Freezing processes based on the refrigerant CO2
performance and modelling of a horizontal plate freezer

Peijnenburg, B.

Award date:
2003
FREEZING PROCESSES BASED ON THE REFRIGERANT \( \text{CO}_2 \)

PERFORMANCE AND MODELLING OF A HORIZONTAL PLATE FREEZER

B. PEIJNENBURG
MARCH, 2003

Report of a graduation project.
Supervisor: Ir. R. J. F. Vermeeren (TNO-MEP, Apeldoorn)
Professor: Prof. dr. A.T.A.M. de Waele (Tu/e).
Abstract

In this thesis, freezing processes based on the refrigerant carbon dioxide (CO$_2$) are studied. This because CO$_2$ has very good physical and thermodynamical properties, and therefore can be a substitute for refrigerants like ammonia (NH$_3$) or freon-22 (R-22), in (new and existing) freezing techniques.

Freezing techniques that are examined: two (new) half-open processes and a horizontal plate freezer, especially designed for CO$_2$. The half-open processes are theoretically studied and compared with a freezing tunnel placed at Northseafood Urk. The horizontal plate freezer, on the other hand, is modeled, and simulations are executed with this model. Finally experiments are performed on a pilot plant, built at TNO-MEP Apeldoorn, to compare the results with the simulations and reference measurement, done at Polskamp Meat Ernelo.

Calculations on the freezing of fish sticks in the half-open processes showed that both processes are viable. Both resulted in a similar (suction process) or a shorter (heat-pipe process) freezing time of the fish sticks compared with the freezing tunnel at Northseafood Urk. Furthermore, both processes can save a significant amount of money on the operational costs, due to the half-open character. Therefore it is recommended to investigate the practicability of these processes.

In the horizontal plate freezer project, simulations and experiments were performed. The simulations showed that the pressure drop caused by the orifices inside the plates is dominant, which implies a uniform distribution of liquid CO$_2$ over the channels. This is nearly similar to what was measured in the experiments. Only one channel was malfunctioning, but this was independent of the conditions, and therefore probably caused by a construction error, or pollution in the channel. In the experiments, at optimal conditions, the freezing time of tylose packages (replacement of meat) was approximately 65 minutes. Compared to the reference at Polskamp (± 60 minutes) this is a satisfying result. Hereby, it can be concluded that the plate freezer, especially designed for CO$_2$, works.

Overall it can be said that CO$_2$ is a good substitute for the examined freezing techniques, but it is necessary to look thoroughly at the bottlenecks, and requirements before it can be implemented in existing or new processes.
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Letters

\( a \) \hspace{1cm} \text{thermal diffusivity} \ [\text{m}^2/\text{s}]
\( A \) \hspace{1cm} \text{area} \ [\text{m}^2]
\( c \) \hspace{1cm} \text{constant} \ [-]
\( c_p \) \hspace{1cm} \text{specific heat} \ [\text{J/kg/K}]
\( D \) \hspace{1cm} \text{diameter} \ [\text{m}]
\( D_h \) \hspace{1cm} \text{hydraulic diameter} \ [\text{m}]
\( f \) \hspace{1cm} \text{friction factor Darcy [-]}
\( f \) \hspace{1cm} \text{design tension [-]}
\( g \) \hspace{1cm} \text{gravitational acceleration} \ [\text{m/s}^2]
\( G \) \hspace{1cm} \text{mass flux} \ [\text{kg/m}^2]
\( h \) \hspace{1cm} \text{specific enthalpy} \ [\text{J/kgK}]
\( H \) \hspace{1cm} \text{enthalpy} \ [\text{J}]
\( K \) \hspace{1cm} \text{tensile strength} \ [\text{N/mm}^2]
\( L \) \hspace{1cm} \text{latent heat} \ [\text{J/kg}]
\( L \) \hspace{1cm} \text{length channel} \ [\text{m}]
\( m \) \hspace{1cm} \text{mass} \ [\text{kg}]
\( \dot{m} \) \hspace{1cm} \text{mass flow} \ [\text{kg/s}]
\( p \) \hspace{1cm} \text{pressure} \ [\text{bar}]
\( P \) \hspace{1cm} \text{refrigeration capacity} \ [\text{kW}]
\( q \) \hspace{1cm} \text{heat flux} \ [\text{W/m}^2]
\( \dot{q} \) \hspace{1cm} \text{heat source} \ [\text{W}]
\( Q \) \hspace{1cm} \text{volume flow} \ [\text{m}^3/\text{s}]
\( r \) \hspace{1cm} \text{radius} \ [\text{m}]
\( S \) \hspace{1cm} \text{slip} [-]
\( S_t \) \hspace{1cm} \text{suppression factor} [-]
\( t \) \hspace{1cm} \text{time} \ [\text{s}]
\( T \) \hspace{1cm} \text{temperature} \ [\text{K}]
\( v \) \hspace{1cm} \text{velocity} \ [\text{m/s}]
\( V \) \hspace{1cm} \text{volume} \ [\text{m}^3]
\( w \) \hspace{1cm} \text{weakness factor} [-]
\( x \) \hspace{1cm} \text{quality} [-]
\( x \) \hspace{1cm} \text{coordinate} \ [\text{m}]
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# Nomenclature

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INTRODUCTION

1.1 CO₂ in general
Carbon dioxide (CO₂) is not a newcomer in the refrigeration technology. Actually, it was already used in the 19th century, but due to the production of 'safe' synthetic refrigerants it drifted into oblivion. However, in view of the knowledge we now have of the negative environmental effects of synthetic refrigerants, the Ozone Depletion Potential (ODP) and the Global Warming Potential (GWP), CO₂ and other natural refrigerants are attracting a great deal of attention. These refrigerants do not harm the ozone layer and have negligible contribution to the greenhouse effect (GWP (CO₂)=1 compared to GWP(R-507)=3300).

Carbon dioxide is a relatively safe, natural refrigerant. Ammonia (NH₃), still used in a lot of cooling/freezing techniques nowadays, is toxic, even in low concentrations, and therefore less desirable as a refrigerant, in direct contact with the product. CO₂, on the other hand, is not toxic, flammable, or explosive. Furthermore, CO₂ has some interesting physical properties: volumetric its refrigeration capacity is high and its heat-transfer properties are quite good. With these characteristics it is possible to engineer more compact and possibly cheaper systems.

Despite the advantages referred to above, CO₂ differs from current refrigerants. The operating pressure is a factor 4 to 8 higher, which demands more of the components in the system. In a stationary situation, the pressure can run up to about 70 bar (approximately 30°C). As a result the application of CO₂ as a refrigerant requires special provisions for systems.

1.2 CO₂ in freezing applications
Freezing is important in the food industry. Mostly the product (meat, vegetables, ice-cream, etc.) is frozen in tunnels or plate freezers. When a product is frozen faster (decrease in freezing time often due to a lower temperature of the refrigerant) the quality of the product stays higher. A smaller formation of crystals occurs. This implies no cell rupture during the freezing process. With CO₂ as refrigerant it is possible to freeze at a lower temperature than with NH₃. The reason is that the pressure for CO₂ is always above 1 bar even close
1. INTRODUCTION

to the triple point. In contrast, liquid NH₃ at a temperature of -35°C has a pressure lower than 1 bar. In other words, systems using liquid NH₃ below -35°C, are sensitive for air leakage.

Due to the high volumetric refrigeration capacity and good heat-transfer, CO₂ is desirable as a refrigerant. Nowadays CO₂ is mainly assessed in air-cooling systems, but can it also be used in freezing? If so, what are the new possibilities and which existing systems can still be used in combination with CO₂? In this report both elements will be examined and discussed.

In the first place two new, alternative freezing processes are studied. Both processes have half-open properties. This means that CO₂ is in direct contact with the product to be frozen, and will be recycled. Both processes are studied on freezing time and energy consumption. These will be compared with a freezing tunnel located at Northseafood Holland in Urk. According to these results a few discussion points emerge and finally some recommendations for improvement of both processes are given.

1.3 Testing performance horizontal plate freezer

The Horizontal Plate Freezer (from now on called HPF) is known worldwide. Most of the time these plate freezers are used with Freon-22 (R-22), or NH₃ as refrigerant. To examine the possibility of using a HPF with CO₂ as refrigerant, TNO-MEP, Polskamp Meat Industry B.V., GTI, and Samifi built a pilot plant especially designed for CO₂ as refrigerant, together with an NH₃-CO₂ installation.

In order to investigate the properties of the HPF in combination with CO₂ as (indirect) refrigerant, a reference measurement was performed in the first place at Polskamp Meat Industry B.V. In this factory a HPF is present which uses R-22. During the cooling of meat several thermocouples were placed to set a reference concerning the freezing time of meat.

Secondly simulations were executed in MATLAB™ to get a better understanding on what is happening inside the plates in the pilot plant, and to compare with the experimental results. In these simulations the pressure drop inside the plate, distribution of CO₂ mass flow inside the plates, and heat-transfer coefficient were calculated.

Finally a series of experiments with tylose test packages and meat were performed. The evaporating temperature of CO₂ was varied as well as the circulation rate. This was done to investigate the influence on the pressure drop over the header and plate, temperature distribution on the plate, and temperature of the product. These parameters were monitored to determine the energy consumption of the system, cooling capacity of the HPF, freezing time of the product and the heat-transfer coefficient. As a result it is known under which conditions (evaporating temperature CO₂ and circulation rate) the HPF performs best.

In the end the reference, simulations and experiments are compared which results in some conclusions and recommendations to let the HPF perform in better.
2

THERMODYNAMICS OF $\text{CO}_2$

2.1 Introduction

Pressure drops in single and two-phase flows, heat-transfer in an evaporating liquid, thermodynamic transitions are all phenomena that occur in a cooling/freezing process. In this chapter the thermodynamical and physical properties of $\text{CO}_2$ will be described and compared with other refrigerants that are used nowadays.

2.2 Thermodynamical and physical properties of refrigerants

The physical and chemical properties of a refrigerant are of great influence on the design and the performance of cooling systems. The most important thermodynamical and physical properties of $\text{CO}_2$ will be described and also which consequences this has for the cooling.

In Table 2.1 the most relevant thermodynamical properties of $\text{CO}_2$, NH$_3$ and R-22 (industrial refrigerants) are shown. The table shows that $\text{CO}_2$ distinguishes itself on a number of points, which will be described in more detail.

**Triple point**

In the triple point there is equilibrium between gas, liquid, and solid. Below the triple point temperature $\text{CO}_2$ may become solid. The value of this temperature determines the lower limit for the application of $\text{CO}_2$ as a refrigerant. This temperature is $-56.6^\circ\text{C}$.

**Critical point**

Above the critical pressure and temperature no liquid-gas transition takes place. $\text{CO}_2$ has a relatively low critical point at $31.1^\circ\text{C}$ and 73.8 bar, which has consequences for the heat emission in a vapour compression cycle. Above the critical temperature heat emission cannot occur through condensation and a gas cooler must be used.

