry B.

As the limit in vapour space loading, still under conditions of a good approach to equilibrium and a very low pressure drop over baffle plate and outlet, were not met in basic geometry A, due to the limiting dimension of the aperture, the test module has been modified.
The modified evaporator stage interior is shown in figure 4-15. This interior has no baffle plate but shows a split level arrangement of interstage wall and vapour outlet. The measurements comprised of two different available flash lengths, namely 500 and 390 mm and representative measured and calculated values are summarized in table 4-2.

<table>
<thead>
<tr>
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<th>B-51-1855 D1 = 500</th>
<th>B-AW-0800 D2 = 390</th>
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<tr>
<td>Salt concentration of brine</td>
<td>ppm NaCl</td>
<td>67600</td>
</tr>
<tr>
<td>Temperature of brine</td>
<td>°C</td>
<td>63.47</td>
</tr>
<tr>
<td>Temperature drop</td>
<td>°C</td>
<td>2.07</td>
</tr>
<tr>
<td>Brine mass flow stage 2</td>
<td>kg/s</td>
<td>3.66</td>
</tr>
<tr>
<td>Distillate mass flow stage 2</td>
<td>kg/s</td>
<td>17.3 x 10^{-3}</td>
</tr>
<tr>
<td>Available vapour space</td>
<td>m³</td>
<td>10.4 x 10^{-3}</td>
</tr>
<tr>
<td>Horizontal vapour velocity</td>
<td>m/s</td>
<td>20.60</td>
</tr>
<tr>
<td>Vertical vapour velocity</td>
<td>m/s</td>
<td>15.0</td>
</tr>
<tr>
<td>Pressure drop over interior</td>
<td>N/m²</td>
<td>110</td>
</tr>
<tr>
<td>Vapour space loading factor</td>
<td>kg/hr dm³</td>
<td>5.99</td>
</tr>
<tr>
<td>Fraction of equilibration</td>
<td></td>
<td>0.985</td>
</tr>
<tr>
<td>Quality of distillate</td>
<td>ppm NaCl</td>
<td>5.3</td>
</tr>
</tbody>
</table>

**TABLE 4-2.**
Representative measured and calculated values obtained with stage geometry B *).

Figures 4-22, 4-23 and 4-24 elucidate typical results of the measurements with geometry B. The figures show the dependence of the produced quality against the vapour space loading factor (figure 4-22), horizontal velocity (figure 4-23) and the aperture h' (figure 4-24).

The experiments show that, although a slight increase in salt concentration in the distillate occurs, only with a very small aperture h' the quality of the distillate deteriorates in a serious way.

*) In subchapter 4.5.1 the evaporator stage length has been identified as $L_n$ whereas in the remaining subchapters stage length (or depth) has been presented as $D_n$. 

- 4.36 -
FIGURE 4-22.
Quality of the product versus the vapour space loading factor (geometry B).

Figure 4-23.
Quality of the product versus the aperture velocity (geometry B).

The configuration with the short flash length shows a higher entrainment due to secondary droplets initiated by the brine where it hits the flash stage wall. The effective lower vapour space is a result of the combination of the split level system without baffle plate and the shorter flash length in the second geometry version. The vapour space loading factor as high as seven kg distillate produced per hour and per dm$^3$ of vapour space, before deterioration starts and as already explained this maximum is limited by the small values for the aperture h'.
FIGURE 4-24.
Quality of the product versus the aperture (geometry B).

<table>
<thead>
<tr>
<th>measurement identification</th>
<th>C-AW-0900 (D₁ = 500)</th>
<th>C-AW-1000 (D₂ = 390)</th>
</tr>
</thead>
<tbody>
<tr>
<td>salt concentration of brine</td>
<td>ppm NaCl</td>
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</tr>
<tr>
<td>temperature of brine</td>
<td>°C</td>
<td>61.60</td>
</tr>
<tr>
<td>temperature drop</td>
<td>°C</td>
<td>2.71</td>
</tr>
<tr>
<td>brine mass flow stage 2</td>
<td>kg/a</td>
<td>3.56</td>
</tr>
<tr>
<td>distillate mass flow stage 2</td>
<td>kg/a</td>
<td>15.7×10⁻³</td>
</tr>
<tr>
<td>available vapour space</td>
<td>m³</td>
<td>8.2×10⁻³</td>
</tr>
<tr>
<td>horizontal vapour velocity</td>
<td>m/a</td>
<td>20.70</td>
</tr>
<tr>
<td>vertical vapour velocity</td>
<td>m/a</td>
<td>14.46</td>
</tr>
<tr>
<td>pressure drop over interior</td>
<td>kW/m²</td>
<td>105</td>
</tr>
<tr>
<td>vapour space loading factor</td>
<td>kg/m² per dm³</td>
<td>6.89</td>
</tr>
<tr>
<td>fraction of equilibration</td>
<td></td>
<td>0.965</td>
</tr>
<tr>
<td>quality of distillate</td>
<td>ppm NaCl</td>
<td>16</td>
</tr>
</tbody>
</table>

TABLE 4-3.
Representative measured and calculated values obtained with stage geometry C.
4.5.2.3. Basic geometry C.

In a continued attempt to increase the vapour space loading factor the effective vapour space above the flashing brine was decreased (basic geometry C) as shown in figure 4-16. Representative measurements are given in table 4-3 and the measured and calculated values are displayed in figures 4-25, 4-26 and 4-27. It can be concluded from these measurements that the flashing of the brine was influenced by the geometry since the approach to equilibrium deteriorated.

FIGURE 4-25.
Quality of the product versus the vapour space loading factor (geometry C).

On the upper side of the flashing brine not sufficient space is available for vapour release and effectively reduces the heat transfer surface of the flashing brine. Measurements with the shorter available flash length Dn as presented in table 4-3 (measurement C-AW-1000) show a reduced fraction of equilibration compared with the measurements based on a longer flash length. It can be concluded that the maximum vapour space loading factor with still good approach to equilibrium (β > 0.960) measured with the geometry version C was lower than the measurements with geometry version B revealed. The geometry version C is not favoured because of high entrainment for both tested flash lengths and deteriorated approach to equilibrium for the version with short flash length.
Subchapter 4.6 will elucidate more of the features of the different geometries and combine these to valuable design criteria with respect to evaporator stage geometry.

**FIGURE 4-26.**
Quality of the product versus the aperture velocity (geometry C).

**FIGURE 4-27.**
Quality of the product versus the aperture (geometry C).
4.5.3. Equilibration.

Measurements were performed to obtain information with respect to the approach to equilibrium in an attempt to formulate the progress of vapour release in the flashing brine. The experimental apparatus as described in subchapter 4.4 was equipped with thermo-elements to measure temperatures in different positions in the flashing flow, vapour space and brine pool. However, no consistent figures could be drawn for a broad range of the parameters for the progress of vapour release as the number of experimental data was too low to develop an empirical model for the vapour release in the flashing brine.

For the configuration of flash stages tested, the resulted fractions of equilibration are shown in the figures 4-28, 4-29 and 4-30.

**FIGURE 4-28.**
Approach to equilibrium measured for three initial superheat values and three salt concentration values (geometry A).

**FIGURE 4-29.**
Approach to equilibrium measured for two initial superheat values and two flash lengths, D (geometry B).
FIGURE 4-30.
Approach to equilibrium measured for two initial superheat values and two flash lengths, D (geometry B).

As a result of the high values of approach to equilibration in the area of the installed thermo-elements in all investigated geometries it was not possible, for curves showing the fraction of equilibration versus evaporation length, to draw a clear distinction between the different geometries. The most likely shape of the curves for the different geometries is presented in the magnified part of the figure 4-31.

FIGURE 4-31.
Equilibration versus flash length.

For a particular set of values of the different parameters an indication of the fraction of equilibration versus the evaporation length is presented in figure 4-31.
As can be seen from this figure, the fraction of equilibration is dependent on the temperature drop or initial super heat.

In order to derive an equilibration model it will be necessary to modify the test module and to have a large number of thermo-elements in the flash stream as well as better observation possibilities for the investigation of the physical process with respect to the flow regions.

As the measurements as described in this chapter mainly served two purposes: entrainment and equilibration, the author comments that the investigated geometries with the exception of the short flash length version in geometry C are all satisfactory with respect to the fraction of equilibration as near equilibration was reached in all variations of the geometries. The governing influences on the design will be discussed more in detail in subchapter 4.6 about the entrainment and entrainment separation performance.
4.6. Some design criteria and concluding remarks.

4.6.1. Entrainment separation:

For an explanation of separation in an MSF/FBE stage it is of value to know about the behaviour of liquid droplets in the vapour stream. The liquid layer passing the orifice in the interstage wall breaks up into thin filaments and droplets due to vapour release from the brine. At the start the waterdroplets move approximately with the water velocity which is in the opposite direction of the condenser bundle. Finally the vapour and the waterdroplets will move in the direction of the condenser bundle. A part of the total quantity of droplets will under the gravity forces drop into the brine pool.

Some of the droplets and filaments of water that collide with the vertical splash plate will break up into smaller droplets. These droplets will be caught partially by the water surface of the pool and partially be carried away with the vapour.

The vapour stream passes underneath the baffle plate and is afterwards vertically directed to the condenser opening. Under the baffle plate the horizontal velocity is increased and will carry a portion of the droplets with a horizontal velocity into the vertical channel.

The horizontal velocity of the droplets will contribute to the catchment on the vertical wall of the flash stage. Those droplets not separated by the combination of changes in direction and the gravity forces now have to move against the gravity force into the vapour channel on their way to the condenser opening.

A last separation action takes place when the vapour once more changes its direction by entering the condenser opening.

For entrainment of droplets and deterioration of the product it is important to study the droplets which passed the baffle plate and entered the vertical channel.

First an analysis of the maximum droplets size and gravitational separation is given and in the second part small liquid droplet behaviour in the vapour channel is analysed.

The friction forces exerted on the liquid droplets will deform the droplets from a purely spherical to an elongated from until the droplet breaks up. Characteristic for this balance between friction forces and surface tension is the Weber number:
The critical Weber number amounts to 22 \[4.27\]. For sudden acceleration being the case in the entrance of the vapour channel of an MSF/FBE the critical Weber number amounts to 13. The maximum droplet size for the reported experiments for a Weber number of 13 ranges between 4 and 9 mm \[4.25\], \[4.29\]. For these droplets it can be calculated if they will drop due to gravity force against the friction forces caused by the upflowing vapour in the vertical channel.

The equation for the apparent weight against the resistance forces yields:

\[
\frac{\pi}{6} a_d^3 \left( \rho_l - \rho_v \right) g_n = c_\omega \frac{\pi}{4} a_d^2 \frac{1}{2} \rho_v V_v^2
\]

As the Reynolds number for the above mentioned experiments ranges between 30 and 120, the relation between \(c_\omega\) and \(Re\) has to be found by numerical solution and therefore the force balance is written as:

\[
c_\omega \ Re^2 = 4 \frac{d^3}{d} \rho_v \left( \rho_l - \rho_v \right) g_n \frac{\eta_v^2}{n_v}
\]

As the vertical velocity in the experiments ranged from 12 to 55 meter per second, it was calculated, using equation (4-35), that the critical vapour velocity for liquid temperature of 60 \(^\circ\)C and droplets of 2 mm in some of the experiments was met by the actual vapour velocity. In these cases 2 mm droplets were just carried to the vapour outlet.

It is evident that also droplets smaller than 2 mm will cause a deterioration of the distillate.

Literature survey showed that in conventional evaporators fifty weight percent of the droplets formed have a diameter between 80 and 120 \(\mu\)m \[4.5\].

Droplets formed in the conventional evaporators (MSF) originate from splashing and secondary droplets formed by vapour bubbles breaking through the surface as has been illustrated in figure 4-1.

Secondary droplets are known to be of average small size \[4.5\].
Since calculations show that entrainment of small droplets is more difficult than for larger droplets, a conservative approach has been followed in assuming that droplets formed in the vertical evaporation process are as small as in the conventional horizontal evaporator. A water droplet passing underneath the baffle plate either will hit the wall or is carried along with the vapour flow. As can be expected the droplet size will be for a large percentage in the range between 10 and 100 μm.

For detailed analysis of droplet behaviour, in order to design proper evaporator stage interiors, droplets of 10 and 20 μm have been selected for illustrative calculations. Behaviour of a droplet with a diameter of 10 μm positioned in the middle of the passage $A_h$ is calculated according to reference [4.26]. As follows from the calculations the droplets will just hit the wall, while in these calculations the gravity influence is neglected. Figure 4-32 elucidates the results of the calculations.

---

**FIGURE 4-32.**

Behaviour of a 10 μm droplet in the cross section $A_h$. 

- 4.46 -
Under idealized circumstances 50 percent of the droplets with a diameter of 10 μm occurring in the passage $A_h$ will be separated by centrifugal forces.

The entrainment of droplets immediately after passing the vertical baffle plate is of extreme importance.

With a decrease of the vertical aperture $h'$ under the baffle plate the horizontal velocity of the vapour and droplets will increase. Higher velocities will lead to better entrainment after the passage.

As can be seen from the results of the experiments presented in figures 4-20, 4-24 and 4-27 the quality of the product water improves with a decrease of the aperture $h'$ until a limit is reached where the wave formation under the baffle plate occurs as described in subchapter 4.5.2.1.

Optimum design of flash stages with a minimum aperture $h'$, however, are in conflict with a flexible operation of vertical evaporators where a certain variation in brine level is essential.

It is for that reason that optimum design of flash stages also must take into consideration the entrainment as can be obtained in the vertical channel before the condenser opening.

Figure 4-33 shows the behaviour of droplets of 10 μm situated in the centre of passage $A_y$.

**FIGURE 4-33.**

*Behaviour of a 10 μm droplet in the cross section $A_y$.***
Calculations of droplet paths clarified that 10 μm droplets in passage $A_v$ at a saturation temperature of 60 °C and a vertical velocity of the vapour of 25 m/s are separated from the vapour flow. Droplets with larger diameter will be removed from the vapour flow even when they appear closer to the vapour outlet than shown. Separated droplets are caught by the wet film on the surface of the evaporator stage and will run down the walls.

It is obvious that the stand-up collar around the vapour discharge opening is needed to guide the separated entrained brine around the discharge opening.

Experiments under comparable conditions without the collar showed salt concentration in the distillate produced up to 7000 milligrams per litre. Finally figure 4-34 shows the combined vapour paths through the zones I, II and III.

\[ \text{FIGURE 4-34.} \]
Illustration of the three separation zones and flow pattern of a 10 μm droplet in the cross section $A_{v2}$. 
This figure also elucidates the influence of a larger zone I or passage under the baffle plate. Due to larger available square area under the baffle plate, the horizontal velocity is lower and consequently the separation action of zone I will be reduced. This also explains the higher salt concentration in the distillate found in the experiments when applying larger aperture $h'$. To an extend this can be overcome by a modified baffle plate [4.29]. This modification is shown in figure 4-35.

**FIGURE 4-35.**
Flow pattern of vapour and 10 μm droplets in a stage version with modified baffle plate.

Multi-Stage Flash Evaporators are commonly designed for a constant temperature drop per stage with certain value for approach to equilibrium and implicitly a pressure loss over the vapour path. This pressure loss is composed of a pressure loss over the condenser bundle and the steam-water separation section. The pressure loss over the separation system and the vapour discharge opening will be analysed and quantified. The pressure loss over the condenser bundle can be calculated with existing formulas and is less sensitive for design and lay-out of the evaporator stage. The pressure losses in the evaporator stage are caused by wall friction, velocity and direction changes and result in a difference between the saturation temperature in the condenser and the saturation temperature in the evaporator stage.

The vapour volume through the channel and the discharge opening reads:

\[
\dot{V}_v = \frac{\dot{m}_v}{\rho_v} \quad (4-36)
\]

while the vapour velocity for a representative passage \( h^* \) is given by:

\[
\dot{V}_v = \frac{\dot{m}_v}{W_\delta \rho_v h^*} \quad (4-37)
\]

with:

\( W_\delta = \) width of the evaporator stage

For small pressure differences the pressure loss can be related to \( \Delta T_{\text{sat}} \) as:

\[
\Delta p_{\text{loss}} = \left( \frac{\partial p}{\partial T_{\text{sat}}} \right)_{\text{sat}} \Delta T_{\text{sat}} \quad (4-38)
\]

The pressure loss reads:

\[
\Delta p_{\text{loss}} = \zeta \frac{1}{2} \rho_v \left( \frac{\dot{m}_v}{W_\delta \rho_v h^*} \right)^2 \quad (4-39)
\]
Elimination of $\Delta p_{\text{loss}}$ yields:

$$
\xi \frac{\phi_m^2}{2 W_0^2 \rho \Delta V} h^* = \left( \frac{\partial p}{\partial T} \right)_{\text{sat}, \text{sat}} \Delta T
$$

(4-40)

Finally the passage $h^*$ can be determined as:

$$
h^* = \frac{\phi_m}{W_0} (\xi)^{0.5} \left( \frac{\rho_v}{2} (\Delta T)_{\text{sat}} \left( \frac{\partial p}{\partial T} \right)_{\text{sat}, \text{sat}} \right)^{-0.5}
$$

(4-41)

For a constant vapour mass flow, evaporator stage width and friction coefficient the passage $h^*$ reads:

$$
h^* = C_1 \left( \frac{\rho_v}{2} (\Delta T)_{\text{sat}} \left( \frac{\partial p}{\partial T} \right)_{\text{sat}, \text{sat}} \right)^{-0.5}
$$

(4-42)

Figure 4-36 presents $\frac{h^*}{C_1}$ against the saturation temperature for different values of $\Delta T_{\text{loss}} = \Delta T_{\text{sat}}$. (curves 1 through 4).

![Diagram showing $\frac{h^*}{C_1}$ against $T_{\text{sat}}$ for different $\Delta T_{\text{loss}}$.]

**FIGURE 4-36.**

Representative value of the steam passage to the condenser $h^*$ versus the saturation temperature in the stage for different values of temperature loss due to pressure drop over the passage.

- 4.51 -
For a constant $\Delta T_{\text{loss}}$ this will result in relatively large values for $h^*$ at low saturation temperatures as the specific volume of the saturated vapour exponentially increases with lower temperatures. Temperature losses due to pressure losses over the internals of an evaporator stage are to be minimized with energy costs in mind but also taking into considerations the investment costs of the evaporator. Very small temperature losses would lead to large values for $h^*$ and would result in higher stages in the lower temperature end of the evaporator. For an optimum design concessions have to be made with respect to the temperature losses versus the stage heights. Based upon energy consumption considerations in dual purpose plants (water and electricity) and investments costs for MSF/FBE designs up to 4000 m$^3$/day the author proposes a design temperature loss of 0.05 °C for temperature above 60 °C and a varying temperature loss according to equation (4-47) for temperatures less than 60 °C:

$$\Delta T_{\text{loss}} = (0.3) \left( \frac{T_{\text{sat}} - 25}{10} \right) + 0.05$$

(4-47)

This function is shown in figure 4-37 while the resulting values for $h^*$ are plotted in figure 4-36.

**FIGURE 4-37.**
Temperature loss against the saturation temperature as proposed by the author and resulting from thermo-economical optimization.
4.6.3. Equilibration.

As already explained in subchapter 4.5.3 the approach to equilibrium for the tested geometries and set of variables is found to be always better than 0.95 with initial super heat as low as 1.8 °C. A flash length of 500 mm has shown to be sufficient with respect to the equilibration for all measured variables. As can be seen from the results with the basic geometry C the vapour release, in order to obtain a high approach to equilibrium in a relative short length, requires a free flow in the vapour space. The configurations with a reasonable vapour space above the jet and dispersed jet result in better values of the approach to equilibration. Reduction of the vapour space above the jet flow not only suppresses the total interfacial surface for heat transfer but also causes a higher entrainment due to the formation of a semi-channel flow with subsequent vapour release through the liquid layer.

Optimization of the combined results as specified in the subchapter 6.4.2 and this subchapter result in a basic configuration of an MSF/FBE stage lay-out as shown in figure 4-38.

**FIGURE 4-38.**

*Visualization of stage design resulting from the investigations.*
In order to create a elongated steam path to the condenser or, in case the condenser length per stage allows for, to reduce the stage height of the MSF/FBE it is possible to design the entire stage in a split-level way.

The advantage in stage height reduction, however, should be balanced by an engineering and manufacturing review of the entire costs of the system. The split-level solution is shown in figure 4-39.

**FIGURE 4-39.**
*Visualisation of the split-level stage design as an alternative.*

In order to visualize the total concept of an MSF/FBE including the items as discussed in this chapter as well as from other chapters a technical impression of unit with an approximated capacity of 3000 m³/day of product water has been drawn and is presented in figure 4-40.
FIGURE 4-40.
Conceptual design of a 3000 m³/day Multi-Stage Flash/Fluidized Bed Evaporator (MSE/FBE).
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5.1. Introductory remarks.

In an early stage of the work reported in this thesis it was decided to build a pilot plant based on available results of early subinvestigations and recommendations by KLAREN [5.1]. The major reason for the decision was the necessity of operational experience with a complete Multi-Stage Flash/Fluidized Bed Evaporator. With this pilot plant valuable information could be gathered with respect to heat transfer data and hydrodynamic stability of the fluidized bed bundle. During the construction and testing of the pilot MSF/FBE a number of the subinvestigations as described in Chapters 2, 3 and 4 were carried out on different test modules and apparatus. Afterwards the results and the consequences of these subinvestigations were as far as possible introduced as modifications, for verification purposes, in the pilot plant.

5.2. Construction of the pilot plant.

5.2.1. Plant lay-out.

Figure 5-1 shows the configuration of the pilot plant. This MSF/FBE is of the "once-through" system as shown in figure 5-2 and consists of 5 stages. A design with five evaporator stages were chosen because of steam availability and bearable investment costs. The pilot plant is manufactured in the Central Manufacturing Facility of the University at Delft and placed in the Central Boilerhouse of the University serving as an extra water supply for the hot water boilers of the central heating system.

The evaporator can be described as a counter-current heat exchanger. The raw feed is heated in the heat recovery section which is designed as a liquid/solid fluidized bed system. After a final heat supply in the integrated fluidized bed heat input section, the brine flashes through all stages via the orifices in the interstage walls. A gradual drop in saturation temperature occurs, resulting in a partial evaporation of the brine in each of the flash stages.

The vapour flows after steam-water separation to the condenser surfaces, which are cooled by the incoming feed. Distillate cascades down the same way as the brine and is collected in the distillate sump below the fifth stage. Non-condensable removing is performed by a two stage steam operated ejector system.
FIGURE 5-1.
Configuration of the pilot plant.

FIGURE 5-2.
Simplified flow scheme of the "once-through" pilot plant.
Table 5-1 shows a summary of design information as they are chosen or concluded from computer calculations.

<table>
<thead>
<tr>
<th>MSF/FBE pilot plant</th>
<th></th>
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<tbody>
<tr>
<td>distillate production</td>
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</tr>
<tr>
<td>steam consumption</td>
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</tr>
<tr>
<td>steam pressure</td>
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<td>bar</td>
</tr>
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<td>°C</td>
</tr>
<tr>
<td>condensate temperature</td>
<td>105.0</td>
<td>°C</td>
</tr>
<tr>
<td>gain-ratio</td>
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<td>-</td>
</tr>
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<td>kJ/kg</td>
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<td>maximum brine temperature</td>
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</tr>
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</tr>
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<td>blow-down temperature</td>
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</tr>
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<td>distillate temperature</td>
<td>34.7</td>
<td>°C</td>
</tr>
<tr>
<td>number of stages</td>
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<td>-</td>
</tr>
<tr>
<td>stage height</td>
<td>800</td>
<td>mm</td>
</tr>
<tr>
<td>concentration of the feed</td>
<td>1000</td>
<td>ppm TDS</td>
</tr>
<tr>
<td>brine flow (feed)</td>
<td>18286</td>
<td>kg/hr</td>
</tr>
<tr>
<td>number of condenser tubes</td>
<td>288</td>
<td>-</td>
</tr>
<tr>
<td>condenser tube material</td>
<td>aluminium-brass</td>
<td></td>
</tr>
<tr>
<td>condenser tube diameter</td>
<td>I.D. 13.44</td>
<td>mm</td>
</tr>
<tr>
<td></td>
<td>O.D. 15.88</td>
<td>mm</td>
</tr>
<tr>
<td>porosity of the fluidized beds</td>
<td>0.82</td>
<td>-</td>
</tr>
<tr>
<td>diameter of the fluidized particles</td>
<td>2.0</td>
<td>mm</td>
</tr>
<tr>
<td>density of the particle material</td>
<td>2550</td>
<td>kg/m³</td>
</tr>
<tr>
<td>total particle weight</td>
<td>94</td>
<td>kg</td>
</tr>
</tbody>
</table>

**TABLE 5-1.**

**Design of the pilot plant.**

The condenser is of the long tube design and consists of 288 condenser tubes of aluminium-brass, each with a total length of 5.22 meter. The outer diameter of the tubes is 15.88 mm and the wall thickness is 1.219 mm (18 ISWG).

The total installed nett heat transfer area amounts to 72.7 m², of which 57.6 m² is installed in the recovery section and 8.4 m² in the brine heater. The spherical solid particles are made of glass and have a diameter of 2 mm.

The plant is of circular construction type. Both diameter and stage height were conservative designed.

A circular configuration is suitable for vacuum and above atmospheric pressure condition without any reinforcements at these sizes of height and diameter.

- 5.3 -
The brine heater of the plant is an extension of the highest temperature stage of the recovery section. Therefore it was not necessary to install extra tube plates, waterboxes and accessory piping as it would be necessary with a separate brine heater.

The interstage walls are not welded to the circular shell, but sealed by a rubber profile as shown in figure 5-3. This construction was chosen as access to the type of flash chamber for piping the condenser bundle and welding of particular parts would be too difficult in view of the short stage height and small space between the condenser bundle and the outer shell. All interstage walls are welded together with the necessary siphons, in this way providing a rigid construction of the condenser bundle, condenser shell, siphons and interstage walls. The condenser bundle, interstage walls, siphons, brine orifices and flash baskets could be assembled outside the evaporator shell and after that with internals these were as a complete package mounted in the outer shell.