For the application of $\text{CO}_2$ in a cooling system a distinction can be made between a subcritical and a transcritical process. When the temperature at heat emission stays below the critical temperature, it is a subcritical process, and for the heat emission a normal condenser can be used. In the case the heat emission temperature increases above $31.1^\circ\text{C}$, for example air cooling above $40^\circ\text{C}$, the medium becomes supercritical. This results in a degradation of
### 2. THERMODYNAMICS OF CO₂

<table>
<thead>
<tr>
<th>Properties</th>
<th>CO₂</th>
<th>NH₃</th>
<th>R-22</th>
</tr>
</thead>
<tbody>
<tr>
<td>Triple point [°C]</td>
<td>-56.6</td>
<td>-77.7</td>
<td>-157.4</td>
</tr>
<tr>
<td>Triple point [bar]</td>
<td>5.2</td>
<td>0.061</td>
<td>??</td>
</tr>
<tr>
<td>Critical temperature [°C]</td>
<td>31.1</td>
<td>132.3</td>
<td>96.2</td>
</tr>
<tr>
<td>Critical pressure [bar]</td>
<td>73.8</td>
<td>113.3</td>
<td>49.9</td>
</tr>
<tr>
<td>Vapour pressure at -35°C [bar]</td>
<td>12.0</td>
<td>0.93</td>
<td>1.32</td>
</tr>
<tr>
<td>Evaporation heat at -35°C [kJ/kg]</td>
<td>312.2</td>
<td>1374</td>
<td>230</td>
</tr>
<tr>
<td>Volumetric refrigeration capacity at -35°C [kJ/m³]</td>
<td>9750</td>
<td>1130</td>
<td>1386</td>
</tr>
<tr>
<td>Liquid pressure ratio at -35°C/0°C [-]</td>
<td>2.90</td>
<td>4.61</td>
<td>3.77</td>
</tr>
<tr>
<td>Density liquid at -35°C [kg/m³]</td>
<td>1096</td>
<td>684</td>
<td>1436</td>
</tr>
<tr>
<td>Density vapour at -35°C [kg/m³]</td>
<td>31.23</td>
<td>0.82</td>
<td>3.09</td>
</tr>
<tr>
<td>Liquid/vapour density ratio at -35°C [-]</td>
<td>35.1</td>
<td>832.3</td>
<td>465.0</td>
</tr>
<tr>
<td>Specific heat liquid at -35°C [kJ/kg]</td>
<td>2.005</td>
<td>4.439</td>
<td>1.097</td>
</tr>
<tr>
<td>Specific heat vapour at -35°C [kJ/kg]</td>
<td>1.101</td>
<td>2.283</td>
<td>0.621</td>
</tr>
<tr>
<td>Thermal conductivity liquid at -35°C [W/Km]</td>
<td>0.153</td>
<td>0.671</td>
<td>0.095</td>
</tr>
<tr>
<td>Thermal conductivity vapour at -35°C [W/Km]</td>
<td>0.013</td>
<td>0.021</td>
<td>0.007</td>
</tr>
<tr>
<td>Kinematic viscosity liquid at -35°C [mm²/s]</td>
<td>0.17</td>
<td>0.38</td>
<td>0.22</td>
</tr>
<tr>
<td>Kinematic viscosity vapour at -35°C [mm²/s]</td>
<td>0.38</td>
<td>9.74</td>
<td>1.66</td>
</tr>
<tr>
<td>Surface tension at 0°C [N/m]</td>
<td>0.0046</td>
<td>0.0329</td>
<td>0.0117</td>
</tr>
</tbody>
</table>

**Table 2.1: Thermodynamical and physical properties of primary refrigerants [20].**

the COP (Coefficient Of Performance), due to the decrease in latent heat. When the heat emission is in the supercritical area, the process is called transcritical. Because CO₂ is in the supercritical state there is no transition in phase. This means that the condenser has to be replaced by a gas cooler [20].

**Vapour pressure**
The vapour pressure of CO₂ is, at the usual cryogenically conditions, a factor 8 to 10 higher than that of refrigerants which are used in current vapour compression systems. Conventional systems and system components, with a design pressure of maximum 25 bar, are therefore not directly usable with CO₂.

**Volumetric refrigeration capacity**
The volumetric refrigeration capacity of CO₂ is a factor 4 to 8 higher than of NH₃ and some other synthetic refrigerants. This results, at steady demand of cold, in smaller compression volumes and pipe diameters in the evaporation area. This offers the possibility for smaller system components and therefore a smaller and lighter system.

**Pressure ratio**
The pressure ratio, defined as the ratio of the pressures at heat emission and evaporation, is for CO₂ 1.5 till 2 times smaller than for other refrigerants. This low pressure ratio has a positive influence on the isentropic as well as on the volumetric efficiency of the compressor.

**Density**
The vapour density of CO₂ is relatively high. This results in a liquid/vapour density ratio

*Freezing processes based on the refrigerant CO₂*
which is much lower than the conventional refrigerants. A smaller density ratio can contribute to a better distribution of the two-phase medium in the evaporator because it will behave more homogeneously.

**Specific heat**
With exception of NH$_3$ the specific heat of CO$_2$ is twice higher than that of conventional refrigerants. In the neighbourhood of the critical point the heat capacity of CO$_2$ rises extremely, which has a strong positive influence on the heat transfer.

**Thermal conductivity**
The thermal conductivity of CO$_2$ is about 1.5 times higher than most of the synthetic refrigerants. The high thermal conductivity has a positive influence on the heat transfer in the evaporator. However, NH$_3$ has a thermal conductivity which is 5 times higher than CO$_2$.

**Kinematic viscosity**
The kinematic viscosity of liquid CO$_2$ at -35$^\circ$C is 30-40% lower than synthetic refrigerants and twice as low as NH$_3$. At 30$^\circ$C it is even 3 till 5 times lower. This results in a smaller pressure drop over the system components and pipes (as far this is determined by the viscosity).

**Surface tension**
The surface tension of CO$_2$ is much lower than of other refrigerants. A lower surface tension can contribute to a higher heat-transfer coefficient during evaporation due to a more homogeneously two-phase mixture, which implies a smaller evaporator.

### 2.3 Chemical interaction of CO$_2$

CO$_2$ can react violently with NH$_3$. A white powder, (NH$_4$)$_2$CO$_3$, can be formed during the contact of CO$_2$ with NH$_3$, which can lead to pollution of the heat exchangers and damage of the compressor. Also NH$_4$(NH$_2$-CO$_2$), which reacts corrosively with steel, can be formed. In combination with water, CO$_2$ also reacts corrosively with steel. Therefore leakage of water into the system must be prevented with driers. Finally the usage of aluminum, in the HPF project, prevents corrosion of the plates.

Freezing processes based on the refrigerant CO$_2$
3

THEORY OF SINGLE PHASE FLOW

3.1 Introduction

In the HPF project, liquid CO2 is injected into the plates, and generates a pressure drop. The pressure drop, created by the friction of single phase flow, drives the distribution of the liquid CO2 in the channels in the plates. In this chapter the theory of single phase flow will be described.

3.2 Pressure drop single phase channel flow

Pressure drops in systems of channels usually depends on a couple of different factors:

- The velocity
- The roughness of the channel
- The fluid properties like the density and the viscosity.

Usually the pressure drop is related to the average speed \( v \) of the flow with the Darcy friction factor \( f \),

\[
\Delta p = -f \frac{L}{D_h} \frac{1}{2} \rho v^2. \quad (3.1)
\]

In this equation \( \Delta p \) is the pressure drop over the channel, \( L \) the length of the channel, \( \rho \) the density of the liquid, and \( D_h \) the hydraulic diameter which is defined as

\[
D_h = 4 \frac{A}{P}, \quad (3.2)
\]

where \( A \) is the cross sectional area, and \( P \) the wetted surrounding as can be seen in Figure 3.1.
3. THEORY OF SINGLE PHASE FLOW

3.2.1 LAMINAR NEWTONIAN FLOW

In the first place look at round channels with laminar Newtonian flow. The pressure is uniform over the whole area and the forces of compression form an equilibrium with the viscous forces. In this figure, \( \tau_s \) is the surface shear tension, which is defined as

\[
\tau_s = - \left( \eta \frac{dv(r)}{dr} \right)_{r=r_s},
\]

where \( \eta \) is the viscosity, and \( r_s \) is the radius till the surface. The mass flow through a channel can be related to the pressure drop in that channel

\[
\dot{m} = \rho \int_0^{r_s} \upsilon(r) 2\pi r dr = \frac{\Delta p \pi \rho}{2L\eta} \int_0^{r_s} (r^2 - r_s^2) r dr = -\frac{\Delta p \pi r_s^4 \rho}{8L\eta}.
\]

The average velocity, independent of the position, \( \langle \upsilon \rangle_p \) is defined as

\[
\langle \upsilon \rangle_p = \frac{\dot{m}}{\rho \pi r_s^2} = -\frac{\Delta p r_s^2}{8L\eta} = -\frac{\Delta p D^2}{32L\eta},
\]

so Eq. (3.5) can be written in the form of Eq. (3.1) as

\[
\Delta p = \frac{64L\eta}{\rho \langle \upsilon \rangle_p D^2 \rho} \frac{\langle \upsilon \rangle_p^2}{2} = \frac{64}{Re} \frac{L}{D^2 \rho \langle \upsilon \rangle_p^2}, \quad Re = \frac{\rho \langle \upsilon \rangle_p D}{\eta}.
\]

For a laminar flow the friction factor \( f \) is a simple function of the Reynolds number, \( f = -64/Re \). The pressure drop is linearly dependent on the velocity.

3.2.2 TURBULENT NEWTONIAN FLOW

The analysis in section 3.2.1 is limited for a laminar flow. The way the result is generated, can also be used in a general case. When the shear tension is known as a function of the velocity and its gradient, the friction factor can also be derived for turbulent flows \( (Re > 2300) \). The relation between the friction and the average velocity in a turbulent flow is not unambiguous and therefore the friction coefficient can only be determined empirically or numerically.

The friction depends on the Reynolds number and the relative wall roughness defined as

Freezing processes based on the refrigerant \( \text{CO}_2 \)
3. THEORY OF SINGLE PHASE FLOW

the ratio of the grain size $\varepsilon$ and the channel diameter $D$. This is an obvious difference compared with the laminar flow, where the wall roughness is nearly an important factor. In turbulent and in laminar flow, the fluid velocity at the wall has to be zero, the so called no-slip condition. The Reynolds tensions, which will emerge by turbulent movement, are larger than the viscous forces, while the flow at the wall is viscous, because the velocity has to go to zero.

The grain size is a characteristic linear unit for the roughness of a channel. The definition is very archaic. Grains of sand are filtered in different classes and glued to the inner side of the channel. Then the pressure drop is measured for different values of the volume flow. These tests are done for lots of combinations of channel diameters and grains. If the pressure drop in a commercial channel is equal to that of a channel with grain size $\varepsilon$, then the channel is characterized by a value of $\varepsilon$. In Table 3.1 some values of the equivalent roughness are shown.

<table>
<thead>
<tr>
<th>Material</th>
<th>$\varepsilon$ [mm]</th>
</tr>
</thead>
<tbody>
<tr>
<td>steel</td>
<td>0.04 - 0.06</td>
</tr>
<tr>
<td>- waved</td>
<td>0.9</td>
</tr>
<tr>
<td>- riveted</td>
<td>0.9</td>
</tr>
<tr>
<td>extruded metals</td>
<td>0.00015</td>
</tr>
<tr>
<td>extruded plastics</td>
<td>0</td>
</tr>
<tr>
<td>concrete/clay</td>
<td>0.3</td>
</tr>
<tr>
<td>foundry iron</td>
<td>0.25</td>
</tr>
</tbody>
</table>

Table 3.1: Equivalent roughness for some materials [13].