**FIGURE 5-3.**
Detail of rubber sealing of the interstage wall.
The rubber string is made of neoprene rubber and tightened to the inter-stage walls as shown in figure 5-3. The flexible string bends upwards when the assembled condenser bundle is pushed into the shell. During operation the circular part of the string comes back by its flexibility and is pushed against the shell by hydrostatic pressure of the liquid in the stage and by the positive pressure difference resulting from the equilibria in the two stages.

For all operation conditions of the fluidized bed heat exchanger the bed requires to be expanded up to the outlet water box. Since the fluidized bed heat exchanger is designed for a certain average porosity in the fluidized bed this leads to a minimum flow to maintain homogeneous fluidization over the full length of the condenser tubes.

In a start-up situation the feedwater is not heated and consequently no saturation pressure differences are available for brine transport. The driving force for the interstage brine flow during start-up procedure consists only of the hydrostatic pressure of the brine height in the stages. Brine height equal to the stage height will be insufficient for a hydrostatic pressure large enough for the minimum brine flow from one stage to another required for a homogeneous fluidization.

*FIGURE 5-4.*

Non-flooding siphon tube connecting the evaporator stages.
This design for the pilot MSF/FBE has a low number of stages and a reasonable temperature span between top temperature and blow down temperature. Flooding of the flash stages is therefore unavoidable unless special devices are applied to increase the interstage brine flow with a certain hydrostatic pressure during start-up and other circumstances with reduced pressure differences over the stages. The for this particular case chosen solution - small plant with not too many stages - is the application of interstage brine siphons running from stage to stage. The principle of this non-flooding precaution is shown in figure 5-4.

When the brine level in the flash stage reaches the siphon inlet an additional cross area between two consecutive stages becomes available. During normal operation the siphon is a water seal and avoids vapour short circuit between the stages. The constructed evaporator contains four of these siphons in the flash chambers.

**FIGURE 5-5.**
Siphon tubes for brine transport as a structural component for interstage wall support.

**FIGURE 5-6.**
Stirring device in upper waterbox.
The siphons have been used as a structure to support the interstage walls by welding the interstage walls to the siphons. The tubes for the siphons are running through the entire evaporator and are also welded to the upper flange. This construction is shown in figure 5-5. A part of the interstage brine can flow through the siphon tube. This is realized by making slits in the siphon tube as shown in figure 5-5.

Under normal operating conditions a layer of particles is in the upper-waterbox on the tube plate and depending on the dimensions of the upper-waterbox the fluidization in this layer might be of non-homogeneous type. Low turbulence in this particles layer might cause fouling of this layer and with reduced mass flow even bridging of the particles above the outlet opening of the tubes. To avoid this problem a simple stirring device as shown in figure 5-6 was mounted on the lid of the upper-waterbox. Steam-water separation in the flash stages is performed by gravity effect and by directing the flow to the condenser bundle.

**FIGURE 5-7.**
Steam-water separation by centrifugal effects.
Figure 5-7 shows the vapour paths from the entrance in a particular stage to the condenser. Both in vertical and in horizontal direction there is a strong bending of the steam paths, causing centrifugal effect on the droplets. As also can be seen in figure 5-7, the water droplets are directed to the brine pool. In the pilot plant described no wire-mesh demisters were installed for steam-water separation. Distillate cascades down the plant and the interstage distillate transport takes place via distillate siphons. The distillate siphons also act as water seal avoiding vapour short circuit between the stages in the condenser bundle. The distillate siphons are welded at the bottom side of the respective interstage wall and end in the middle of the lower stage behind a screen as can be seen in figure 5-8. The lowest point of each of these distillate siphons is in the inlet waterbox and is connected to a distillate sample system.

FIGURE 5-8.
Distillate siphon tube system (one out of four tubes shown).
5.2.2. Manufacturing and erection of the pilot plant.

The decision to build a Multi Stage Flash/Fluidized Bed Evaporator was made in 1974 and negotiations started with the Central Boiler House of the University where the unit was to be placed to serve as an additional process water supply for the Central Heating Boilers. The pilot plant is engineered together with Piencon Consulting Engineers now incorporated in Esmil International b.v., Amsterdam.

Manufacturing of the pilot plant is carried out in the Central Workshop of the University and erection of the plant took place in early 1975. After completion on piping and control systems the first start-up was performed in April 1975. Figure 5-9 shows a simplified scheme of the completed pilot plant including pumps, intermediate storage tanks, steam supply and non-condensables removing equipment.

**FIGURE 5-9.**
Flow scheme of pilot plant.
FIGURE 5-10.
Remote control panel in operator room.
5.3. Instrumentation of the pilot plant.

This subchapter describes the instrumentation of the pilot plant and is divided in a paragraph for instrumentation for measurement and in a paragraph for instrumentation for process control purposes. All measured values necessary for a proper control of the process are displayed on the remote control panel of the pilot plant. This remote control panel is integrated in the centralized instrumentation of the Central Boiler House. Necessary process adjustments as well as start-up and shut-down procedures can be completed from the control room. Figure 5-10 shown the remote control panel which is of the "blind scheme" type.

5.3.1. Instrumentation for measurements.

Calibrated platinum-resistance elements (PT 100) are installed for temperature measurements.

Canal water temperature, feedwater temperature, maximum brine temperature, supplied steam temperature, brine temperature in the various evaporation stages, condensate-, distillate- and blow down temperature can be measured with the installed elements.

Measuring points are indicated on the remote control panel by means of Light Emitting Diodes (LED's) showing which element in the measuring circuit is activated by the selective multi-function switch. The measured values are displayed on the panel mounted digital volt meter in degrees Celsius.

The flow of condensed steam from the brine heater is measured by a flow meter of the positive displacement type. The flow meter is provided with a flow dependent puls generator and transmits the pulses to the integrating flow meter in the remote control panel where the cumulative amount of consumed steam is displayed. The distillate or product flow is identically measured and the cumulative product water amount is also displayed on the panel. A pressure indicator in the panel shows the reduced pressure of the steam supplied to the brine water.

Each evaporator stage is supplied with a calibrated pressure indicator. Product water and returned condensate qualities are measured with conductivity-cells and their respective values are displayed analogously on the control panel.
5.3.2. Instrumentation for process control.

Figure 5-9 shows a simplified piping and instrumentation scheme. To achieve a high degree of flexibility of operation and a proper control of operation the pilot plant is equipped with six control circuits.

- Feedwater flow control; as stated before the flow through the fluidized bed condenser bundle is to be kept at nominal value because of fluidization over the entire length of the tubes. A special self-contained flow control valve is provided and mounted between the feed-pump and the entrance of the condenser bundle. Set point of this valve can be varied from 0 to 30 m³/hr and the selected flow is kept by the valve independent of pump and back pressure.

- Feedwater temperature control; a three way valve is provided to maintain a constant inlet temperature independent of surface water temperature. The control valve is of a self-contained type and feeds the feedwater pump with brine from the blow-down pump.

- Steam pressure control; steam from a "steam block"-type boiler is used as external heat supply for the pilot plant. This steam is generated at a pressure of 9 bar and the pilot plant is equipped with a steam pressure reducing station delivering steam with a pressure of 2.5 bar to the brine heater.

- Steam temperature control; to reduce the steam temperature to the saturation temperature a steam cooler of the spray-type is mounted in the steam line. A self-contained temperature control system regulates the amount of condensate to the spray cooler.

- Maximum temperature control; top temperature of the feed i.e. the temperature of the heated feed to the first evaporation stage is controlled by an electrical circuit consisting of:
  - electrical actuated control valve
  - servo drive
  - electrical temperature controller
  - maximum value thermostat
  - set point potentiometer.

This systems allows for adjusting of maximum temperature by steam supply regulation. The system is electrical operated enabling process control from the remote control panel.
Conductivity control; conductivity-measurement connected to a control device and solenoId valves in the connecting pipelines both for distillate and condensate is provided. Exceeding of conductivity-values over a pre-set value causes an activation of blow-down solenoId valves and the respective flow is dumped. Necessary alarm functions are provided and signal also for low level in the constant head tank and increased pressure drop over the dual filters in the feed line. Distillate, condensate and blow-down pump are protected against superheating by means of a back flow system controlled by a float acted valve in the respective hold-up tanks.
5.4. **Specific items of investigation.**

Apart from the basic and detail engineering of the pilot plant a number of items are investigated in particular and reported in this subchapter. These items are:

1. **Hydraulic behaviour.**
   
   For the fluidized bed heat exchanger it is of great importance to know the hydraulic behaviour of the fluidized bed and the mass flow distribution over the parallel condenser tubes.

2. **Automatic control of an MSF/FBE.**
   
   The vertical evaporator with a fluidized bed condenser has a flexible operation mode and due to absence of interaction between interstage brine mass flow and temperature distribution over the stages an MSF/FBE is suitable for a high degree of automatic control as well for automation in start and shut-down procedures.

3. **Constructional changes.**
   
   The main purpose of the pilot plant as stated in subchapter 5.1 is the necessity of operational experience. Inevitable some adjustments of design have to be made after operational experience. Also results of simultaneous carried out investigations with other test modules are as far as possible to be implemented in the pilot plant. In some cases long term experience is necessary before changes in construction can be proposed.

4. **Non-condensables removing.**
   
   In a vertical Multi Stage Flash/Fluidized Bed Evaporator special attention is to be paid to the venting of non-condensable gases because of the specific configuration of flash stage and condenser cross section area in conjunction with the small stage height.

5. **Heat transfer.**
   
   With regard to heat transfer a confirmation of heat transfer data with data from other test modules is important. It is also necessary to establish long term values for heat transfer and tube fouling data. For the first mentioned reason a special operation mode was developed enabling temperature measurement in any of the condenser tubes during operation of the pilot plant.
5.4.1. Hydraulic behaviour.

The hydraulic behaviour of the fluidized bed means the pressure distribution over the various components of each of the fluidized bed condenser tubes and the influence of this pressure distribution on the interaction of a large number of parallel operating fluidized bed tubes. Stable operation of multiple fluidized bed can be obtained by a pressure drop over a throttling device at the inlet of the fluidized bed tubes.

According to KLAREN [5.8] stable operation of a bundle can be achieved if all individual beds satisfy the equation:

\[
\frac{\Delta p_{\text{thr}}}{\Delta p_{\text{w}}} \left| \text{operation} \right| > \frac{1}{2n} \left( \frac{E_x}{P} - 1 \right) \tag{5-1}
\]

where:

\( \Delta p_{\text{thr}} \) = pressure loss over the throttling device at the tube inlet
\( \Delta p_{\text{w}} \) = pressure loss over the bed due to the weight of the bed
\( n \) = empirical factor = 2.39
\( E_x \) = expansion factor of the fluidized bed \( \left( \frac{\text{fluidized bed length}}{\text{fixed bed length}} \right) \)
\( P \) = volume fraction of the particles in a fixed bed (0.60)

The constructed throttling device is shown in figure 5-11.

**FIGURE 5-11.**
Throttling device of pilot plant with detail of original design.
The pressure loss over this device is caused by the wall friction exerted on the liquid passing the relatively narrow inlet tube. A collision plate is used to converse the kinetic energy into heat. This plate also distributes the liquid evenly over the cross sectional area of the tube.

The expression for the pressure loss over this type of throttling device reads:

$$\Delta p_{\text{thr}} = \left( \zeta + \frac{L}{d} \right) \frac{\rho}{\rho_1} u_{\alpha,s} \left( \frac{D}{d} \right)^{0.4}$$  \hspace{1cm} (5-2)

where:

- $\zeta$ = pressure loss coefficient for the combined entrance losses and discharge losses of the throttling device tubes
- $L$ = length of the throttling device tubes
- $d$ = inner diameter of the throttling device tubes
- $\rho$ = density of the liquid
- $u_{\alpha,s}$ = superficial velocity of the fluidized bed
- $D$ = inner diameter of the fluidized bed tube,

while the wall friction factor is determined by the Blasius equation:

$$\lambda = (100 \, \text{Re})^{-0.25}$$  \hspace{1cm} (5-3)

for Reynolds numbers not exceeding $10^5$.

The minimum required pressure drop over the throttling device follows from equation (5-1).

Once the fluidized bed tube inner diameter, the particle size and the porosity of the fluidized bed have been selected the remaining two independent variables for the pressure loss as per equation (5-2) are:

- inner diameter of the throttling device tube
- length of the throttling device tube.

The relation between the pressure drop over a throttling device tube of one meter length and the inner diameter of the throttling device tube is presented in figure 5-12. The relations between the pressure drop and the length of the throttling tubes for various diameters are illustrated in figure 5-13.
Optimization of the length of the throttling device tube versus the inner diameter is governed by the following mechanisms:

- large inner diameter reduces the risk of blocking of the inlet by foreign objects;
- smaller inner diameter reduces the total column height.

Balancing of the importance of the above influences will lead to selection of length and inner diameter of the tubes of the throttling device.

For this plant a pressure loss over the throttling device of 28000 \( \frac{N}{m^2} \) was chosen. This had to be achieved by a throttling device at the tube inlet where only a superficial velocity of the liquid in the tube of approximate 0.14 m/s is available.

The throttling device of the pilot plant was selected with an inner diameter of 3 mm and a length of 750 mm.

In practice it is not possible to manufacture a large number of simple throttling devices with identical pressure loss coefficients. This means that the pressure drop-mass flow-curves for the various fluidized beds will generally not coincide. The consequences of dissimilar pressure loss coefficients are differences in fixed bed height.
Making use of the right throttling device it is possible to realize a kind of automatic "trimming" of the beds. If the mass flow through the bundle is gradually increased, the beds which already have reached their maximum fluidization height will spill particles to those beds which have not yet fully expanded. While the former beds remain stable the latter increase their fluidization height until they also reach the outlet water box. In the bundle of the MSF/FBE a number of throttling devices have different values. Table 5-2 informs about the safety factors found in the bundle of the MSF/FBE after a number of stability tests were performed.

<table>
<thead>
<tr>
<th>tube identification</th>
<th>fixed bed height</th>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>row # number</td>
<td>(cm)</td>
<td>total pressure drop $\Delta p_w + \Delta p_{thr}$</td>
<td>bed pressure drop $\Delta p_w$</td>
<td>orifice pressure drop $\Delta p_{thr}$</td>
<td>ratio $\frac{\Delta p_{thr}}{\Delta p_w}$</td>
<td>theoretical ratio $\frac{1}{2n}\left(\frac{\rho}{\rho_x} - 1\right)$</td>
<td>safety factor $f$</td>
</tr>
<tr>
<td>4 # 12</td>
<td>207</td>
<td>45000</td>
<td>18905</td>
<td>26094</td>
<td>1.38</td>
<td>0.67</td>
<td>2.06</td>
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<td>45000</td>
<td>17079</td>
<td>27920</td>
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<td>0.76</td>
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<td>14339</td>
<td>30660</td>
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<td>0.95</td>
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</tr>
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<td>45000</td>
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<td>2.31</td>
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<td>2.73</td>
<td>1.17</td>
<td>2.33</td>
</tr>
</tbody>
</table>

**TABLE 5-2.**

**Pressure drop data of fluidized bed and throttling device.**

During the initial period of operation with the fluidized bed heat exchanger problems occurred. This may be explained by figure 5-14 which shows two parallel operating fluidized beds, each containing an equal weight of particles of the same size and type. The extra pressure drop at the point of transition from fixed bed to fluidized bed is different for each bed. The small irregularity in the pressure drop is due to a slight sticking of the fixed bed in the tube. This difficulty is avoided by increasing the entrance pressure loss of each bed by means of a throttling device at the bed inlet.
FIGURE 5-14.
Pressure drop curves of fluidized bed tube and throttling device.

The great irregularity in the pressure drop is caused by a serious sticking of the fixed bed. For the first bed the pressure drop $\Delta p_1$ already corresponds with its maximum value at the transition point of the fluidized bed. A further increase of the pressure drop to $\Delta p_2$ forces the first bed to move through the unstable region of its pressure drop-mass flow characteristic. Now the first bed is fluidized to a large extent while the second bed is still fixed. The first bed will reach the outlet box and spill particles to the second bed which has not expanded yet. The second tube will be filled completely with particles and the first tube will therefore blow out its particles.

The above mentioned problem is caused by the effect of a slight corrosion of the aluminium-brass tubes. These tubes had been stored in the open air for about 1½ year. The difficulty is solved by emptying the bundle and filling it again with much less particles.

In a period of 24 hours the bundle was cleaned by the particles itself and filled with the charge of particles necessary for a porosity of 0.85. The procedure for charging the parallel condenser tubes is as follows: the mass flow through the condenser tubes is raised to a level where almost no particles can remain in the condenser tubes. Then the amount of particles is charged in the outlet box. The fluidization in this box is still in a fixed state. Then the mass flow is decreased and the tubes will fill themselves with the right amount of particles. Several experiments have been carried out to establish a stable hydraulic behaviour of the bundle.
A special experiment reported below was performed to find out whether a great number of start and shut-down procedures has an influence on the distribution of the particles over the bundle or not. The distribution of particles over the various tubes in the bundle can be located by measuring the fixed bed height in each of the tubes at zero mass flow through the bundle. A special precaution start procedure was performed and this procedure is elucidated in figure 5-15.

![Graph showing pressure drop vs mass flow](image)

**FIGURE 5-15.**

*Figure explaining the various points in start-up and shut-down procedure of the fluidized bed.*

The mass flow is raised to point I and held for about 2 minutes to afford the fixed bed to loosen. Then the mass flow is increased to point II and in a few minutes the fixed bed will expand. At last the mass flow will be increased until it reaches the nominal value where the fluidized bed extends into the outlet box (point III).

The shut down procedure is easier and consists of the points IV and V. Mass flow IV is introduced to afford the excess of particles in the water box to move from the outlet box into the fluidized beds. After about 2 minutes the mass flow is reduced to zero (point V or 0).

The above mentioned procedure was carried out more than 50 times. Finally all fixed bed heights were measured and compared with the initial values. No differences of any importance were found. A change of fixed bed height due to a great number of start and shut-down procedures is therefore negligible.
5.4.2. Automatic control of an MSF/FBE.

MSF/FBE operational results and MSF/FBE features with regard to start-up and operation caused a recognition of the possibilities for high degree automatic control of an MSF/FBE.

Major reasons for possible automatic control for an MSF/FBE are:

A. "Once-through" design.
An operation mode where brine is not recycled through the plant is called "once-through". The "once-through" version in Multi-Stage Flash Evaporation is recognized as the most preferable design and operation mode because of its simplicity. In general, evaporator design are forced to be of the recirculation mode because of chemical dosing and other control measures. As an MSF/FBE can operate at elevated temperatures without dosing of any chemical as described in Chapter 3 it is understood that "once-through" design and operation mode is possible for an MSF/FBE.
The "once-through" mode makes an automatic control more feasible because of absence of an independent reject cycle with its pump and separate brine concentration and blow-down control.

B. Major drawback of a conventional (horizontal) evaporator is the unfavourable interaction between interstage brine mass flow and actual temperature distribution over the various stages of the evaporator. With equal brine levels in consecutive stages mass flow one stage to another is only based on differences in saturation pressure in the stages.
Different brine levels in consecutive stages will either increase or decrease the total available pressure drop. This interaction shows that a change of available pressure difference leads to a change of interstage mass flow or to undesirable brine heights.
A vertical evaporator is different in this respect because of two reasons:

* The hydrostatic pressure of the brine height is added to the pressure difference resulting from the saturation temperatures in the stages.
Figure 5-16 shows the breakdown of the total available pressure drop over the stages, based on a saturation temperature drop of 4 °C.

**FIGURE 5-16.**

Impression of pressure drop for brine transport in a vertical evaporator.

Figure 5-17 elucidates the differences in interstage wall sealing and contributing brine height in an MSF/FBE. Especially in the last stages with lower temperatures the favourable influence of the gravity forces is obvious.

**FIGURE 5-17.**

Differences between conventional and vertical evaporator with respect to interstage wall sealing and effect of brine level.
Since the mass flow through the fluidized bed is to be kept at a nominal value for homogeneous fluidization over the total length of the condenser tubes it was necessary to provide additional cross area for conditions of significant lower pressure drop per stage. Once installed such a device as the siphons mentioned in subchapter 5.2.1, it is obvious that favourable consequences with regard to flexibility of interstage mass flow are gained.

**FIGURE 5-18.** Hierarchical control scheme for automated operation.

The rapid development and implementation of solid state electronic equipment and advanced "chip"-development makes it possible to simplify control equipment and bring an automatic control system within economic feasibility.

For the Delft pilot plant an automatic control system was evaluated by BROEKENS [5.2] and figure 5-18 illustrates the hierarchical control scheme for automatic start up, operation as well as shut down procedures for an MSF/FBE.
5.4.3. Constructional changes.

5.4.3.1. Flash device.

In the evaporator, brine is heated under pressure to an elevated temperature and discharged through five stages each at a reduced saturation pressure compared with the previous stage. Brine entering a stage flows through an orifice and flashes in a flash device named flash basket. BENJAMIN and MILLER \[5.3\] demonstrated that flashing liquid through an orifice can be treated as non-saturated water. The mass flow through the orifice follows from the pressure drop and reads:

\[
\phi_1 = \psi_1 \rho_1 A_1 \left( \frac{2 \Delta p}{\rho_1} \right)^{0.5} \tag{5-4}
\]

where:

- $\phi_1$ = mass flow through the orifice
- $\psi_1$ = discharge coefficient
- $\rho_1$ = density of the liquid through the orifice
- $\Delta p$ = total pressure difference over the orifice
- $A_1$ = cross area of the orifice

with:

\[
\Delta p = (P_n - P_{n-1}) + \rho_1 g_n H_n \tag{5-5}
\]

where:

- $P_n$ = saturated pressure in stage $n$
- $H_n$ = brine height in stage $n$
- $g_n$ = acceleration of gravity

As stated in Chapter 4 due to the gradual evaporation of the brine and the highly turbulent films descending from the chutes, the liquid is cooled by surface evaporation in a very efficient way and the interstage brine orifice should have a slit form.

Mainly because of constructural layout the interstage brine orifice for the 50 t/day pilot plant had to be cylindrical. After a few test runs at maximum operation temperature it became obvious that a serious atomization occurred in the first evaporator stages.
This atomization had mainly two reasons:

a. The constructed flash basket as shown in figure 5-19 had a large volume allowing for almost all evaporation within the basket. The vapour flow produced in the basket follows from:

\[
\phi_{vd} = \frac{\phi_{mb} c_{p1} \Delta \theta}{h_{lg} \rho_v}
\]

(5-6)

with:

- \( c_{p1} \) = specific heat
- \( \Delta \theta \) = saturated temperature difference between two stages
- \( h_{lg} \) = latent heat of evaporation
- \( \phi_{mb} \) = mass flow from stage \( n \) to stage \( n+1 \)

**FIGURE 5-19.**

*Original design of flash basket.*
This produced vapour volume had to pass the narrow spaced wire-mesh on entering the vapour space of the evaporator stage. The high velocity of the two-phase mixture caused an atomizing of the brine entering the evaporator stage.

b. The brine velocity in the orifice can be calculated with equation (5-4) and (5-5) and reads:

\[
V_{1\text{o}} = \psi \left( \frac{2 \Delta P}{\rho_1} \right)^{0.5}
\]  

\hspace{1cm} (5-7)

The velocity of the brine through the cylindrical orifice is high due to the large saturation temperature difference over the first evaporator stages.

The kinetic energy of the brine caused a deflection of the evaporating brine on the bottom plate of the flash basket and contributed to the atomization of the brine entering the evaporator stage.

**FIGURE 5-20.**  
Screen plate installed to reduce entrainment.
To avoid the atomized brine entering the condenser section a screen plate construction as shown in figure 5-20 was installed in the stages. The screen plate causes a change in direction of the vapour flow and small droplets are caught by the thin liquid film on the screen plate and the outer part of the condenser shell.

Distillate quality improved from approx. 50 ppm T.D.S. to approx. 12 ppm T.D.S. and it was decided at that time to design a new flash device which also could be based in results of simultaneous carried out research on flash stage design as discussed in Chapter 4. After one year of operation a new flash device as shown in figure 5-21 was installed in all stages.

**FIGURE 5-21.**
Improved design of flash device.

This new device was instructed to serve two purposes:

I. To minimize atomization of the brine in this specific pilot plant with its relatively large saturation pressure differences.

II. To prove the results of continued research on flash volume requirements which showed the vertical flash process could reach equilibrium in much smaller volumes. With the new flash device the actual process is terminated within the cylinder which means the remaining volume of the original flash stage is non-active and only carries the vapour to the condenser.
The results obtained with the new device are summarized:

A. Low entrainment resulting in a product water quality of less than 1 ppm T.D.S. with untreated Schie canal water as feed to the system.

B. A good approach to equilibrium in a "flash stage" with a specific volume requirement of only 0.05 m³/ton distillate per hour, based on the high temperature stage.

5.4.3.2. Inspection and measurement features.

Pilot plant operation is characterized by requirements for measured values and additional information normally not or not easy available from commercial plants. To investigate the heat transfer performance of the fluidized bed heat exchanger two additional connecting pipes with valves were installed between the upper waterbox and the vapour space of the first evaporator stage. Figure 5-22 shows measurement feature.

FIGURE 5-22.
Measurement feature to inspect the fluidized bed.
The regular brine flow goes through valve 1 and the upper water box is closed. Partially opening of valve 2 allows for atmospheric pressure of the upper waterbox whereafter the water box can be opened. Diverting the total mass flows from valves 1 and 2 to valves 2 and 3 one allows for liquid level decrease in the upper water box. Finally valve 1 and 2 are closed and the total mass flow enters the first stage through valve 3.

Low liquid level on the upper tube plate allows for inspection of the fluidization over the total pattern of the bundle. The inspection was performed every month. At the same time it becomes possible to measure the bulk temperature of the liquid in the condenser tubes at any level. With a special developed thin thermocouple probe it was possible to investigate the heat exchanger performance over all stages and total length enabling the investigators also to obtain information about the blanketing by non-condensables in the lower stages.