Chen derived an expression from which $f$ can be determined directly as a function of $Re$ and $\varepsilon/D$. This expression is only valid if $Re > 2300$. The relation is

$$
\frac{1}{\sqrt{f}} = -2.0 \log_{10} \left\{ \frac{\varepsilon}{3.71D} - \frac{5.05}{Re} \log_{10} \left( \frac{1}{2.85} \left[ \frac{\varepsilon}{D} \right]^{1.11} + \frac{5.85}{Re^{0.90}} \right) \right\} .
$$

(3.7)

The experimental results are presented in a Moody Diagram, see Figure 3.2. This diagram shows the friction factor as a function of the relative roughness, the diameter of the channel and the Reynolds number.
3.3 Pressure drop orifice

The plates used in the HPF project contain several orifices. These orifices (see Figure 3.3) generate a pressure drop. This pressure drop can be calculated with the Bernoulli equation. The Bernoulli equation, as shown in Eq. (3.8), has a restriction. It can only be used on a streamline.

\[ \Delta p = \frac{1}{2} \rho v_1^2 \left( \frac{A_1^2}{A_2^2} - 1 \right), \]

with \( v_1 = \frac{Q_{or}}{A_1} \). Where \( Q_{or} \) is the volume flow through each orifice. The way the liquid \( \text{CO}_2 \) is distributed over the channels will be described in chapter 8.
4

THEORY OF TWO-PHASE FLOW

4.1 Introduction

The plates in the HPF project will be cooled with liquid CO₂. These plates contain a defined amount of channels. Inside these channels liquid CO₂ will evaporate, due to heat-transfer, and the flow inside the channels will be of the two-phase flow type. The theory behind these flow patterns and the exact way to determine the pressure drop, inside these channels, will be described in this chapter.

4.2 Flow patterns in horizontal channels

In this report only mixtures of liquids with gases or vapours in horizontal channels are used. The flow patterns in horizontal two-phase flow can be defined as:

1. Dispersed bubble flow: The bubbles are dispersed in a liquid continuum. In horizontal flow, the bubbles tend to congregate near the top of the channel as illustrated. At high velocities, the bubble may be more uniformly distributed and appear as a froth.
2. **Stratified flow**: The gravitational separation is complete: liquid flows along the bottom of the channel and the gas along the top part of the channel with little phase interaction.

3. **Stratified wavy flow**: As the gas velocity is increased in stratified flows, waves are formed on the gas-liquid interface giving the stratified wavy or wavy flow regime.

A variety of intermittent flows can exist in horizontal channels and it is sometimes appropriate to treat all such flows as being of a single generic type of flow regime. However, this class of intermittent flows can be conveniently divided into a further three subdivisions:

4a. **Plug flow**: The bullet-shaped bubbles are observed, but they tend to move along in a position closer to the top of the channel as illustrated in Figure 4.1.

4b. **Semi-slug flow**: The frothy slug takes the form of a wave on the surface of the stratified layer at the bottom of the channel: this wave does not actually touch the top of the channel as it does in slug flow (see 4c).

4c. **Slug flow**: This is characterized by the passage along the channel of frothy "slugs", in which the liquid phase is continuous but a large amount of gas bubbles are entrained.

5. **Annular-dispersed flow**: The film thickness, the "thickness" of the liquid layer, is nonuniform. The film is much thicker at the bottom of the channel. Partial liquid entrainment in the gas core is the rule rather than the exception.

One of the most used flow-pattern map for horizontal two-phase flow is that of Baker. Baker took the fluid properties into account by introducing the following parameters

\[
\lambda_B = \frac{\rho_G \rho_L}{\sqrt{\rho_A \rho_W}}, \quad (4.1)
\]

\[
\psi_B = \frac{\sigma}{\eta} \left[ \frac{\eta_L}{\eta_W} \left( \frac{\rho_W}{\rho_L} \right) \right]^{1/3}, \quad (4.2)
\]

where \(\sigma\) represents the surface tension. The subscripts \(G\) and \(L\) represent the gas and the liquid phases, and the subscript \(A\) and \(W\) represent the values for air and water at atmospheric conditions (typically 20°C and atmospheric pressure). In SI units, appropriate values for the air-water standard conditions are

\[
\rho_A = 1.205 \text{ kg/m}^3, \quad \eta_W = 0.001002 \text{ kg/ms},
\]

\[
\rho_W = 1000 \text{ kg/m}^3, \quad \sigma_W = 0.0728 \text{ kg/s}^2.
\]

It is convenient to plot the Baker map in terms of the coordinates \(\dot{m}_G/\lambda_B\) and \(\dot{m}_L/\psi_B\). A plot in this form is shown in Figure 4.2, where \(\dot{m}_G\) and \(\dot{m}_L\) are the mass flows for, respectively, the gas and the liquid phase.
4.3 Standard definitions

4.3.1 INTRODUCTION

The amount of gas or vapour in relation with the total volume, is called void fraction, indicated with $\varepsilon$. So the void fraction is defined as the gas volume divided by the total volume. Now take a small volume element, $\Delta V$. The local void fraction can be defined as

$$\varepsilon = \varepsilon(\bar{x}, t) = \lim_{\Delta V \to 0} \frac{\text{volume of gas in } \Delta V}{\Delta V}.$$  \hspace{1cm} (4.3)

The volume and time averaged global void fraction, $\langle \varepsilon \rangle_{p,t}$, is defined as the local void fraction, averaged over place and time

$$\langle \varepsilon \rangle_{p,t} = \langle \varepsilon \rangle = \frac{\int \int \varepsilon(\bar{x}, t)dVdt}{\int dV \int dt}.$$  \hspace{1cm} (4.4)

Another important quantity is the quality defined as

$$x = \frac{\int \int G_G \, dA \, dt}{\int \int G \, dA \, dt},$$  \hspace{1cm} (4.5)

where $G_G$ is the local mass flux in the gas phase, and $G$ the total local mass flux. The local mass flux is the local mass flow per unit area and per unit time. The total mass flow, $\dot{m}$, of a two-phase mixture through an area $A$ is defined as

$$\dot{m} = \dot{m}_L + \dot{m}_G,$$  \hspace{1cm} (4.6)

where

$$\dot{m}_i = \int G_i \, dA \quad (i=L,G).$$  \hspace{1cm} (4.7)
4. THEORY OF TWO-PHASE FLOW

Here $\dot{m}_L$ is the mass flow in the liquid phase, and $\dot{m}_G$ the mass flow in the gas phase. The volume flow, $Q$, of a two-phase mixture is defined in an analogous way

$$Q = Q_L + Q_G,$$  \hfill (4.8)

with $Q_L$ the volume flow of the liquid phase and $Q_G$ the volume flow of the gas phase. The relation between mass flow and volume flow is:

$$Q_i = \int \frac{G_i}{\rho_i} \, dA \quad (i=L,G),$$  \hfill (4.9)

where $\rho_i$ is the density of phase $i$.

4.3.2 TRIANGULAR RELATIONSHIP

According to [4] the average void fraction can be written as

$$\langle \varepsilon \rangle = \frac{\rho_L x}{S \rho_G (1-x) + \rho_L x}. \hfill (4.10)$$

In this equation $S$ is the slip defined as

$$S = \frac{\bar{v}_G}{\bar{v}_L}. \hfill (4.11)$$

In this equation $\bar{v}_G$ and $\bar{v}_L$ are respectively the average gas and liquid velocities. These can be defined as

$$\bar{v}_i = \frac{\dot{m}_x}{\rho_i A \langle \varepsilon \rangle} \quad (i=L,G). \hfill (4.12)$$

Looking at the definitions of $x$, $S$, and $\langle \varepsilon \rangle$, we see that they form a triangular relationship. Now let's say $S=1$, which means $\bar{v}_G=\bar{v}_L$. This gives the following relationship:

$$\langle \varepsilon \rangle = \frac{1}{1 + \frac{(1-x) \rho_G}{x \rho_L}}. \hfill (4.13)$$

This is a well known relationship (homogeneous model), and is often used to estimate the void fraction in a two-phase flow. With this information it is possible to give a relationship for the density of the two-phase flow ($\rho_{tp}$). The density of a two-phase flow is defined as

$$\rho_{tp} = \frac{1}{A} \int \left[ (1-\varepsilon) \rho_L + \varepsilon \rho_G \right] dA. \hfill (4.14)$$

In the isothermal case the two-phase density equals

$$\rho_{tp, isotherm} = (1-\langle \varepsilon \rangle) \rho_L + \langle \varepsilon \rangle \rho_G. \hfill (4.15)$$

Freezing processes based on the refrigerant CO$_2$
4. THEORY OF TWO-PHASE FLOW

4.4 Pressure drop

4.4.1 Introduction

In practice it is most common to define the total pressure drop ($\Delta p$) as the sum of the frictional ($\Delta p_{\text{fric}}$), the gravitational ($\rho_{\text{tp}} g \Delta z$), and accelerational ($\Delta p_{\text{acc}}$) pressure drop.

$$\Delta p = \Delta p_{\text{fric}} + \rho_{\text{tp}} g \Delta z + \Delta p_{\text{acc}} .$$  (4.16)

4.4.2 Frictional pressure drop

The frictional pressure drop in circular channels can be written as

$$\Delta p_{\text{fric}} = -f \frac{L}{D} \frac{1}{2} \rho_{\text{tp}} \left( \frac{Q}{A} \right)^2 .$$  (4.17)

To determine the frictional pressure drop, several methods can be used. Methods to determine the pressure drop are the Lockhart & Martinelli and Friedel methods.

4.4.2.1 Lockhart & Martinelli

To take advantage of, already existing, single phase correlations, frictional multipliers have been introduced. This implies:

$$\phi_{G}^2 \frac{dp}{dz} \bigg|_{G} = \frac{dp}{dz} ,$$  (4.18)

where $\frac{dp}{dz} \bigg|_{G}$ is the pressure gradient in the special case there is only flowing gas through the channel, with the same mass flow, $\dot{m}_{G}$, as in the two-phase mixture:

$$-\frac{dp}{dz} \bigg|_{G} = 2 f \frac{(Gx)^2}{\rho_{G}} ,$$  (4.19)

$$Re_{G} = \frac{GxD}{\mu_{G}} \text{ and } Re_{L} = \frac{G(1-x)D}{\mu_{L}} .$$  (4.20)

The factor $\phi_{G}^2$ is defined in a similar way, in the case there is only liquid flowing through the channel. According to Lockhart & Martinelli [4] these multipliers can be defined as

$$\phi_{G}^2 = 1 + cX + X^2 ,$$  (4.21a)

$$\phi_{L}^2 = 1 + \frac{c}{X} + \frac{1}{X^2} .$$  (4.21b)

In these equations $X$ is defined as:

$$X^2 = \frac{dp}{dz} \bigg|_{L} / \frac{dp}{dz} \bigg|_{G} .$$  (4.22)

The value of $c$ is dependent on the flow profile of both phases. See Table 4.1 for the determination of $c$. 

Freezing processes based on the refrigerant CO₂
4. THEORY OF TWO-PHASE FLOW

<table>
<thead>
<tr>
<th>Possible Combinations</th>
<th>Case tt</th>
<th>Case lt</th>
<th>Case tl</th>
<th>Case ll</th>
</tr>
</thead>
<tbody>
<tr>
<td>Liquid Phase</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>( Re_L )</td>
<td>turbulent</td>
<td>laminar</td>
<td>turbulent</td>
<td>laminar</td>
</tr>
<tr>
<td>( \geq 2300 )</td>
<td></td>
<td>(&lt; 2300)</td>
<td>(\geq 2300)</td>
<td>(&lt; 2300)</td>
</tr>
<tr>
<td>Gas Phase</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>( Re_G )</td>
<td>turbulent</td>
<td>turbulent</td>
<td>laminar</td>
<td>laminar</td>
</tr>
<tr>
<td>( \geq 2300 )</td>
<td></td>
<td>(\geq 2300)</td>
<td>(&lt; 2300)</td>
<td>(&lt; 2300)</td>
</tr>
<tr>
<td>Factor ( c )</td>
<td>(c_{tt} \approx 20)</td>
<td>(c_{lt} \approx 12)</td>
<td>(c_{tl} \approx 10)</td>
<td>(c_{ll} \approx 5)</td>
</tr>
</tbody>
</table>

Table 4.1: Possible combinations for \( c \), at different Reynolds numbers \( (Re_L \) or \( Re_G \), where \( t \) = turbulent, and \( l \) = laminar.

4.4.2.2 Friedel

Another correlation to determine the frictional pressure drop, is purposed by Friedel. Friedel used a different definition for the multiplier

\[
\frac{dp}{dz} \bigg|_{\text{Liquid only, with total G-A}} = \frac{dp}{dz}. \quad (4.23)
\]

The correlation itself is an empirical formula

\[
\phi_{L,o}^2 = c_{F_1} + 3.24 c_{F_2} F_{r}^{-0.045} W_\epsilon^{0.035}, \quad (4.24)
\]

where

\[
c_{F_1} = (1 - x)^2 + X^2 \left( \frac{\rho_L}{\rho_G} \right) f_{Go} f_{Lo} \quad (4.25a)
\]

\[
c_{F_2} = x^{0.78} (1 - x)^{0.24} \left( \frac{\rho_L}{\rho_G} \right)^{0.91} \left( \frac{\eta_G}{\eta_L} \right)^{0.19} \left( 1 - \frac{\eta_G}{\eta_L} \right)^{0.7} \quad (4.25b)
\]

Here \( X \) is the Lockhart & Martinelli parameter defined in section 4.4.2.1, and \( f_i \) the friction factor for only gas \((i=Go)\) and liquid \((i=Lo)\) flowing through the channel. In Eq. (4.24) the Froude \((Fr)\) and Weber \((We)\) numbers are used. These are defined as

\[
Fr = \frac{G \eta^2}{g D \rho_{tp}^2}, \quad (4.25c)
\]

\[
We = \frac{G \eta^2 D}{\rho_{tp} \sigma}, \quad (4.25d)
\]

where \( g \) is the gravitational acceleration.

4.4.3 GRAVITATIONAL PRESSURE DROP

The gravitational pressure drop is defined as \( \rho_{tp} g \Delta z \). In Eq. (4.15) the two-phase density \((\rho_{tp})\) is only dependent of \( \langle \epsilon \rangle \). There are several models to determine the average void fraction. This is done by writing the averaged void fraction in the following way:

\[
\frac{1}{\langle \epsilon \rangle} = 1 + n_1 \left( \frac{1 - x}{x} \right)^{n_3} \left( \frac{\rho_G}{\rho_L} \right)^{n_2} \left( \frac{\eta_L}{\eta_G} \right)^{n_1}. \quad (4.26)
\]

Freezing processes based on the refrigerant CO₂
4. THEORY OF TWO-PHASE FLOW

In Table 4.2 the correlation factors for a couple of different models are shown. These models are all determined empirically. None of them is valid in general. Each model has its own boundary conditions. The conditions for the homogeneous model are already described in section 4.3.2 which results in Eq. (4.13). To describe all models would reach too far for this report. The model that will be used in the calculations is the Lockhart & Martinelli correlation.

<table>
<thead>
<tr>
<th>Model</th>
<th>$n_1$</th>
<th>$n_2$</th>
<th>$n_3$</th>
<th>$n_4$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Homeogeneous model</td>
<td>0</td>
<td>1</td>
<td>1</td>
<td>1</td>
</tr>
<tr>
<td>Zivi correlation</td>
<td>0</td>
<td>0.67</td>
<td>1</td>
<td>1</td>
</tr>
<tr>
<td>Thom correlation</td>
<td>0.18</td>
<td>0.89</td>
<td>1</td>
<td>1</td>
</tr>
<tr>
<td>Lockhart &amp; Martinelli correlation</td>
<td>0.07</td>
<td>0.36</td>
<td>0.64</td>
<td>0.28</td>
</tr>
</tbody>
</table>

**Table 4.2:** Correlation factors for determining $(\varepsilon)$ with different models [4].

4.4.4 ACCELERATIONAL PRESSURE DROP

The accelerational pressure drop in Eq. (4.16) is often [4] put in the following form

$$- \frac{dp}{dz}_{acc} = G^2 \frac{d}{dz} \left[ \frac{x^2}{\rho_G(\varepsilon)} + \frac{(1-x)^2}{\rho_L(1-(\varepsilon))} \right].$$

(4.27)

This equation shows that there will be a pressure drop due to changes in the phase of the mixture. For example the evaporation of liquid into vapour. In case of evaporation/boiling there is a strong link between pressure, flow velocity and the creation of bubbles. This will be described in chapter 5.
5

THEORY OF HEAT-TRANSFER

5.1 Introduction

Whenever there exists a temperature difference in a medium or between media, heat-transfer occurs. Heat-transfer can be characterized by three different modes (conduction, convection and radiation, see Figure 5.1).

When a temperature gradient exists in a stationary medium, which may be a solid or a fluid, we use the term conduction to refer to the heat-transfer that will occur across the medium. In contrast, the term convection refers to heat-transfer that will occur between a surface and a moving fluid when they are at different temperatures. The third mode of heat-transfer is termed thermal radiation.

From the three modes mentioned before, only conduction and convection are of interest for this report. So these two will be described in this chapter.
5. THEORY OF HEAT-TRANSFER

5.2 Conduction

The first mode of heat-transfer is conduction. The equation, used to compute the amount of energy being transferred per unit time, for the one-dimensional case is the Fourier equation

\[ q_x = -\lambda \frac{dT}{dx}. \]  

(5.1)

The heat flux, \( q_x \), is the heat flow in the x-direction per unit area, \( \lambda \) is the thermal conductivity.

A major objective in a conduction analysis is to determine the temperature field in a medium resulting from the boundary condition. Once this distribution is known, the conduction heat flux at any point in the medium or on its surface may be computed from Fourier's equation (Eq. 5.1).

In three dimensions the heat diffusion equation, for a homogeneous medium at rest, is expressed as

\[ \frac{\partial}{\partial x} \left( \lambda \frac{\partial T}{\partial x} \right) + \frac{\partial}{\partial y} \left( \lambda \frac{\partial T}{\partial y} \right) + \frac{\partial}{\partial z} \left( \lambda \frac{\partial T}{\partial z} \right) + \dot{q} = \rho c_p \frac{\partial T}{\partial t}. \]  

(5.2)

In this equation \( \dot{q} \) is the heat source term and \( c_p \) the specific heat of the medium. It is often possible to work with a simplified version of this equation. When the thermal conductivity is constant, and the heat source term is zero \( (\dot{q}=0) \), this results in

\[ \frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} + \frac{\partial^2 T}{\partial z^2} = \frac{\lambda}{\rho c_p} \frac{\partial T}{\partial t}. \]  

(5.3)

where \( \lambda/\rho c_p \) is the thermal diffusivity. For a one dimensional case the boundary conditions are \( T(x,0) = T_1 \) and \( T(0,t) = T_3. \) This results in the solution

\[ \frac{T(x,t) - T_s}{T_1 - T_s} = \text{erf} \left( \frac{x}{2 \sqrt{\rho c_p t}} \right). \]  

(5.4)

In this equation \( \text{erf}(..) \) is the error function [5] defined as

\[ \text{erf} x = \frac{2}{\sqrt{\pi}} \int_0^x e^{-w^2} dw. \]  

(5.5)

5.3 Convection

In convection diffusion contributes to the heat-transfer but the dominant contribution is generally made by the bulk or gross motion of the fluid. In this section two different types of convection will be treated, the single phase flow and two-phase flow heat-transfer.

5.3.1 Single phase flow

Starting with the heat-transfer due to convection for single phase flows, the Newtonian approach can be used

\[ q'' = \alpha (T_\infty - T_s). \]  

(5.6)
where $\alpha$ is the heat-transfer coefficient. The temperatures $T_\infty$ and $T_s$ are defined in Figure 5.2.

$$\alpha_L = 0.023 \frac{\lambda}{D} Pr^{0.4} Re_L^{0.8}. \quad (5.7)$$

In this equation $Pr$ is the Prandtl number of the liquid.

### 5.3.2 TWO-PHASE FLOW

To examine the heat-transfer of a liquid while boiling, the Chen correlation is well known and often used [2]. In this correlation a single phase heat-transfer coefficient, $\alpha_L$, see Eq. (5.7), and the Lockhart & Martinelli parameter for turbulent-turbulent flows, $X_{tt}$, is used. The extra mixture in the liquid, caused by bubbles, increases the heat-transfer coefficient with a factor $F$:

$$\alpha = \alpha_{mic} + \alpha_L F, \quad (5.8)$$

where

$$F = \begin{cases} 1 & \text{if } X_{tt} > 10 \\ 2.35 \left[0.213 + 1/X_{tt}\right]^{0.736} & \text{if } X_{tt} \leq 10 \end{cases} \quad (5.9)$$

If there are bubbles, then $Re_L$ in Eq. 5.7 has to be replaced with $Re_L = G(1 - x) D_t / \eta_L$. The parameter $X_{tt}$ shows that there is a relation between heat-transfer and pressure drop. The Forster-Zuber ($\alpha_{FZ}$) correlation for heat-transfer at nucleate boiling is

$$\alpha_{mic} = \alpha_{FZ} S_t, \quad (5.10)$$

$$\alpha_{FZ} = \frac{0.00122\left[T_s - T_{sat}(p_L)\right]^{0.24}[p_{sat}(T_s) - p_L]^{0.75} \lambda_L^{0.79} \sigma_{PL}^{0.45} \rho_L^{0.49}}{\sigma^{0.5} \eta_L^{0.29} C^{0.24} \rho_G^{0.24}}, \quad (5.11)$$
where $\mathcal{L}$ is the latent heat, $T_{\text{sat}}$ and $p_{\text{sat}}$ the saturation temperature and pressure. Bubble creation is suppressed by bulk flow when it flows by. That is why there is a suppression factor $S_{f}$ in $\alpha_{\text{mic}}$:

$$S_{f} \approx \left( 1 + 2.56 \cdot 10^{-6} R_{e}^{1.17} \right)^{-1}, \quad (5.12)$$

where

$$R_{e} = \frac{\rho_{L}}{\rho_{G}} \frac{L}{F(X_{tt})} \frac{1}{\varepsilon_p}$$

Here is $F = F(X_{tt})$ written to indicate the dependency of $X_{tt}$.