5.4.3.3. Throttling device.

As stated in subchapter 5.4.1 stable operation of this multiple fluidized bed is obtained by a pressure drop over the throttling device required at the inlet of the fluidized bed tubes. The pressure difference over this throttling device is caused by the wall friction exerted on the liquid passing the relatively narrow inlet tubes. A collision plate was used to convert the kinetic energy into heat while it also distributes the liquid evenly over the cross area of the condenser tube. The total throttling device consists of 288 stainless steel (AISI 316) tubes with an inner diameter of 3 mm and a total length of 750 mm.
The construction of the original installed throttling device is shown in figure 5-23. Leakage between the support plate and the tube plate caused a severe reduction of the initial provided safety factor on the individual pressure drop. After an attempt to seal the support plate on the tube plate, which attempt failed, it was decided to design a new attachment for the throttling device. The new construction is shown in figure 5-24 while subchapter 5.4.1 deals with the hydraulic behaviour of this system.

FIGURE 5-23.
Originally installed throttling device.

FIGURE 5-24.
Improved design of throttling device.
5.4.3.4. Stirring device.

In a bundle of parallel fluidized beds some particles of each condenser tube reach the outlet water box. The particles in the outlet water box are almost in a fixed state and during operation this excess of particles enables the tubes of the condenser to remain filled with fluidized particles in case of a reduction of the mass flow through the condenser tubes.

A layer of particles with a low porosity in the upper water box can act as a filter and bridging as shown in figure 5-25 above the outlet opening could take place.

To avoid any possibility of disturbing the homogeneous fluidization in all condenser tubes a stirring device as shown in figure 5-6 was originally installed.

**FIGURE 5-25.**
Impression of bridging phenomenon above fluidized bed tubes.

After one year of operation with the plant without this device actually working it was decided to remove the stirring device completely to simplify the design of the evaporator.

The upper waterbox was slightly modified by welding a circular bended stainless steel strip in the dead corner between the tube plate and the waterbox shell in order to facilitate a low porosity fluidized bed in the upper waterbox.

- 5.31 -
5.4.4. Non-condensables venting.

Venting of non-condensable gases present in the vapour flow approaching the condenser bundle is important because of the jeopardizing influence on the heat transfer coefficient as explained below.

Non-condensable gases are present in the vapour entering the condenser bundle of the MSF/FBE due to release of gases from the feed during heating, decompositions of bicarbonates and air leaking into the evaporator and condenser stages.

Condensation of the vapour on the heat exchanger surface enhances the concentration of non-condensable gases in the zone around the heat exchanger tube. The partial vapour pressure of the non-condensable gases will therefore increase. Neglecting the kinetic energy of the vapour flow in the condenser bundle results in a constant total pressure in the bundle.

With a constant total pressure in the vapour space of the condenser bundle and an increased partial vapour pressure of the non-condensable gases this results in a reduction of the partial vapour pressure of the condensing steam. The condensation temperature of the steam will also lower and this results in a reduced driving force for the heat transfer to the tube wall and leads to a reduced efficiency of the heat exchanger.

RENKER [5.4] developed an emperical relation for the influence of the presence of non-condensable gases in condensing steam at the inside of a tube. Although the non-condensable gases appear on the outside of the tubes in an MSF/FBE, below the relation is elaborated to indicate the influence of non-condensables in condensing steam in heat exchangers:

\[
\alpha = \frac{\alpha_C}{1 + \frac{\alpha_C}{C_1 \left( \frac{P_1}{P_2} \right)^{C_2}}} 
\]  \hspace{1cm} (5-8)

where:

\( \alpha \) = condensation heat transfer coefficient with the presence of non-condensable gases

\( \alpha_C \) = condensation heat transfer coefficient without the presence of non-condensable gases

\( P_1 \) = partial vapour pressure of steam

\( P_2 \) = partial vapour pressure of non-condensable gases
with:

\[
C_1 = 0.0105 \frac{h_{lg} D_v P_{t_m}}{(T_{v_m} - T_w) R_1 (T_{a_v} + 273) D_i} \text{Re}^{0.95} \text{Sc}^{0.92} \quad (5-9)
\]

and constant factor:

\[
C_2 = 0.38 \left( \text{Sc}_v \right)^{-0.27} \quad (5-10)
\]

where:

- \( h_{lg} \) = latent heat of evaporation
- \( D_v \) = coefficient of diffusion of steam in non-condensable gases
- \( P_{t_m} \) = total vapour pressure of the mixture
- \( T_{v_m} \) = vapour temperature of the mixture
- \( T_w \) = temperature at the surface of the heat exchanger tube
- \( R_1 \) = gas constant for steam
- \( T_{a_v} \) = temperature of the steam
- \( D_i \) = inner diameter of the heat exchanger tube

and:

\[
\text{Re} = \frac{\rho_{v_m} U_{v_m} D_i}{\eta_{v_m}} \quad (5-11)
\]

\[
\text{Sc}_v = \frac{\eta_{v_m}}{\rho_v D_v} \quad (5-12)
\]

where:

- \( \rho_{v_m} \) = density of the vapour mixture
- \( U_{v_m} \) = velocity of the vapour mixture
- \( \eta_{v_m} \) = dynamic viscosity of the vapour mixture
- \( \rho_v \) = density of the steam
The results of the work of RENKER translated into actual situation of the evaporator design is shown in figure 5-26. This figure has been calculated for mixtures of steam vapour and air at a total pressure of 2 bar and based on a temperature difference of 4 °C between mixture temperature and wall temperature.

The presented curves for the heat transfer coefficient versus the molecular weight fraction are for mixture velocities between 5 and 25 m/s. As can be seen in figure 5-26 the heat transfer coefficient decreases rapidly with an increase of the amount of non-condensable gases. With low velocity of the vapour and non-condensables mixture the decrease of the heat transfer coefficient becomes even more pronounced.

**FIGURE 5-26.**

Influence of non-condensable gases upon heat transfer coefficient.
5.4.4.1. Non-condensables venting system in the pilot plant.

Multi Stage Flash/Fluidized Bed Evaporators basically will be designed for a large number of evaporator stages and low specific energy consumption per unit product water. Consequently driving temperature differences for heat transfer will be small. As also can be seen from the relation derived by Renker, adequate venting of non-condensables is required for the design of the MSF/FBE. Venting of non-condensable gases in the pilot plant, subject of the present investigations, is performed by cascading the non-condensable gases from stage to stage and finally to a two stage steam driven ejector set. As the non-condensable gases have to be removed from the place in the condenser bundle at the lowest pressure of the gas mixture, an opening to the next stage is provided in the middle of the condenser bundle at the bottom of the stage. The vapour flow is controlled by calibrated orifices in the openings between stages. Finally the cumulative amount of non-condensable gases is removed from the bottom of the last stage condenser bundle.

5.4.4.2. Operational experience.

During the initial period of operation with the pilot plant it was observed that the design saturation pressure and temperature in the lower stages could not be reached. Investigations learned that due to the large size of manholes for inspection and the applied sealing gaskets as well as due to the number of nozzles for measurements the amount of inleaking non-condensables was higher than designed for. The originally installed orifices in the centre of the condenser bundle were inaccessible for enlargement of the orifice size. It was therefore decided that, in addition to the replacement of sealing gaskets of the manholes, another way of venting of the excessive quantity of non-condensables had to be found. Four condenser tubes around the centre of the condenser bundle as shown in figure 5-27 are sacrificed for this purpose. In each of the tubes a hole was drilled at the specified level and as shown in figure 5-28. The openings of the four sacrificed tubes were sealed in the upper tube plate and the entrance of each of the tubes was connected with the two stage non-condensables venting ejector system. Although the heat transfer coefficients were improved due to the venting of the excessive non-condensable it is understood that in future designs considerable attention have to be paid to this subject.
It is advised to design non-condensable venting openings in such a way that access to the orifice is guaranteed in case after initial operating experience adjustments of the venting system are required.

**FIGURE 5-27.**

Sacrificed condenser tubes for improvement of venting.

References [5.5] and [5.6] describe measurements carried out on the pilot plant subject of this chapter and design and use of the required drilling tool.

BRINK [5.7] has commented the non-condensables venting system of the demonstration MSF/PBE as described in subchapter 5.5.1.
5.4.5. Heat transfer coefficient.

The expression for the overall heat transfer coefficient \( K \), based on the outer heat exchange surface of the condenser tubes reads:

\[
K = \frac{1}{\frac{1}{\alpha_C} + \frac{D_o}{2\lambda_t} \ln \frac{D_o}{D_i} + \frac{D_o}{D_i} \frac{1}{\alpha_{w,l,i}} + f_o \frac{D_o}{D_i} f_i}
\]  

(5-13)

where:

- \( \alpha_C \) = heat transfer coefficient for condensation
- \( \lambda_t \) = heat conductivity of the tube material
- \( \alpha_{w,l,i} \) = wall to liquid heat transfer coefficient based on the inner diameter
- \( D_o \) = outer diameter of the heat transfer tube
- \( D_i \) = inner diameter of the tube
- \( f_o \) = fouling factor for the outer surface of the condenser tube
- \( f_i \) = fouling factor for the inner surface of the condenser tube
As also reported in references [5.1] and [5.8] the correlation of RÜCKENSTEIN and SHORR [5.9] has been used in the design stage for the computation of the wall to liquid heat transfer coefficient $\alpha_1$, i.e.:

$$\text{Nu} = \frac{\alpha_1 d_p}{\lambda_1} = (0.067 \text{ Re}^{0.237} \Pr^{0.333} \text{ Ar}^{0.522})$$  \hspace{1cm} (5-14)

where:

- $d_p$ = fluidized particle diameter
- $\lambda_1$ = heat conductivity of the liquid

For the calculation of the heat transfer coefficient $\alpha_C$ the correlation of NUSSELT [5.10] for condensation on a vertical wall and laminar flow of the condensate film is used:

$$\alpha_C = 0.943 \left( \frac{\rho_C^2 \eta_C \Delta T_C}{H \lambda_C \eta_C h_{lg}} \right)^{0.25}$$  \hspace{1cm} (5-15)

where:

- $H$ = undisturbed length of the condensate film
- $\Delta T_C$ = temperature difference over the condensate
- $\rho_C$ = density of the condensate
- $\lambda_C$ = heat conductivity of the condensate
- $\eta_C$ = dynamic viscosity of the condensate
- $h_{lg}$ = latent heat of evaporation
- $g_n$ = acceleration of gravity

As fouling factor for the outer surface of the condenser tube the normal value usual for flash evaporators is used.

The fouling factor for the inner surface is only a fraction of the fouling factor of the outer surface.

The inner side of the tube is kept clean due to the abrasive action of the fluidized particles on the wall of the heated surface. The abrasive action removes the initially formed fouling on the wall of the heated surface. Even a scale removing capacity can be attributed to the fluidized particles. No insulating layer is formed on the inner side of the wall which should greatly impede the flow of heat. Therefore the wall to liquid heat transfer coefficient is not only enhanced compared to a conventional heat exchanger but also these high values are maintained.
Table 5-3 presents the design values of the pilot plant in respect of temperatures, mass flows, porosities, calculated heat transfer coefficients and dimensions.

<table>
<thead>
<tr>
<th>stage</th>
<th>brine temperature (^{\circ}C)</th>
<th>distillate temperature (^{\circ}C)</th>
<th>logarithmic difference (^{\circ}C)</th>
<th>end-saturating temperature difference (^{\circ}C)</th>
<th>distillate production (\text{kg/h})</th>
<th>brine flow (\text{kg/h})</th>
</tr>
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<tr>
<td>1</td>
<td>86.05</td>
<td>85.94</td>
<td>8.042</td>
<td>2.983</td>
<td>465.9</td>
<td>18285.9</td>
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<td>72.54</td>
<td>72.30</td>
<td>8.322</td>
<td>3.417</td>
<td>433.2</td>
<td>17820.1</td>
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<tr>
<td>3</td>
<td>59.53</td>
<td>59.15</td>
<td>8.576</td>
<td>3.724</td>
<td>401.6</td>
<td>17386.9</td>
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<tr>
<td>4</td>
<td>47.09</td>
<td>46.38</td>
<td>9.081</td>
<td>4.122</td>
<td>370.4</td>
<td>16985.3</td>
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<tr>
<td>5</td>
<td>35.32</td>
<td>34.67</td>
<td>9.534</td>
<td>4.810</td>
<td>338.8</td>
<td>16614.9</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>stage</th>
<th>porosity</th>
<th>heat transfer coefficient inside (\text{kJ/m}^2\text{C}\text{h})</th>
<th>heat transfer coefficient outside (\text{kJ/m}^2\text{C}\text{h})</th>
<th>heat transfer coefficient overall (\text{kJ/m}^2\text{C}\text{h})</th>
<th>heat transfer surface stage</th>
<th>height (\text{m})</th>
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<tr>
<td>1</td>
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<td>28857</td>
<td>32480</td>
<td>11730</td>
<td>11.3</td>
<td>0.8</td>
</tr>
<tr>
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<td>0.797</td>
<td>26108</td>
<td>30884</td>
<td>10971</td>
<td>11.3</td>
<td>0.8</td>
</tr>
<tr>
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<td>23329</td>
<td>29137</td>
<td>10154</td>
<td>11.3</td>
<td>0.8</td>
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<tr>
<td>4</td>
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<td>20523</td>
<td>27256</td>
<td>9278</td>
<td>11.3</td>
<td>0.8</td>
</tr>
<tr>
<td>5</td>
<td>0.825</td>
<td>17765</td>
<td>25278</td>
<td>8362</td>
<td>11.3</td>
<td>0.8</td>
</tr>
</tbody>
</table>

**TABLE 5-3.**

Design values of the pilot plant.