Bennet & Chen generalized the above mentioned correlations for all non-metallic liquids, by multiplying $\alpha_{L} F$ with $P_{r}^{0.296}$. Other correlations often use the Boiling number which is defined as:

$$Bo = \frac{q}{(GL)}. \quad (5.14)$$

Another number that is often used is the convection number

$$Ko = \left( \frac{1 - x}{x} \right)^{0.8} \left( \frac{\rho_{G}}{\rho_{L}} \right)^{0.5}. \quad (5.15)$$

Some correlations are given by Schrock & Grossmann ($\alpha_{SG}$, see Eqs. 5.16a-5.16b)

$$\alpha_{SG} = \alpha_{L} c_{1} \left\{ Bo + c_{2} X_{tt}^{-0.66} \right\}, \quad (5.16a)$$

$$c_{1} \approx 7.39 \cdot 10^{-3} \quad \text{and} \quad c_{2} \approx 1.5 \cdot 10^{-4}, \quad (5.16b)$$

and Gunger & Wingerton ($\alpha_{GW}$, see Eq. 5.17).

$$\alpha_{GW} = \alpha_{L} \left\{ 1 + 3000 Bo^{0.86} + \left( \frac{x}{1 - x} \right)^{0.75} \left( \frac{\rho_{L}}{\rho_{G}} \right)^{0.41} \right\}. \quad (5.17)$$
HALF-OPEN CO₂ PROCESSES

6.1 Introduction

To accomplish faster freezing some new processes can be used. These processes are based on the so-called half-open character. The half-open character implies that the gaseous CO₂ is being pumped out of the 'cell', which will be fluidized by compressing it in a compressor, and can be used again. This has a positive effect on the efficiency of the process.

Some companies who already work with liquid CO₂, like Air Liquide, Air Products, Hoekloos, Hydrogas and AGA, could benefit from these innovations. The processes that are examined theoretically are:

1. Suction Process: In a cell, liquid CO₂, at a pressure of 5.2 bar and a temperature of 223 K, is sprayed with nozzles on the product. The cell pressure is 1 bar. The liquid CO₂ will, due to the cell pressure, solidify by adiabatic expansion. The temperature of the solid CO₂ will decrease till 193 K.

![Diagram of Suction Process with a half-open CO₂ character.](image)

**Figure 6.1:** Suction process with a half-open CO₂ character.

The product will transfer heat towards the solid CO₂, and the product will hereby drop in temperature. The absorbed heat, by the solid CO₂, will result in sublimation of the
CO$_2$ until it is completely gaseous. After a while the product will reach the desired temperature. The gaseous CO$_2$ can be pumped out of the cell and will be compressed and condensed. This can be done with a two-step cascade system. From here the process can start over again. This process generates some questions and bottlenecks which will be discussed in section 6.4.1.

2. HEAT-PIPE PROCESS: Liquid CO$_2$ is sprayed on the product in the cell at a pressure of 7.3 bar. Due to the heat-transfer from the product towards the liquid CO$_2$, the liquid CO$_2$ evaporates. The product drops in temperature, and the gaseous CO$_2$ ascends.

![Diagram of heat-pipe process](image)

**Figure 6.2:** heat-pipe process with a half-open CO$_2$ character.

In top of the cell a pipe is present at a temperature of 223 K. The gaseous CO$_2$ condenses on the pipe and falls down on a drip tray. The captured liquid CO$_2$ is sprayed on the product again, so the process can start over again. The questions and bottlenecks of this process will be discussed in section 6.4.2.

The following advantages of these processes are expected with respect to existing processes:

- Good heat-transfer between CO$_2$ (solid as well as liquid) and the product.
- Fast freezing due to the low temperatures of solid/liquid CO$_2$.
- High efficiency of the freezing process, so also lower energy costs. The currently used refrigerant (N$_2$) has probably higher energy costs, because the refrigerant is not recycled in the process.

This study has the purpose to compare the new processes with already used processes. Also to get a better view on the bottlenecks that could emerge when the processes are developed further.

A first draft and explanation of the two processes will be given. Both processes will also be examined thermodynamically and finally an example, using fish as reference, will be given to compare the processes with each other. Contiguous some bottlenecks will be discussed. Finally a conclusion and some recommendations will be given.
6.2 Enthalpy - pressure diagrams

6.2.1 Suction process

A thermodynamical analysis, based on an enthalpy pressure diagram \((H-p)\), can give more insight in the process. In Figure 6.3 the \(H-p\) diagram can be seen for the suction process as described in the Introduction. In this figure isotherms are drawn with temperatures of 263 K and 308 K. Also the coexistence areas for gas and liquid, and for solid and gas are present. The purple bold line is the suction process. In this figure are four points \((A, B, C\) and \(D)\). These points are used to describe the process.

![Figure 6.3: H-p diagram of CO\(_2\) for the suction process.](image)

At point \(A\), liquid \(CO_2\) is present with a pressure of 7.3 bar, and a temperature of 223 K. Adiabatic expansion down line \(AB\) transforms the liquid \(CO_2\) into 58% solid and 42% gaseous \(CO_2\). The temperature drops till 193 K. The solid \(CO_2\) evaporates isothermally according to line \(BC\). The specific enthalpy difference \(\Delta h\) is 325 kJ/kg. From point \(C\) the gaseous \(CO_2\) is pumped out and isentropic compressed until it has a pressure of 7.3 bar (line \(CD\)). Finally the compressed \(CO_2\) enters a condenser and becomes gaseous again (line \(DA\)). From here the process can start all over again.

6.2.2 Heat-pipe process

The heat-pipe process which is already described in the Introduction, is plotted in a \(H-p\) diagram. This can be seen in Figure 6.4. At point \(A\) gaseous \(CO_2\), at a temperature of 293 K and a pressure of 1 bar, is compressed (line \(AB\)) till 7.3 bar. Along line \(BC\) it condenses and reaches a temperature of 223 K. Now it is ready to be used in the heat-pipe process. The liquid \(CO_2\) is sprayed on the product and evaporates (line \(CD\)) due to the absorbed heat from the product. The specific enthalpy difference \(\Delta h=345\) kJ/kg. Subsequently the gaseous \(CO_2\) condenses against the pipe in top of the cell (line \(DC\)). The condensed \(CO_2\) can be used again.
6.3 Calculations on freezing of fish

6.3.1 General

To investigate the feasibility of the processes, the freezing of fish (see Table 6.1 for properties) is taken as reference. The questions which rise are: How long will it take to freeze the fish, and which amount of CO₂ (liquid/solid) is necessary? Finally the energy consumption is calculated with STIMEK (version 24) to make a comparison in energy efficiency for each process. A freezing tunnel (ADVANTEC) located at Northseafood Holland in Urk (see Table 6.2 for characteristics) is used as reference in these calculations.

<table>
<thead>
<tr>
<th>Name</th>
<th>Value</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>Density</td>
<td>1047</td>
<td>[kg/m³]</td>
</tr>
<tr>
<td>Heat capacity</td>
<td>3.34</td>
<td>[kJ/kgK]</td>
</tr>
<tr>
<td>Heat conduction coeff.</td>
<td>0.42</td>
<td>[W/Km]</td>
</tr>
<tr>
<td>Thermal diffusivity</td>
<td>0.120</td>
<td>[mm²/s]</td>
</tr>
<tr>
<td>Dimension fish</td>
<td>128 × 64 × 8</td>
<td>[mm]</td>
</tr>
<tr>
<td>Mass fish</td>
<td>68</td>
<td>[g]</td>
</tr>
<tr>
<td>Surface fish</td>
<td>0.023</td>
<td>[m²]</td>
</tr>
<tr>
<td>Temperature in</td>
<td>281</td>
<td>[K]</td>
</tr>
<tr>
<td>Temperature out</td>
<td>255</td>
<td>[K]</td>
</tr>
<tr>
<td>Enthalpy difference</td>
<td>220</td>
<td>[kJ/kg]</td>
</tr>
</tbody>
</table>

Table 6.1: Properties of fish necessary for the calculations.
6. HALF-OPEN CO₂ PROCESSES

<table>
<thead>
<tr>
<th>Properties</th>
<th>Value</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>Freezing time fish (68 grams)</td>
<td>162</td>
<td>sec</td>
</tr>
<tr>
<td>Production</td>
<td>1000</td>
<td>kg/hour</td>
</tr>
<tr>
<td>Capacity</td>
<td>218</td>
<td>kW</td>
</tr>
<tr>
<td>Product temperature</td>
<td>255</td>
<td>K</td>
</tr>
<tr>
<td>Evaporation temperature</td>
<td>241</td>
<td>K</td>
</tr>
</tbody>
</table>

Table 6.2: Properties of reference ADVANTEC Frigoscandia located at Northseafood Holland.

6.3.2 FREEZING TIME FISH AND REQUIRED AMOUNT CO₂

6.3.2.1 Suction process

To determine the freezing time of fish and the required amount of CO₂ for freezing fish of 68 grams each, the data given in Table 6.1 is necessary. Together with Eq. (5.4) this results in a freezing time of 220 seconds. The required amount CO₂ can be determined with Figure 6.3 and the specifications given in Table 6.1. According to line BC in the figure, 1 kg CO₂ provides 325 kJ. To freeze one fish costs 15 kJ. To freeze 1 kg fish requires 0.68 kg CO₂.

6.3.2.2 Heat-pipe process

The most important difference of the heat-pipe process between the suction process, is that freezing now occurs through convection and conduction instead of only conduction. To make an estimation of the freezing time there has to be made an assumption of the velocity of the liquid CO₂ flowing along the fish. This is assumed to be 1 m/s. After applying the data of fish and the assumption made above this results in a freezing time of 70 seconds. The amount CO₂ that is required can be calculated in almost a similar way as in the previous section. The only difference is the specific enthalpy, see line CD in Figure 6.4, which is 345 kJ for 1 kg of CO₂. This results in 0.64 kg CO₂ is required to freeze 1 kg of fish.

6.3.3 ENERGY CONSUMPTION

6.3.3.1 Suction process

Freezing 1 kg fish requires 0.68 kg of CO₂; 1 kg CO₂ costs 0.068 EUR. The costs for freezing 1 kg of fish are 0.046 EUR. The production, according to the specifications of the reference, is 1000 kg/hour. This implies a capacity per year of 2,000 tons (50 weeks, 40 hours/week). The total savings on the operational costs of the suction process compared to a freezing tunnel of Frigoscandia are: 0.046*2,000,000 = 92,000 EUR.

With STIMEK (version 24) the energy consumption has been calculated. As there is no data in STIMEK of CO₂, NH₃ is used as refrigerant for as well the reference as the new processes. This will give other absolute results, but when compared the results are relative. According to STIMEK, the yearly energy consumption is 504 MWh. According the specifications of Frigoscandia it appears that the yearly energy consumption is 400 MWh. This means a difference of 104 MWh between the reference and the new process. Because every kWh costs 0.045 EUR the extra costs are 4,647 EUR.

Freezing processes based on the refrigerant CO₂
6. HALF-OPEN CO₂ PROCESSES

The total savings on the operational costs of the suction process compared with a freezing tunnel of Frigoscandia are: 92,000 - 4,647 = 87,353 EUR

6.3.3.2 Heat-pipe process

Freezing 1 kg of fish requires 0.64 kg of CO₂. CO₂ costs 0.068 EUR/kg. The costs for freezing 1 kg of fish are 0.044 EUR. The saved costs for freezing fish at a production of 2,000 tons are 87,040 EUR.

For this process the energy consumption is also determined with the help of STIMEK (version 24). The total energy consumption is estimated at 394 MWh, which is almost similar to the consumption of the reference of Frigoscandia (400 MWh). The difference will therefore be neglected.

The total savings on the operational costs of the heat-pipe process compared with a freezing tunnel of Frigoscandia are: 87,040 EUR.

6.4 Discussion

6.4.1 Suction process

The suction process seems to have (on first sight) thermodynamically no bottlenecks. Because CO₂ is used as refrigerant in direct contact with the product, there will rise some questions concerning the applicability in this freezing process. Some of these questions are.

1. Does the formation of solid CO₂ have an influence on the quality of the product?
   The creation of solid CO₂ will probably have no influence on the quality of the product. The reason is that (solid) CO₂ is not poisonous, and causes little damage to the product. CO₂ is already been used as a cryogenic refrigerant, so it is already proven that it is harmless.

2. What happens with the nozzles after injecting CO₂ on the product?
   After injecting liquid CO₂, it will almost instantaneously be transformed into solid CO₂. This could cause troubles at the nozzles, which can get obstructed. The obstruction of the nozzles has to be prevented to keep the process running.

3. Will all products freeze according to a uniform temperature distribution?
   The goal is that all products leave the cell at the same temperature. The products have to be sprayed on evenly.

4. What kind of influence has the creation of gaseous CO₂ on the heat-transfer?
   Whenever solid CO₂ is on the products, a layer of gaseous CO₂ can be formed between the product and the solid CO₂. This has a negative influence on the heat-transfer coefficient, because this is an isolating layer. The same happens when a product is dipped in liquid nitrogen. The created gaseous nitrogen can’t escape, and stays around the product, forming an isolating layer.

5. Can the gaseous CO₂ be pumped out the cell pure enough?
The last question is of essential importance for the efficiency of the process. In the Introduction the ideal case has been described, this implies there is pure CO₂ pumped out of the cell. In reality this is almost never the case. When a mixture of CO₂ and non-condensables (oxygen, nitrogen, water, etc.) is pumped out, the performance of the total process will decrease.

### 6.4.2 Heat-pipe process

Looking at the process then there are some questions that arise. A couple of these questions are:

1. Is it possible to work at a pressure of 7.3 bar?

   According to a calculation, it appears that the wall thickness of the cell should be at least 3 millimeters for a cell of 1 meter diameter. The equation used in this calculation is

   \[ d_{\text{wall}} \geq \frac{p_{\text{cell}}D_{\text{cell}}}{2w f K + p_{\text{cell}}}, \]

   In this equation [3], \( p_{\text{cell}} \) is the pressure inside the cell, \( D_{\text{cell}} \) is the diameter of the cell, \( w \) the weakness factor of steel (0.60), \( f \) the design tension of steel (0.65), and \( K \) the tensile strength of steel (360 N/mm²). This results in a wall thickness of at least 3 mm stainless steel.

2. What is the influence on the quality of the products at such a pressure?

   Whenever there is pressure on a product, this will deform. Fries or fish contain about 60 - 65% water. Water is nearly compressed at a pressure of 7.3 bar. This implies that products that consists for a majority out of water, practically will not deform. A study [18] is used as reference. According to this study the heat-pipe process will not experience any problems working on a pressure of 7.3 bar concerning the product quality.

3. Will all products freeze according to a uniform temperature distribution?

   This question is also present in the suction process, and the answer is almost similar. The only difference is that there is chosen for a drip-tray, so the condensed CO₂ can be captured, and finally be distributed on the products evenly. This gives an uniform heat-transfer distribution between product and refrigerant.

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7

HORIZONTAL PLATE FREEZER PROJECT

7.1 Horizontal plate freezer in practice

7.1.1 Description HPF at Polskamp Meat B.V. ErmeLO

With a HPF it is possible to freeze a product more rapidly than with a tunnel or cell. This method is operated in several factories worldwide mostly with refrigerant R-22 or NH3. An example of such a HPF is stationed at Polskamp Meat B.V. ErmeLO in the Netherlands.

At Polskamp they freeze Pressurized Deboned Meat (PDM), from now on simply called meat. This meat is gained from chicken carcasses. The remains that are still attached to the bones are removed by putting the carcasses under high pressure. The collected meat (PDM) is put into plastic bags of about 11 kilograms and is frozen in a HPF of 29 plates. Each station contains 30 trays with 11 kilograms of meat. This means there is a total of 28 times 330 kg equals 9,240 kg meat in the plate freezer. The production is 4,900 kg meat/hour, which implies that the trays with meat are approximately 110 minutes in the plate freezer.

![Photograph of meat packages being sealed in plastic bags at Polskamp.](image)

The plate freezer uses R-22 as a refrigerant at an evaporation temperature of \(-48^\circ\text{C}\), with a refrigeration capacity of 300–350 kW.
7. Horizontal Plate Freezer Project

7.1.2 Reference Measurement

To set a reference for the HPF with refrigerant CO₂, a reference measurement is performed at Polskamp. How this is done will be described below.

ESCORT™ data loggers are designed stand alone temperature sensors, which are programmed to measure the temperature each minute. Ten of these data loggers are placed inside the plate freezer to monitor the temperature and temperature distribution during the freezing process.

The data loggers are positioned in the plate freezer as can be seen in Figure 7.2. This way the temperature distribution over the plate and over the height, which are the most important issues in HPFs, can be determined. The results of these measurements can be found in section 9.2.

7.2 Experimental setup pilot plant

To examine the influence on the HPF of changing the refrigerant from R-22 into CO₂, a pilot plant was built at TNO-MEP Apeldoorn. This pilot plant consists of two important parts, an NH₃-CO₂ two-stage cascade system (designed in cooperation with GTI), and a HPF especially designed for the refrigerant CO₂ by Samifi (Italy). The companies GTI and Samifi both are partners in this project.

7.2.1 Refrigeration installation

Cooling of the refrigerant CO₂ (here used as an evaporating secondary fluid) is performed with a two-stage cascade system using NH₃ (NH₃-CO₂-installation). A schematic overview...
of the total system can be found in Figure 7.3. In this figure red numbers are present (1-6).
Each number corresponds with an element of the installation.

Liquid CO₂, in the CO₂ separation vessel, (3), is pumped towards the plate freezer. The temperature of the plates drops. The liquid CO₂ evaporates. The gaseous CO₂ enters the heat exchanger (2) where it condenses. The liquid NH₃ at the other side of the heat exchanger evaporates. The compressors (4a and 4b) compress the gaseous NH₃. In the condenser (6) the NH₃ condenses. It is subsequently throttled over an expansion valve which will result in a pressure and temperature drop of the NH₃. The liquid NH₃ flows into the heat exchanger, where it exchanges the heat with the gaseous CO₂, that is inside there. The gaseous CO₂ condenses, and the NH₃ evaporates, so the process is a closed cycle.

7.2.2 PILOT PLANT HORIZONTAL PLATE FREEZER

The second major part in this project is the plate freezer itself. The plates are manufactured at Samifi, and have been mounted in the plate freezer by GTI KoudeTechniek. The plate freezer, consisting of four plates, can be seen in Figure 7.4.
The plates are made of aluminum (Al). A single plate has a header, with a certain amount of orifices in it which is the entrance of a channel that alternates a number of times to finally exit on the other end in a collecting header. This header leads to the suction side of the plate.

The plate freezer was placed in a cell at a temperature of +8°C. Polskamp has its plate freezer also inside a cell at this temperature. The plates in the pilot plant are numbered starting from the bottom. Plate two is the so-called test plate. It is instrumented with thermocouples and pressure sensors. It represents the reality more than the other plates since these plates do not have product above or below.

![Image of the HPF](image_url)

**Figure 7.4:** Photograph of the HPF as manufactured by Samifi and placed by GTI and TNO-MEP.

### 7.2.3 Measuring Program

This section clarifies which parameters (quantities), and where in the plant (position), will be measured to get an understanding of the following subjects:

- The pressure drop inside the plate
- The heat-transfer from CO\(_2\) to the product
- The temperature distribution over the plate
- The influence of different liquid/vapour ratios
- The freezing time at different configurations
- The energy consumption for the freezer during an experiment.

#### 7.2.3.1 Conditions

The circulation rate is defined as the ratio between the mass flow of the liquid and evaporated CO\(_2\). A circulation rate of 3 means that of 3 kg liquid CO\(_2\), 1 kg evaporates in the HPF. In
7. HORIZONTAL PLATE FREEZER PROJECT

this series of experiments only three plates were loaded with product. The upper plate (plate 4) was blocked otherwise the refrigeration capacity was too high for the NH₃-CO₂-installation. The conditions measured with at the pilot plant can be seen in Table 7.1.

<table>
<thead>
<tr>
<th>Product</th>
<th>CO₂ Ev. Temperature</th>
<th>Circulation rate</th>
</tr>
</thead>
<tbody>
<tr>
<td>Tylose</td>
<td>-25°C</td>
<td>3</td>
</tr>
<tr>
<td></td>
<td>-30°C</td>
<td>1.2 / 3 / 5</td>
</tr>
<tr>
<td></td>
<td>-37°C</td>
<td>3</td>
</tr>
<tr>
<td>Meat</td>
<td>-37°C</td>
<td>3 / 5</td>
</tr>
</tbody>
</table>

Table 7.1: Conditions measured with at the pilot plant.

7.2.3.2 Tylose Packages

At Polskamp the HPF is filled with meat. However, in our experiments, we used also tylose packages, representing meat. These packages are placed directly on the plate, so without trays. The trays were omitted to eliminate effects on the heat-transfer from the packages to the plate. Two sorts of packages were obtainable for the experiments. The first sort (1 kg) is a bulk package, and the second one (0.5 kg) is equipped with a Copper-Constantan thermocouple.

![Freezing curves of tylose packages frozen in air at -15°C](image)

Figure 7.5: Freezing curves of tylose packages in air at -15°C.

On the testplate 32 packages containing a thermocouple are placed and the rest of the plates are filled with bulk packages without thermocouple. This implies each plate contains a total weight of 400 kg of tylose packages. At Polskamp this is about 330 kg meat. This difference (≈ 20%) has an influence on the capacity of the HPF. What this influence is can be found in

Freezing processes based on the refrigerant CO₂
chapter 9.

To investigate the accuracy of the packages, they are frozen in circulating air at -15°C. The resulting characteristics can be seen in Figure 7.5.

This figure shows that the average time it takes to cool the tylose packages to -8°C is 13 hours. The deviation in the freezing time is 2 - 2.5 hours. This means a deviation of 15 - 20%. This deviation has to be taken in account in the final results. The difference can be a result of the placement of the thermocouple. When this is placed in a 'greasy' region inside the package instead of a 'watery', the freezing curve has a different characteristic. Another reason can be the position. When the thermocouple is not placed exactly in the middle of the package but more to the outside, the thermocouple will 'faster' sense a lower temperature.

7.2.3.3 Meat packages

In order to compare the tylose measurements with the reference measurement also two experiments with meat were performed. The meat was delivered in the same trays used in the factory of Polskamp, and frozen in the pilot plant.