Table 5-4 shows the heat transfer data computed with the equations (5-13), (5-14) and (5-15). In the second part of this table the results of experimental values of these data are presented, as far as they could be measured with the equipment installed on the MSF/FBE.

**COMPUTED**

<table>
<thead>
<tr>
<th>stage</th>
<th>average temperature inside tube (^{\circ}C)</th>
<th>Re</th>
<th>Pr</th>
<th>Ar (\times 10^{-6})</th>
<th>heat transfer coefficient inside (\text{kJ/m}^2\text{C}\text{h})</th>
<th>heat transfer coefficient outside (\text{kJ/m}^2\text{C}\text{h})</th>
<th>heat transfer coefficient overall (\text{kJ/m}^2\text{C}\text{h})</th>
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<tbody>
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<td>8362</td>
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**EXPERIMENTAL VALUES**

<table>
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<th>stage</th>
<th>average temperature inside tube (^{\circ}C)</th>
<th>Re</th>
<th>Pr</th>
<th>Ar (\times 10^{-6})</th>
<th>heat transfer coefficient inside (\text{kJ/m}^2\text{C}\text{h})</th>
<th>heat transfer coefficient outside (\text{kJ/m}^2\text{C}\text{h})</th>
<th>heat transfer coefficient overall (\text{kJ/m}^2\text{C}\text{h})</th>
</tr>
</thead>
<tbody>
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<td>30560</td>
<td>11030</td>
</tr>
<tr>
<td>2</td>
<td>55.0</td>
<td>518</td>
<td>3.42</td>
<td>0.435</td>
<td>23377</td>
<td>29360</td>
<td>10300</td>
</tr>
<tr>
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<td>462</td>
<td>4.27</td>
<td>0.291</td>
<td>20975</td>
<td>27000</td>
<td>9570</td>
</tr>
<tr>
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<td>18890</td>
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<tr>
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<td>6.40</td>
<td>0.141</td>
<td>17222</td>
<td>20500</td>
<td>7700</td>
</tr>
</tbody>
</table>

**TABLE 5-4.**

Computer and measured values of the pilot plant heat transfer.
FIGURE 5-29.
Calculated and measured temperature diagram.

Figure 5-29 shows the temperature diagram of this MSF/FBE. There is a difference between the temperature lines as concluded from computer calculations and the temperature lines found during operation of the MSF/FBE. The difference between the calculated values and the measured values of the heat transfer coefficients can be explained as follows:

- Measurements of wall to liquid heat transfer coefficients as described in Chapter 2 indicated lower values than predicted with the correlation of Rickenstein and Shorr as also reported by SCHOEN [5.11] and DE GROOT [5.12]. This results in combination with the influence of non-condensable gases as described below in lower actual transfer coefficients than designed for.

- The described plant is of the once-through type and no deaerator is used. Dissolved gases are removed from the brine in the hottest chamber by means of a connection places in the middle of the condenser bundle to the next chamber.
FIGURE 5-30.
Assembled condenser bundle showing brine and distillate siphon tube as well as the interstage walls.
Left at the picture the bottom tube plate of the fluidized bed heat exchanger (without throttling device).
FIGURE 5-31.
Picture of final assembly of the condenser bundle, brine and distillate siphons and interstage wall into the shell.
The white rubber profile as explained in figure 5-3 around the interstage wall is clearly shown.
The two-stage steam jet ejector is connected to the distillate sump and removed the non-condensables. All non-condensables from both feed and inleakage cascade down through the stages to the distillate sump. The volume of this amount of gases increases remarkably due to the lower pressures in the cold end of the plant. The total volume of gases is sucked off through the condenser bundle to the connecting pipe line. A certain blanketing of the heat transfer surface by the non-condensables is a partial explanation of the lower values for the heat transfer coefficients in the lower stages as described in subchapter 5.4.4.

Finally illustrations as presented in figures 5-30 and 5-31 show design details of the assembled condenser bundle of the pilot plant such as, brine siphon tubes, distillate siphon tubes, the concentric condenser bundle and the interstage walls, as described in subchapter 5.2.1.

![Diagram of the pilot plant process](image)

**FIGURE 5-32.**

Flow scheme of the pilot plant.
(Data on top if the figure corresponds with numbers as identified on the scheme).
Figure 5-31 of the final assembly clearly shows the rubber profile for interstage brine sealing and the slits of the non-flooding precaution provided for by the brine siphon tubes. These siphon tubes also act as structural support for the interstage walls.

Figure 5-32 shows the flow scheme of the 50 t/day pilot plant also for comparison reasons with the schemes presented in subchapter 5.5.

FIGURE 5-33.

Overall view of the complete pilot plant.
Figures 5-33 and 5-34 provide overall views of the pilot plant in operation in the Central Boilerhouse of Delft University of Technology.

FIGURE 5-34.
Front view of the pilot plant.
5.5. Design developments.

In this subchapter a brief review is presented on the design development of the Multi-Stage Flash/Fluidized Bed Evaporator (MSF/FBE). The review is made in order to complete the work reported in this thesis enabling the reader to make a proper assessment of the entire development to date.

5.5.1. 500 m$^3$/day demonstration MSF/FBE.

After a period of cooperation between Delft University of Technology and the water distillation industry of the Netherlands, represented by ESMIL, the latter decided to increase the development efforts and to design, build and operate a 500 m$^3$/day Multi-Stage Flash/Fluidized Bed Evaporator (MSF/FBE) as a first step towards commercialisation of the MSF/FBE [5.13]. Basic reasons for this industrial development step are:

- further evaluation of the total MSF/FBE concept
- investigation of plant operators experience
- investigation of plant operating performance.

The actual decision to built this plant was made after a careful evaluation of the basic ideas and the successful operation of the 50 m$^3$/day MSF/FBE pilot plant at Delft University of Technology. Funds for realisation of this plant and its operation for a period of approximately two years were made available partly by the Dutch Government and partly by ESMIL. The plant site is located in Texel which is an island in the North of Holland and which obtains its power and drinking water from a combined power station and sea water distillation plant on the island itself. The existing conventional evaporator is of the multi-stage flash evaporator type (MSF). The main characteristics are:

- production = 125 m$^3$/h
- maximum brine temperature = 115 °C
- gain-ratio = 9.45 : 1.

The conventional plant consists of 35 recuperation stages, built in five vessels and a heat reject section of three stages, contained in the sixth vessel. The plant is of the so-called long-tube design and acid treatment of the feed is applied for the removal of bicarbonates.
The MSF/FBE demonstration plant operated parallel to the conventional flash evaporator; it used the intake facilities and discharge line of the conventional plant. As a consequence both evaporators use the same sea water quality which should enable a fair comparison of these plants. This is particularly interesting with respect to fouling, from which the conventional plant suffers seriously, while it was expected that the MSF/FBE demonstration plant would experience hardly any. Figure 5-35 shows an aerial view of the plant site and the location of the MSF/FBE demonstration plant on this site is next to the existing evaporator.

FIGURE 5-35.
Site of the MSF/FBE demonstration plant.
The demonstration plant has been erected on the area in the center of the picture and next to the conventional horizontal evaporator.
Figure 5-36 provides a front view of the MSF/FRB demonstration plant. The demonstration plant, of which the flow scheme is shown in figure 5-37 is basically designed as an "once-through" plant with the following main characteristics:

- production = 21.997 kh/h (500 m³/d)
- maximum brine temperature = 115 °C
- gain-ratio = 10.3 : 1.

The plant consists of 26 recuperation stages and a brine heater in one vessel. The cooling water pump takes the water from the cooling water outlet of the distillate coolers of the conventional flash evaporator after mixing with a fraction of the cooling water outlet of the heat reject section, in order to maintain a fairly constant cooling water inlet temperature.
This concept ensures a minimum of adjustments in the control of the installation with varying cooling water temperatures. The feed is pumped through the condenser tubes of the stages 26 to 1 and the brine heater. By passing the stages, the brine is heated by the condensing steam from the flashing brine and is given its final heating-up in the brine heater. Then it expands in stage 1, where part of it evaporates by flashing. The steam produced in the flash chambers flows into the condensers where it condenses on the tubes and the distillate formed is collected in a trough. Degraded in temperature, the brine moves to stage 2, in which a lower temperature and pressure exist and where the flashing and condensing process is repeated.

<table>
<thead>
<tr>
<th>1. BRINE</th>
<th>2. STEAM</th>
<th>3. CONDENSATE</th>
</tr>
</thead>
<tbody>
<tr>
<td>pressure</td>
<td>bar</td>
<td>2.6</td>
</tr>
<tr>
<td>temperature</td>
<td>°C</td>
<td>115.0</td>
</tr>
<tr>
<td>quantity</td>
<td>kg/h</td>
<td>164000</td>
</tr>
<tr>
<td>T.D.S.</td>
<td>ppm</td>
<td>25250</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>4. BLOW DOWN</th>
<th>5. FEED WATER</th>
<th>6. DISTILLATE</th>
</tr>
</thead>
<tbody>
<tr>
<td>pressure</td>
<td>bar</td>
<td>2.8</td>
</tr>
<tr>
<td>temperature</td>
<td>°C</td>
<td>32.7</td>
</tr>
<tr>
<td>quantity</td>
<td>kg/h</td>
<td>142000</td>
</tr>
<tr>
<td>T.D.S.</td>
<td>ppm</td>
<td>29381</td>
</tr>
</tbody>
</table>

**FIGURE 5-37.**
Flow scheme of the demonstration plant.
This continues through all 26 stages; then the brine is collected in a vessel and removed by the reject pump. The distillate produced in the stage cascades down from stage to stage and is finally collected in a vessel. From this vessel the distillate is pumped to distillate storage tanks after final check of its quality by conductivity measurement.

The condensate from the brine heater is also collected in a vessel from where it is returned to the storage tank after a check of its quality. Non-condensables are removed by a two-stage ejector system provided with two mixing condensers cooled by sea water.

For removal of non-condensables, the stages 1, 2 to 7 as well as the stages 8 to 26 are connected to a common venting line.

Table 5-5 shows the specification of the data of this demonstration plant.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>distillate production</td>
<td>22000 kg/hr</td>
</tr>
<tr>
<td>steam consumption</td>
<td>2135 kg/hr</td>
</tr>
<tr>
<td>steam pressure</td>
<td>2.56 bar</td>
</tr>
<tr>
<td>steam temperature</td>
<td>129.0 °C</td>
</tr>
<tr>
<td>gain-ratio</td>
<td>10.3:1</td>
</tr>
<tr>
<td>specific heat consumption</td>
<td>211 kg/kg</td>
</tr>
<tr>
<td>maximum brine temperature</td>
<td>115.0 °C</td>
</tr>
<tr>
<td>feed temperature</td>
<td>25.0 °C</td>
</tr>
<tr>
<td>blow-down temperature</td>
<td>32.7 °C</td>
</tr>
<tr>
<td>distillate temperature</td>
<td>30.9 °C</td>
</tr>
<tr>
<td>total number of stages</td>
<td>26</td>
</tr>
<tr>
<td>concentration of feed</td>
<td>25250 ppm TDS</td>
</tr>
<tr>
<td>concentration of blow-down</td>
<td>29381 ppm TDS</td>
</tr>
<tr>
<td>feed</td>
<td>164000 kg/hr</td>
</tr>
<tr>
<td>blow-down</td>
<td>142000 kg/hr</td>
</tr>
<tr>
<td>total number of condenser tubes</td>
<td>1557</td>
</tr>
<tr>
<td>condenser tube material</td>
<td>aluminium-brass</td>
</tr>
<tr>
<td>condenser tube diameters</td>
<td>I.D. 17.23 mm</td>
</tr>
<tr>
<td></td>
<td>O.D. 19.05 mm</td>
</tr>
<tr>
<td>porosity</td>
<td>0.77</td>
</tr>
<tr>
<td>diameter of fluidized particles</td>
<td>2.0 mm</td>
</tr>
<tr>
<td>density of particle material</td>
<td>2500 kg/m³</td>
</tr>
</tbody>
</table>

**TABLE 5-2.**
Pressure drop data of fluidized bed and throttling device.
The evaporator vessel contains 26 stages and a brine heater. Each stage consists of a flash chamber and a condenser.

The integrated construction of the 26 stage condensers and the brine heater only requires two tube sheets, one water inlet chamber and one water outlet chamber. The total tube length over the tube sheets is approximately 12.4 m; this length is partitioned as follows:

- condenser tube length of the brine heater 0.60 m
- condenser tube length of the first stage 0.60 m
- condenser tube length of stage 2 to stage 23 (24 x 0.40) 9.60 m
- condenser tube length of stage 26 1.05 m
- miscellaneous; wall thickness of tube sheets and inter- 0.53 m section plates.