![Figure 7.6: Photograph of the production process of meat. Here it is inserted in trays.](image)

7.2.3.4 Test cycle

Tylose packages The plates were loaded once with tylose packages at a temperature of +8°C. At Polskamp, meat is inserted at this temperature in the HPF. On the packages, containing a thermocouple (40 in total), a small piece of polystyrene was placed to be sure that there was full contact between package and plate, after putting the upper plate on the one below. After the plates were assembled, a canvas was placed to cover the whole HPF to reduce heat losses by radiation and natural convection towards the cell. Finally the CO₂ pump was switched on to start a flow of liquid CO₂ towards the HPF. At the end of the

Freezing processes based on the refrigerant CO₂
experiment the CO\textsubscript{2} pump was stopped, the canvas released, the plates rose and the heater, present in the cell, was activated to start the defrosting process. Whenever the packages were defrosted until +8°C, another experiment is ready to start.

**Meat packages** During the experiments with meat, plate two was loaded with meat; the other ones still contained tylose packages. Plate two was loaded with meat and in the meantime a single thermocouple was placed into each tray (see Figure 7.7).

![Thermocouple Diagram](image)

**Figure 7.7:** Method how the thermocouple is placed into the meat.

After inserting the thermocouples, the plates were lowered and the canvas was put in place. Finally the NH\textsubscript{3}-CO\textsubscript{2} was started, and the experiment could begin. At the end of the experiment the installation was stopped, the canvas released, the plates rose. To perform a new experiment the tylose packages were not defrosted, the meat on the other hand needs to be removed from the test plate and 30 new trays of meat, and also the thermocouples, have to be inserted.

### 7.2.4 Measuring points

![Sensor Diagram](image)

**Figure 7.8:** Positions of the sensors on the total plate freezer (left) and on plate 2 (right).

During the experiments several parameters are monitored. The quantities that qualify the HPF are the freezing time of the product, plate temperature distribution, pressure drop inside the plate, heat-transfer of CO\textsubscript{2} towards the product and the energy consumption of the system.

In Figure 7.8 the positions of the various sensors on the plates and the total plate freezer can be seen. The points marked with \(\Delta p\) and \(p\) are respectively pressure drop and absolute pressure points. Points marked with \(T\) are temperature points. The red dots in the right
7.2.4.1 Tylose packages

On plate two, the test plate, 32 packages contained a thermocouple. On plate one and three, there were each 4 packages with a thermocouple. On the test plate the packages are placed on almost every channel and spread over the channel.

7.2.4.2 Meat packages

Plate one and three were used in the same way as in the experiments with tylose packages. The only difference was plate two. On this plate 30 trays with meat containing a thermocouple were placed.

7.2.5 DATA ACQUISITION

The data acquired during an experiment was processed with a data logger containing 150 separate channels. Each minute a scan is made of all these channels. The development during an experiment can be traced with the help of the software program HPVEE.

7.3 Performance

7.3.1 INTRODUCTION

The performance of the HPF plant is characterized by a couple of parameters determined during the experiments. Before the results of the experiments were reliable, some troubleshooting was necessary to eliminate uncertainties.

7.3.2 TROUBLESHOOTING HPF PLANT

In the first experiments there were problems with the NH₃-CO₂-installation and the tylose packages. In the end it showed the problems were mainly based on the big differences in freezing time of the tylose packages.

7.3.2.1 Capacity problem

Initially, the refrigeration capacity was extremely high. This was caused by the high mass flow over the condenser (2 in Figure 7.3), due to the high amount of evaporation inside the plates in the beginning of the experiment. The hot gas temperature of compressor 4a therefore rose. Changing the intermediate pressure (pressure between compressor 4a and 4b in the flash gas vessel 5) transferred the problem towards compressor 4b and led to an extreme high liquid level in the flash gas vessel. This was solved by blocking the upper plate in the HPF pilot plant to lower the refrigeration capacity, and to increase the security of the hot gas temperature from 85°C till 100°C.

At the end of the experiment the refrigeration capacity decreases, due to low evaporation.
inside the plates. As a result the evaporation temperature can not be controlled and decreases rapidly. To test at a constant evaporation temperature, two cylinders of compressor 4a were bypassed.

7.3.2.2 Tylose packages

According to the first experiments a huge deviation was present in the freezing time of the tylose packages. Some packages, reached the desired end temperature of -18°C only after many hours so it is impossible to determine the operation of the HPF, or the temperature distribution. Possible reasons for the behaviour are

Ice formation Because the packages were frozen and defrosted after every experiment, and stayed on their position, condensed water dripped on the plate between the plate and the packages. When a new experiment starts, this thin layer of water freezes, and a thin layer of ice between the packages and plate is formed. This harms the freezing process because the thermal conductivity of ice and aluminum are respectively \( \lambda_{\text{ice}} \approx 2.2 \text{ W/Km}, \lambda_{\text{Al}} \approx 240 \text{ W/Km} \) [22]. To solve this problem, pressurized air was blown into the cell, where the HPF was placed, to lower the dew point till \(-20°C\) (relative humidity approximately 10%). The condense almost vanished. To be sure there was no layer of water between package and plate, all packages were examined before starting a new experiment.

Package deformation The tylose packages, in the first experiments, were placed really close together; clamped between the edges of the plates in the HPF pilot plant. During the freezing process, the packages expand. (The composition of tylose is 80% water, so it will expand 8% volume fraction when frozen). As these packages could not expand horizontally, the packages started to expand in the vertical direction. After lifting the plates, at the end of the experiment, the packages seemed to have formed a sort of 'bridge' where the packages in the middle of the plate did not make any contact with the plate any more. This affects the heat-transfer because air (between package and plate) is a good insulator (\( \lambda_{\text{air}} \approx 0.02 \text{ W/Km} \)). To eliminate this problem, the packages were placed 'free' so they were able to expand in all three dimensions during the freezing process.

Bad contact between plate and packages Due to freezing and defrosting the packages, the composition changes and therefore the bottom deforms. When the packages are frozen again, after defrosting, the packages will not have 100% contact with the plate, which has a negative effect on the heat-transfer. Polystyrene layers were placed on the packages, so after dropping the upper plate, the packages will be squeezed on the plate. This implies better contact and therefore better heat-transfer.

Convection as a consequence of the Cell Regulation The HPF pilot plant is placed in a cell with a climate regulation program containing a heater system placed above the HPF. During the experiments the cell temperature was regulated at a temperature of +8°C. The plate freezer absorbs heat from the environment (due to natural convection, conduction and radiation), so the cell temperature will decrease when the heater system is not regulated. When the heater is on, hot air is flowing over the edges of the HPF. This initiates an extra heat load, and the packages placed at the edges of the plates will warm up. To prevent this,
the heater was not active during the experiments and the cell temperature dropped from +8°C till +4°C.

**Deviation in characteristics of packages** The last element that could have an effect on the freezing time is the deviation in the properties of the tylose packages itself. This phenomenon is already described briefly in section 7.2.3.2. The packages where frozen in air of -15°C till they reached a temperature of -10°C. The effect of the contact between plate and package is hereby eliminated and therefore the deviation can be determined. The maximum deviation in the packages was approximately 15 - 20%. This deviation has to be taken in account while determining the freezing time of the packages in the experiments.

The difference can be a result of the placement of the thermocouple. When this is placed in a 'greasy' region inside the package instead of a 'watery', the freezing curve has a different characteristic. Another reason can be the position. When the thermocouple is not placed exactly in the middle of the package but more to the outside, the thermocouple will 'faster' sense a lower temperature.

### 7.3.3 Parameter determination

In this section is described how the parameters (refrigeration capacity of the HPF, freezing time, energy consumption and heat-transfer coefficient) are determined.

#### 7.3.3.1 Refrigeration capacity plate freezer

The time-dependent refrigeration capacity, $P_{pf}(t)$, of the plate freezer can be determined by the equation

$$ P_{pf}(t) = \left( m(t) \Delta h \right) - \left( P_{pump}(t) \eta_{pump} \right) , \tag{7.1} $$

with $m(t)$ the time-dependent CO$_2$ mass flow over the condenser, $\Delta h$ the specific enthalpy difference of CO$_2$ before and after the condenser, $P_{pump}(t)$ the time-dependent electric power that is needed to drive the CO$_2$ pump (see Figure 7.3) and $\eta_{pump}$ the total efficiency of the pump.

#### 7.3.3.2 Freezing time

For the food industry the freezing time of the product is one of the most interesting issues. The way the freezing time is determined is of essential importance to compare different measurements. In this project the freezing time is determined as the difference between the moment the product is at 0°C, $(t_{start})$ and when it reaches the temperature of -18°C $(t_{end})$.

#### 7.3.3.3 Energy consumption

To compare different processes for energy consumption, several parameters can be determined. The most valid parameters are Coefficient Of Performance (COP), Carnot efficiency and COP/Carnot ratio.
The $COP$ is determined by

$$COP(t) = \frac{P_{pr}(t)}{P_{HT}(t) + P_{LT}(t)}.$$  \hspace{1cm} (7.2)

Where $P_{HT}$, $P_{LT}$ is the time-dependent shaft power for respectively the high and low temperature compressors (see Figure 7.3, 4a and 4b).

**Carnot efficiency** The time-dependent Carnot efficiency, $\eta_{carnot}(t)$, is the maximum achievable efficiency at given evaporation- and condensation temperatures. This is equal to

$$\eta_{carnot}(t) = \frac{T_{ev}(t)}{T_c(t) - T_{ev}(t)},$$ \hspace{1cm} (7.3)

where $T_{ev}(t)$, is the time-dependent evaporation temperature, and $T_c(t)$ the time-dependent condensation temperature both in [K]!

$COP/Carnot$ ratio The last determined entity, concerning the energy consumption, is the $COP/\eta_{Carnot}$ ratio. This is Eq. (7.2) divided by Eq. (7.3).

### 7.3.3.4 Heat-transfer coefficient

The heat-transfer was determined in two ways. These where the external, and the overall heat-transfer.

**External heat-transfer coefficient** First the external heat-transfer was determined by the definition

$$\alpha_{ext}(t) = \frac{P_{pr}(t)}{A \times \left(\langle T_{pack}\rangle_p(t) - \langle T_{pr}\rangle_p(t)\right)},$$ \hspace{1cm} (7.4)

with $A$ the effective area of the plates (24 m²), $\langle T_{pack}\rangle_p(t)$, the position averaged time-dependent temperature of the packages and $\langle T_{pr}\rangle_p(t)$ the position averaged time-dependent temperature of the plates in the HPF.

**Overall heat-transfer coefficient** The time-dependent overall heat-transfer coefficient is defined as

$$\alpha_{ov}(t) = \frac{P_{pf}}{A \times \left(\langle T_{pack}\rangle_p(t) - \langle T_{CO2,pr}\rangle(t)\right)},$$ \hspace{1cm} (7.5)

with $\langle T_{CO2,pr}\rangle(t)$ the time-dependent temperature of the CO₂ inside the plate freezer.
SIMULATIONS IN MATLAB™ AND RESULTS

8.1 Introduction

With the help of MATLAB™ the behaviour of the HPF with CO₂ as refrigerant was simulated for different conditions. In this chapter the model and results of the simulations can be found. Elements which were calculated in these simulations are the pressure drop in the header, orifices and channels, distribution of liquid CO₂, and heat-transfer coefficient.

8.2 Model

In order to simulate a process certain things have to be defined. First a model is required. This is made in order to simplify the process. Finally the boundary conditions of the model have to be given.