The very short condenser tube length per stage is one of the most typical characteristics of the MSF/FBE; it is caused by the good heat transfer performance of the fluid bed at very low liquid velocities in the condenser tubes.

In this design the condenser tube length per stage corresponds with the flash chamber height, except for stage 26 where the flash chamber height is somewhat less than the installed condenser tube length, i.e. 0.60 m versus 1.05 m.

The cross section of the recuperation vessel is a square with sides equal to 1.8 m. Approximately 40% of the area of this cross section is occupied by the condenser and the remaining part by the actual flash chamber. Figure 5-38 shows the various cross sections of a stage of the evaporator vessel.

The brine flashes through an adjustable slit after which it is diverted by a curved baffle plate made of stainless steel. This plate protects the mild steel wall from being struck by the flashing brine. Brine droplets in the vapour are removed by centrifugal effect. These are particularly exerted when the vapour flows from the flash chamber to the condenser through the passage openings which are shielded by simple flat plates that sufficiently overlap these openings.

This construction achieves sufficient entrainment separation to guarantee a distillate quality of approximately 40 ppm T.D.S. from brine with 40,000 ppm T.D.S. (total dissolved solids).

For better distillate qualities, conventional wire-mesh demisters should be applied. The MSF/FBE demonstration plant can also be equipped with such demister pads.
FIGURE 5-38.

Cross sections of the 500 m³/day demonstration plant.

One of the major problems to be solved for an MSF/FBE with a stage height of only 0.4 m is the level control of the brine in the stages during all modes of operation. Brine level control becomes particularly necessary during the start-up of the MSF/FBE, since the fluid bed condenser requires expansion of the fluidized solids over the total tube length for all modes of operation; this also means that the flow through the condenser tubes cannot be varied much.

As a consequence, during the start-up of the plant there is too much brine flow for the available pressure differences over the stage. This lack of driving force for the interstage brine flow should be compensated for by increasing the brine flow area between the stages. Besides during the start-up, brine levels in the stages also tend to rise when operating at part load, i.e. operation at normal flow and a limited flash range.
Therefore the MSF/FBE demonstration plant is equipped with a flow control method by means of interstage valves which lifting height is governed by the brine level in the highest temperature stage. Actually, two parallel-operating interstage valve systems made of stainless steel are mounted behind the curved baffle plate to reduce entrainment caused by the splashing of the flashing brine flow through these valves.

The condensers of the recuperation section and the brine heater contain 1557 condenser tubes placed in an equilateral triangular pattern with a pitch of 1.25 times the tube outer diameter.

The condenser have baffles to shield the cold section and to maintain sufficiently high vapour velocities to avoid stagnation of non-condensables in the condenser bundle.

The construction material for the evaporator is carbon steel, except for the first two stages which are made of stainless steel. The components in the remaining stages, which are subject to high brine velocities and/or high brine droplets impingement rates, are also made of stainless steel.

The condenser tubes are made of Aluminium-Brass while the tube sheets are manufactured from Copper Nickel alloy (CuNi 90/10). The outlet water chamber of the condenser bundle is made of stainless steel (316 L); the inlet water chamber of carbon steel provided with a rubber lining.

Each flash chamber contains four sight glasses for observation of the process, while each condenser has two sight glasses.

Access to the flash chambers is possible through two manholes per stage. Each stage contains a number of nozzles which can be used for inserting the necessary measuring equipment into the stage.

The demonstration MSF/FBE in the island of Texel has been in operation for approx. 12,000 hours in its contractual operation period under the above mentioned development program and has provided a major contribution for the development of commercial evaporators of the MSF/FBE design.

Scale-free operation of the fluidized bed heat exchanger upto the maximum operation temperature of 155 °C has been observed in its period of operation. Quality of the distillate produced in this evaporator, with its basic design of the evaporator stages in general in accordance with the results and conclusions of Chapter 4 of this thesis, proved to be predictable as per guidelines as specified in Chapter 4.
Heat transfer coefficients as resulting from measurements are confirmed to be lower than calculated with the correlation of Rückenstein as per equation (5-14), but slightly higher than found for particle diameter of 2 mm, porosity of 0.77 in Chapter 2 of this thesis. Further analysis of the heat transfer data are required to confirm the above found results, since only overall heat transfer coefficients could be established from the data obtained in the operation of the demonstration MSF/FBE.

The demonstration plant has been operated producing water for the Island of Texel in parallel with the existing conventional evaporator. The operation in parallel with a conventional evaporator by the same operators has proven to be an excellent opportunity to obtain operational and comparison information. The absence of an unfavourable interaction between interstage brine mass flow and actual temperature distribution over the various stage in an MSF/FBE as explained in subchapter 5.4.2 has clearly been shown in the operation. Start-up procedure on average took the operators less than three hours whereas the conventional evaporator requires a minimum of twelve hours. Flexibility of operation has also shown to be an advantage of the MSF/FBE. Load variations even up to zero output could be introduced without any interruption of production proving excellent dynamic behaviour features. The experience with the conventional evaporator had evolved to a constant load situation (or off) because of high sensitivity load variation invariably leading to loss of production and long period of sometimes days before stable output and settings could be obtained again. Shut of procedure of the demonstration MSF/FBE can be reported as comparable with a conventional evaporator.

The results of scale experiments are reported in Chapter 3. Throughout the operation period of the demonstration plant no scale adherence in the fluidized bed heat exchanger has been observed. It is, however, reported that scale particles formed in the heat exchanger tubes but removed by the fluidized bed particle were found to grow in the flash stage after flashing-off the brine and release of carbon dioxide from the first stages. Due to specific design of the adjustable slit in the orifice opening in the interstage wall as shown in figure 5-38 some fouling and blockage was observed by calcium carbonate scale formed in the stages.
Design changes made afterwards in the further development of the MSF/FBE are aimed to overcome this disadvantage by allowing eventually formed scale to pass and to be discharged.

Venting of non-condensables at lower temperatures than the design temperatures proved to be insufficient due to only marginally oversized orifices in the lower temperature end of the evaporator condenser section. To avoid increase of energy consumption caused by insufficient removing of non-condensable gases at these low temperatures back-mixing of brine with feedwater was introduced to maintain design temperatures.

5.5.2. 100 m³/day commercial MSF/FBE.

On the basis of the results of the research and development work as reported in Chapter 2 through Chapter 5 and complemented with operational experience gained with the demonstration MSF/FBE described in subchapter 5.5.1, ESMIL further developed the MSF/FBE. Development work was carried out in their research facilities and value engineering was conducted by the engineering staff.

As a result a small scale commercial MSF/FBE was contracted, designed manufactured and put into operation at the Municipal Refuse Incineration station of the city of Amsterdam, The Netherlands, figure 5-39. This special application MSF/FBE with a capacity of 100 m³/day is used to provide boiler feed water from high brackish and polluted surface water. Table 5-6 presents the basic design data of the 100 t/d MSF/FBE.

The first commercial generation MSF/FBE is entirely made out of stainless steel to minimise corrosion and is provided with a fluidized bed heat exchanger made of titanium.

The MSF/FBE is in operation since early 1982. The operation of the unit is according to the specifications without no major problems observed after an initial trial period.

Based on the results obtained with the 100 m³/day unit the owner of the installation has placed an order for another similar unit of 100 m³/day to be in operation by the end of 1984.
FIGURE 5-39.
Overall view of the 100 m$^3$/day plant.
<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>distillate production</td>
<td>4275 kg/hr</td>
</tr>
<tr>
<td>steam consumption</td>
<td>2085 kg/hr</td>
</tr>
<tr>
<td>steam pressure</td>
<td>1.60 bar</td>
</tr>
<tr>
<td>steam temperature</td>
<td>113.0 °C</td>
</tr>
<tr>
<td>gain-ratio</td>
<td>2.0 -</td>
</tr>
<tr>
<td>specific heat consumption</td>
<td>1073 kJ/kg</td>
</tr>
<tr>
<td>maximum brine temperature</td>
<td>100.0 °C</td>
</tr>
<tr>
<td>feed temperature</td>
<td>20.0 °C</td>
</tr>
<tr>
<td>blow-down temperature</td>
<td>44.8 °C</td>
</tr>
<tr>
<td>distillate temperature</td>
<td>43.9 °C</td>
</tr>
<tr>
<td>total number of stages</td>
<td>6 -</td>
</tr>
<tr>
<td>concentration of feed</td>
<td>2480 ppm TDS</td>
</tr>
<tr>
<td>concentration of blow-down feed</td>
<td>2733 ppm TDS</td>
</tr>
<tr>
<td>feed</td>
<td>45575 kg/hr</td>
</tr>
<tr>
<td>blow-down</td>
<td>41300 kg/hr</td>
</tr>
<tr>
<td>total number of condenser tubes</td>
<td>203 -</td>
</tr>
<tr>
<td>condenser tube material</td>
<td>titanium</td>
</tr>
<tr>
<td>condenser tube diameters</td>
<td>I.D. 24.0 mm</td>
</tr>
<tr>
<td></td>
<td>O.D. 25.4 mm</td>
</tr>
<tr>
<td>porosity</td>
<td>0.77 -</td>
</tr>
<tr>
<td>diameter of fluidized particles</td>
<td>2.0 mm</td>
</tr>
<tr>
<td>density of particle material</td>
<td>2700 kg/m³</td>
</tr>
</tbody>
</table>

**TABLE 5-6.**

*Design values of the 100 m³/day plant.*
Figure 5-40 shows the flow scheme of this MSF/FBE designed to operate as an once-through evaporator with a design maximum brine temperature of 100.0 °C.

<table>
<thead>
<tr>
<th>1. BRINE</th>
<th>2. STEAM</th>
<th>3. CONDENSATE</th>
</tr>
</thead>
<tbody>
<tr>
<td>pressure bar</td>
<td>2.0</td>
<td>pressure bar</td>
</tr>
<tr>
<td>temperature °C</td>
<td>100.0</td>
<td>temperature °C</td>
</tr>
<tr>
<td>quantity kg/h</td>
<td>45659</td>
<td>quantity kg/h</td>
</tr>
<tr>
<td>T.D.S. ppm</td>
<td>2480</td>
<td></td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>4. BLOW DOWN</th>
<th>5. FEED WATER</th>
<th>6. DISTILLATE</th>
</tr>
</thead>
<tbody>
<tr>
<td>pressure bar</td>
<td>2.0</td>
<td>pressure bar</td>
</tr>
<tr>
<td>temperature °C</td>
<td>44.8</td>
<td>temperature °C</td>
</tr>
<tr>
<td>quantity kg/h</td>
<td>41384</td>
<td>quantity kg/h</td>
</tr>
<tr>
<td>T.D.S. ppm</td>
<td>2733</td>
<td></td>
</tr>
</tbody>
</table>

**FIGURE 5-40.**

*Flow scheme of the 100 m³/day MSF/FBE.*
Figure 5-41 reveals the basic layout of the evaporator stage and fluidized bed condenser bundle.

**FIGURE 5-41.**
Cross section of the 100 m³/day MSF/FBB.
5.5.3. 4000 m³/day MSF/FBE (design).

As a further development and commercialisation of the Multi Stage Flash/Fluidized Bed Evaporator a design for a 4000 m³/day MSF/FBE has been worked out by ESMIL.

Figure 5-42 presents the flow diagram of this design where it can be seen from table 5-7 that the design philosophy is based on low energy consumption per produced unit of product water and noble material selection for minimal corrosion problems and consequently high availability and long technical lifetime.

Typical cross sections are shown in figure 5-43.

![Flow scheme diagram](image)

<table>
<thead>
<tr>
<th>1. BRINE</th>
<th>2. STEAM</th>
<th>3. CONDENSATE</th>
</tr>
</thead>
<tbody>
<tr>
<td>pressure bar</td>
<td>pressure bar</td>
<td>pressure bar</td>
</tr>
<tr>
<td>2.0</td>
<td>4.0</td>
<td>1.5</td>
</tr>
<tr>
<td>temperature °C</td>
<td>temperature °C</td>
<td>temperature °C</td>
</tr>
<tr>
<td>105.0</td>
<td>144.0</td>
<td>110.0</td>
</tr>
<tr>
<td>quantity kg/h</td>
<td>quantity kg/h</td>
<td>quantity kg/h</td>
</tr>
<tr>
<td>1533754</td>
<td>13400</td>
<td>13107</td>
</tr>
<tr>
<td>T.D.S. ppm</td>
<td></td>
<td></td>
</tr>
<tr>
<td>45000</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>4. BLOW DOWN</th>
<th>5. FEED WATER</th>
<th>6. DISTILLATE</th>
</tr>
</thead>
<tbody>
<tr>
<td>pressure bar</td>
<td>pressure bar</td>
<td>pressure bar</td>
</tr>
<tr>
<td>1.0</td>
<td>2.0</td>
<td>3.0</td>
</tr>
<tr>
<td>temperature °C</td>
<td>temperature °C</td>
<td>temperature °C</td>
</tr>
<tr>
<td>37.4</td>
<td>33.0</td>
<td>36.1</td>
</tr>
<tr>
<td>quantity kg/h</td>
<td>quantity kg/h</td>
<td>quantity kg/h</td>
</tr>
<tr>
<td>1367095</td>
<td>1546500</td>
<td>166659</td>
</tr>
<tr>
<td>T.D.S. ppm</td>
<td></td>
<td></td>
</tr>
<tr>
<td>50475</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

**FIGURE 5-42.**

Flow scheme of the 4000 m³/day MSF/FBE (in design stage).
<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>distillate production</td>
<td>166659 kg/hr</td>
</tr>
<tr>
<td>steam consumption</td>
<td>13107 kg/hr</td>
</tr>
<tr>
<td>steam pressure</td>
<td>4.0 bar</td>
</tr>
<tr>
<td>steam temperature</td>
<td>144.0 °C</td>
</tr>
<tr>
<td>gain-ratio</td>
<td>14.0 -</td>
</tr>
<tr>
<td>specific heat consumption</td>
<td>157.9 kJ/kg</td>
</tr>
<tr>
<td>maximum brine temperature</td>
<td>105.0 °C</td>
</tr>
<tr>
<td>feed temperature</td>
<td>31.0 °C</td>
</tr>
<tr>
<td>blow-down temperature</td>
<td>37.4 °C</td>
</tr>
<tr>
<td>distillate temperature</td>
<td>36.1 °C</td>
</tr>
<tr>
<td>total number of stages</td>
<td>39 -</td>
</tr>
<tr>
<td>concentration of feed</td>
<td>45000 ppm TDS</td>
</tr>
<tr>
<td>concentration of blow-down</td>
<td>50475 ppm TDS</td>
</tr>
<tr>
<td>feed</td>
<td>1533659 kg/hr</td>
</tr>
<tr>
<td>blow-down</td>
<td>1367000 kg/hr</td>
</tr>
<tr>
<td>total number of condenser tubes</td>
<td>9736 -</td>
</tr>
<tr>
<td>condenser tube material</td>
<td>titanium</td>
</tr>
<tr>
<td>condenser tube diameters</td>
<td>I.D. 20.82 mm</td>
</tr>
<tr>
<td></td>
<td>O.D. 22.22 mm</td>
</tr>
<tr>
<td>porosity</td>
<td>0.75 -</td>
</tr>
<tr>
<td>diameter of fluidized particles</td>
<td>2.0 mm</td>
</tr>
<tr>
<td>density of particle material</td>
<td>2700 kg/m³</td>
</tr>
</tbody>
</table>

**TABLE 5-7.**
Design values of the 4000 m³/day plant.

**FIGURE 5-43.**
Cross section of the 4000 m³/day MSF/FBE.
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The research project was initiated by and executed under the scientific responsibility of prof.ir. J.J.C. van Lier. The frequent discussions with prof.ir. J.J.C. van Lier have proven to be of essential value to this thesis.

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The author is thankful to ESMIL B.V. for the offered opportunities for discussions and the cooperation in the research program for the development of the Multi-Stage Flash/Fluidized Bed Evaporator. Particularly the discussions with dr.ir. D.G. Klaren, ir. G.A. Pieper and drs.ir. J.P.P. Tholen are very much appreciated.

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SAMENVATTING

Dit proefschrift behandelt de ontwikkeling van de vertikale meertraps-ontspanverdamper met geïflueïseerd bed warmtewisselaar (de Multi-Stage Flash/Fluidized Bed Evaporator).

Na een algemene uiteenzetting over de meertrapsontspanverdamper, die verder als referentie zal worden gebruikt, wordt in Hoofdstuk 1 de ontwikkeling van de MSF/FBE geplaatst naast andere ontwikkelingen op het vakgebied van de thermische ontziling. Aansluitend wordt in Hoofdstuk 1 de aannemethoden beschreven van het uitgevoerde ontwikkelingswerk.

Dit ontwikkelingswerk omvat de volgende facetten:

- warmteoverdracht van de wand naar de vloeistof in een geïflueïseerd bed
- vervuilingssedimentatie en ketelsteenvorming in een geïflueïseerd bed
- volledige uit-damping en meesleuren van druppeltjes in de verdamperkamer van een MSF/FBE
- ontwerp- en bedrijfsaspecten van een MSF/FBE.

De uitgevoerde metingen voor het bepalen van de warmteoverdracht van de wand naar de vloeistof in een geïflueïseerd bed worden beschreven in Hoofdstuk 2 en zijn gebaseerd op twee meetprincipes:

A. De warmtestroom naar de vloeistof in de warmtewisselaarpijp wordt vastgesteld met behulp van een thermo-element, ingebed in de wand van de warmtewisselaar, onder gebruikmaking van de Nusselt-korrelatie voor kondenserende stoom aan de buitenzijde van een pijp.

B. De warmtestroom naar de vloeistof in de warmtewisselaarpijp wordt vastgesteld met behulp van twee thermo-elementen, die in dezelfde dwarsdoorsnede van de pijp zijn geplaatst, maar zich op verschillende afstand van de oppervlakte van de pijpwand bevinden. Methode B wordt toegepast omdat met methode A geen voldoende betrouwbare meetgegevens konden worden verkregen.

Naast de beschrijving van de proefopstelling wordt in Hoofdstuk 2 tevens ingegaan op de toegepaste methode voor het meten van wandtemperatuur met ingebedde thermo-elementen en op de berekening ter bepaling van de fiktieve afstand van deze thermo-elementen tot het pijppoppervlak.
Vervuiling van en ketelsteenvorming op de warmteoverdrachtsoppervlakken van de warmtewisselaars is een belangrijk aspekt in thermische ontziling. Hierop wordt in Hoofdstuk 3 nader ingegaan.

Na een overzicht van bekende chemische methoden ter voorkoming van ketelsteen bij conventionele thermische ontziltingstechnieken wordt het gedrag van een gefluïdeerd bed met betrekking tot vervuiling en ketelsteenvorming besproken. Ingegaan wordt op het ontstaan en de groei van ketelsteenkristallen en een verklaring wordt gegeven voor het afwezig zijn van vervuiling bij installaties met gefluïdeerd bed.

Voor verontreinigd brak oppervlaktewater, zeewater en kunstmatig zeewater zijn experimenten uitgevoerd om de mate waarin ketelsteenvorming bij toevoeging van een gefluïdeerd bed kan worden voorkomen, vast te stellen. De experimenten met oppervlaktewater en een deel van de experimenten met zeewater hebben betrekking op ketelsteenvorming gebaseerd op calciumcarbonaat en magnesiumhydroxide. De overige experimenten met zeewater en de experimenten met kunstmatig zeewater hebben betrekking op ketelsteenvorming door calciumsulfaat.

Na een kort overzicht van de in de literatuur gepubliceerde resultaten betreffende onderzoek en formulering van modellen ter beschrijving van het uitkookproces bij "flashen" gaat Hoofdstuk 4 nader in op het experimentele werk met betrekking tot het uitkookproces en het meesleuren van druppeltjes in een verdamperkamer van een MSF/FBE.

Uitgegaan is van 3 typen verdamperkamers en wel de kamers A, B en C. Tijdens het experiment wordt bij de betreffende types verdamperkamers een aantal wijzigingen in het interieur aangebracht. Per type verdamperkamer zijn metingen uitgevoerd voor een reeks van zoutconcentraties, temperatuur niveaus en temperatuursvallen. Een aansluitende analyse van het gedrag van druppeltjes in het uittredekanaal van de verdamperkamer heeft uiteindelijk geresulteerd tot voorstellen voor een definitief ontwerp van een MSF/FBE verdamperkamer.

Aan het slot van Hoofdstuk 4 is het toelaatbaar verzadigingstemperatuurverlies, resulterend uit verliezen door wrijving en snelheidsveranderingen in het dampkanaal, geanalyseerd en is een mathematische relatie gepresenteerd voor dit temperatuurverlies.
Hoofdstuk 5 behandelt de bedrijfsaspekten van de eerste generatie MSF/FBE. De konstruktie van de 50 m³/dag eenheid wordt beschreven alsmede het hydraulisch gedrag van de parallelle, gefluidiseerde bedden en de toegepast afsmoorinrichting. Konstruktiewijzigingen als gevolg van de opgedane bedrijfservaring worden besproken. Ook wordt een beschrijving gegeven van een meetmethode waarbij temperaturen in de gefluidiseerde bedden tijdens het in bedrijf zijn van de verdamper kunnen worden gemeten. Daarnaast wordt aandacht besteed aan de invloed van niet-kondenseerbare gassen op het kondensatieproces en wordt vermeld welke maatregelen zijn genomen om de afvoer van niet-kondenseerbare gassen te verbeteren.

Aan het slot van Hoofdstuk 5 wordt een overzicht gegeven van de ontwikkelingen van de MSF/FBE technologie zoals die na de eerste generatie MSF/FBE van 50 m³/dag tot stand zijn gekomen.

Dit overzicht omvat:

- een 500 m³/dag experimentele eenheid (op het eiland Texel)
- een 100 m³/dag commerciële eenheid (stad Amsterdam) en
- een 100 m³/dag ontwerp studie.
Stellingen bij het proefschrift:
"Contributions to the development of a Multi-Stage Flash/Fluidized Evaporator (MSF/FBE)".

door

A.W. Veenman

1. Het vaststellen van warmteoverdrachtskoëfficiënten in een gefluidiseerd bed systeem waarbij aan de buitenzijde van de warmtewisselaarpijp kondensatie plaatsvindt is moeilijker dan veelal wordt aangenomen en gaat gepaard met een aanzienlijke onnauwkeurigheid in de gevonden resultaten.

2. De combinatie van de mogelijkheden om een MSF/FBE te ontwerpen met een laag specifiek warmteverbruik, het onbreken van de noodzaak om chemieën te doseren ter voorkoming van ketelsteen in de warmtewisselaar en het geringe volume van de verdamer maakt de MSF/FBE tot een aantrekkelijk alternatief voor het ontzilten van zeewater of verontreinigd brakwater (Hoofdstukken 2, 3 en 4).

3. Oppervlakteruwheid van warmtewisselaarpijpen bij toepassing van gefluidiseerde bedden heeft slechts een kleine invloed op eventuele permanente ketelsteenvorming (Hoofdstuk 3).

4. Onderzoek naar op chemische wijze voorkomen van ketelsteenvorming in ontziltingsinstallaties kan worden versneld door studie naar de omstandigheden van niersteenvorming (Hoofdstuk 3).

5. Het meesleuren van waterdruppeltjes in de damp van een vertikale meertrapsontspannerdamper (MSF/FBE) is veel minder afhankelijk van de brijnmaastrroom dan van de initiële oververhitting van de vloeistof (Hoofdstuk 4).

6. De combinatie van een hoge belasting van de dampruimte van de verdamperkamer en de geringe ruimte ingenomen door vloeistof resulteert in een zodanig klein volume van de MSF/FBE verdamperkolom dat toepassing van korrosiebestendige materialen niet leidt tot excessief hogere investeringen.

7. De toelaatbare - zeer hoge - belasting van een verdamperkamer van een MSF/FBE wordt zeer sterk beïnvloed door de geometrie van de kamer en de daarin toegepaste schotten: wijziging van één der afmetingen zal dan ook zeer weloverwogen dienen te geschieden en veelal andere afmetingen sterk doen wijzigen (Hoofdstuk 4).
8. Een kleine afstand tussen het vloeistofoppervlak in een verdamperkamer van een MSF/FBE teneinde een zo gering mogelijk meesleuren van druppeltjes te bewerkstelligen is in konflikt met de regelbaarheid van een verdamper en zal leiden tot variabele openingen in de schedingswanden tussen de verdamperkamers (Hoofdstuk 4).

9. Ontziling door middel van thermische processen en door middel van membraanprocessen vertonen meer overeenkomsten dan beoefenaren van de vakgebieden in het algemeen willen aannemen.

10. Subsidiëring van water geproduceerd met ontziltingsinstallaties heeft geleid tot keuzen voor ontwerpen en systemen die energetisch-economisch niet optimaal zijn.

11. Vormen van samenwerking tussen onderzoekinstellingen en het bedrijfsleven zijn aan te bevelen zolang elk van de partijen gericht blijft op zijn basisdoelstelling.

12. De van oorsprong Amerikaanse gewoonte om gekoelde dranken te serveren met een overmaat aan ijsblokjes is energieverspilling.

13. De afwijking tussen 1024 en de betekenis van het voorvoegsel "kilo" in de aanduiding van de geheugencapaciteit "kilobytes" in de computertechniek staat in schril contrast tot de nauwkeurigheid van de werking van de computer.

14. De door luchtvaartmaatschappijen gehanteerde begrippen in relatie tot comfort dienen alleen voor onderling vergelijk van luchtvaartmaatschappijen gebruikt te worden.

15. De arabische mentaliteit in het tot stand komen en afwikkelen van zaken vertoont in vele opzichten overeenkomst met de west-europese mentaliteit.

16. Belangrijke besprekingen direkt aansluitend op interkontinentale reizen per vliegtuig en waarbij grote tijdsverschillen worden overbrugd zouden alleen in uiterste noodzaak mogen worden gehouden.

17. In gezinnen met kleine kinderen leidt de toepassing van eengatsmengkranen met centrale bediening voor zowel de temperatuurs- als de hoeveelheidsregeling tot energieverspilling.
Stellingen bij het proefschrift:
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