8.2.1 Network of plate

![Diagram of pressure drop components in the plate freezer.]

FIGURE 8.1: Drawing of the pressure drop components in the plate freezer.
The HPF pilot plant (see Figure 7.4) can be drawn as a schematic network of flow resistors as can be seen in Figure 8.1. In this figure each differently colored box represents a pressure drop element; pressure drop inside the header, suction side, orifices and two phase pressure drop (inside the channels). These elements can only be determined when the boundary conditions and assumptions are well defined.

### 8.2.2 Boundary conditions and assumptions

In simulations, assumptions and boundary conditions are necessary to determine certain parameters. In the performed simulations, it is assumed that the fluid that enters the plate is pure liquid, and remains this during the entire header. In reality there is meat present on the header, and evaporation of the liquid CO$_2$ already starts here. In order to simplify the calculations, the liquid CO$_2$ will now start to evaporate after passing the orifice. The channel behind the orifice is assumed to be a straight pipe, without U-turns, of a defined length. In reality this is different, then the channel alternates a defined amount of times. This would normally contribute an extra friction factor, and therefore a higher pressure drop per channel.

The total refrigeration capacity is assumed to be constant (30 kW) during the whole process. This value is determined by the reference of 29 plates at Polskamp, which have a refrigeration capacity of 300–350 kW. So each plate has a refrigeration capacity of approximately 10 kW. In the pilot plant three plates are used, which results in 30 kW refrigeration capacity.

The liquid CO$_2$ will evaporate inside the channels, and causes a two-phase flow. The amount of gaseous CO$_2$ increases down-stream of the channel ($\text{channel}_{\text{in}} \approx 0\%$, $\text{channel}_{\text{out}} \approx (1/\text{Circulation Rate}) \times 100\%$). This has an effect on the pressure drop for two-phase flows which is hard to determine. To simplify the calculations, the circulation rate is assumed to be constant over the entire channel.

Inside the header at the suction side, the amount of gaseous CO$_2$ is assumed to be the same as $\text{channel}_{\text{out}}$. This means there will be no evaporation in this header and the pressure drop is frictional with a two-phase character.

The elements that are present in Figure 8.1 can be determined with the theory described in chapter 3 till 5. To determine the distribution of liquid CO$_2$ over each channel an electrical analagon can be used to determine the substitution resistance of the network. In a parallel network it is most common to use the equation

$$\frac{1}{R_{\text{subst}}} = \sum_{i=1}^{N} \frac{1}{R_i},$$

(8.1)

For a serial network the equation is

$$R_{\text{subst}} = \sum_{i=1}^{N} R_i.$$  

(8.2)
8. SIMULATIONS IN MATLAB™ AND RESULTS

8.2.3 Conditions for simulations

The conditions that were varied in the simulations are the circulation rate and the evaporation temperature of CO₂. This implies a difference in the value of the physical properties of CO₂, and mass flow of liquid CO₂ entering the plates. The conditions are given in Table 8.1.

<table>
<thead>
<tr>
<th>( T_{ev} )</th>
<th>Circulation Rate</th>
</tr>
</thead>
<tbody>
<tr>
<td>-30°C</td>
<td>1.2 / 3 / 5</td>
</tr>
<tr>
<td>-40°C</td>
<td>1.2 / 3 / 5</td>
</tr>
</tbody>
</table>

Table 8.1: Conditions simulated with at the pilot plant.

The reason for these simulations, is that the experiments (see Table 7.1) were performed under almost the same conditions.

8.3 Results

In the simulations different parameters were determined with the help of the above mentioned assumptions and boundary conditions. The results will follow in a separate section for each parameter.

8.3.1 Pressure drop header

Because of the assumption of injecting liquid CO₂ into the plates and that remains liquid all over the header the pressure drop will have a one phase character inside the header. The pressure drop inside the header can be determined for as well a laminar flow as a turbulent flow, which is described in Chapter 3. Whether the flow is turbulent or laminar before every orifice can be determined. Still it is good to compare both flow patterns for each condition to see if there are major differences in the pressure drop.

In Figure 8.2 the absolute pressure before each orifice is plotted. The pressure drop in the header is defined as: \( \Delta p = p_{\text{end}} - p_{\text{begin}} \). According to the plot the pressure drop for turbulent flow is approximately 4 mbar, and for laminar flow 2 mbar for this specific case. For all other conditions the pressure drop inside the header is shown in Figure 8.3 and in Table 8.2.

<table>
<thead>
<tr>
<th>( T_{ev} )</th>
<th>Circulation Rate</th>
<th>Turbulent flow [Pa]</th>
<th>Laminar flow [Pa]</th>
</tr>
</thead>
<tbody>
<tr>
<td>-30°C</td>
<td>1.2</td>
<td>200</td>
<td>100</td>
</tr>
<tr>
<td></td>
<td>3</td>
<td>600</td>
<td>400</td>
</tr>
<tr>
<td></td>
<td>5</td>
<td>1,800</td>
<td>1,100</td>
</tr>
<tr>
<td>-40°C</td>
<td>1.2</td>
<td>250</td>
<td>150</td>
</tr>
<tr>
<td></td>
<td>3</td>
<td>700</td>
<td>400</td>
</tr>
<tr>
<td></td>
<td>5</td>
<td>20</td>
<td>1,300</td>
</tr>
</tbody>
</table>

Table 8.2: Results of simulations for the pressure drop in the header for different conditions.

Freezing processes based on the refrigerant CO₂
The pressure drop inside the header increases with the circulation rate. This is expected, because the mass flow through the header increases, and also the frictional pressure drop will. The reason the pressure drop at lower evaporation temperatures is higher, has to do with the physical properties of CO\textsubscript{2}. The viscosity and density will increase as an influence of the temperature. According to the theory of single phase flow this implies a higher pressure drop due to friction.

The pressure drop inside the header is of such minor importance that it can be neglected in the network. This implies the distribution of liquid CO\textsubscript{2} is equal for each orifice ($\frac{m}{\#\text{orifices}}$). It seems the dimension of the header is designed big enough as recommended in [10].
8. SIMULATIONS IN MATLAB™ AND RESULTS

8.3.2 DISTRIBUTION OF LIQUID CO₂ INSIDE THE PLATES

Knowing now that the mass flow is equal for each element, the mass flow under different conditions can be determined using the following equation

\[ \dot{m} = \frac{P_{pf} CR}{\Delta h}, \]

where \( CR \) is the circulation rate. As we know the refrigeration capacity of the plate freezer (30 kW), the specific enthalpy difference (302 kJ/kg for CO₂ at -30°C, and 320 kJ/kg at -40°C) and the initiated circulation rate, it is easy to determine the mass flow over the total plate freezer. This results in the following mass flows for different conditions as can be seen in Figure 8.4 and Table 8.3.

![Mass flow over the plate freezer](image)

**Figure 8.4:** Mass flow over the plate freezer for different circulation rates (CR) and evaporation temperatures. At the x-axis the evaporation temperature [°C] is shown, and on the y-axis the mass flow, in [kg/s].

<table>
<thead>
<tr>
<th>( T \text{[°C]} )</th>
<th>Circulation Rate</th>
<th>Mass flow [kg/s]</th>
</tr>
</thead>
<tbody>
<tr>
<td>-30°C</td>
<td>1.2</td>
<td>0.119</td>
</tr>
<tr>
<td></td>
<td>3</td>
<td>0.298</td>
</tr>
<tr>
<td></td>
<td>5</td>
<td>0.497</td>
</tr>
<tr>
<td>-40°C</td>
<td>1.2</td>
<td>0.112</td>
</tr>
<tr>
<td></td>
<td>3</td>
<td>0.282</td>
</tr>
<tr>
<td></td>
<td>5</td>
<td>0.469</td>
</tr>
</tbody>
</table>

**Table 8.3:** Results of the mass flow over the plate freezer for different conditions.

The mass flow per channel is the total mass flow divided by the number of plates (in this case 3) times the number of channels per plate. This can be said because of the conclusion in the previous section.
8.3.3 Pressure drop orifices

After determining the mass flow per channel in the previous section, it is possible to determine the pressure drop induced by each orifice with the help of the Bernoulli equation (Eq. 3.8). In combination with the equal mass flow through each orifice (found in the previous section) this results in a pressure drop caused by the orifice for different conditions as shown in Figure 8.5 and Table 8.4.

![Figure 8.5: Pressure drop of the orifice for different circulation rates (CR) and evaporation temperatures. At the x-axis the evaporation temperature [°C] is shown, and on the y-axis the pressure drop, in [Pa].](image)

<table>
<thead>
<tr>
<th>T&lt;sub&gt;ev&lt;/sub&gt;</th>
<th>Circulation Rate</th>
<th>Pressure drop [Pa]</th>
</tr>
</thead>
<tbody>
<tr>
<td>-30°C</td>
<td>1.2</td>
<td>1,800</td>
</tr>
<tr>
<td></td>
<td>3</td>
<td>11,600</td>
</tr>
<tr>
<td></td>
<td>5</td>
<td>35,000</td>
</tr>
<tr>
<td>-40°C</td>
<td>1.2</td>
<td>1,895</td>
</tr>
<tr>
<td></td>
<td>3</td>
<td>12,000</td>
</tr>
<tr>
<td></td>
<td>5</td>
<td>36,850</td>
</tr>
</tbody>
</table>

Table 8.4: Results of simulations for the pressure drop due to the orifice for different conditions.

The pressure drop due to the orifice increases when the circulation rate is increased. This is very good to understand, because the pressure drop is dependent of the mass flow, and this becomes higher when the circulation rate is increased. Looking at the influence of the evaporation temperature it is obvious that the pressure drop is a fraction higher (± 5%). This is exactly the difference in density for liquid CO<sub>2</sub> at the two different evaporation temperatures. According to the theory, the density is the only variable in Eq. (3.8) that changes at a different evaporation temperature.
8.3.4 Two-phase pressure drop (inside channels)

Inside the channels of the HPF, liquid CO$_2$ will evaporate. Due to the evaporation a mixture of gas and liquid will be created. The flow pattern changes from a single phase character into a two-phase character which causes a pressure drop. This pressure drop can be divided in two different elements: frictional and accelerational pressure drop. Both can be calculated using two methods: Lockhart & Martinelli and Friedel. The theory of these methods can be found in section 4.4. For different conditions the results for two methods (Friedel and Lockart & Martinelli) can be found in Figure 8.6 and Table 8.5.

The accelerational pressure drop is much lower than the frictional pressure drop. The accelerational pressure drop is caused by the increase of the specific volume (and the average velocity) of the flow during evaporation. The frictional pressure drop is the pressure drop caused by the flow of the mixture, which changes continuously of composition. Compared with [10] these results emerge the same conclusion. In [10] the pressure drop due to two-phase flow is much lower than the pressure drop generated by the orifice. This implies the pressure drop over the orifice is dominant, so the pressure drop element due to two-phase flow is negligible.

Looking at both methods, under the same conditions, it shows that the frictional pressure drop is much higher for the Lockhart & Martinelli method than for the Friedel method. The difference decreases at higher circulation rates. A reason is that the Friedel method is keen for liquid flow and Lockhart & Martinelli more for gas flow. As there is almost no liquid flowing through the channel at a low circulation rate (and so only gas is), the frictional pressure drop for the Friedel method will be much lower than for the Lockhart & Martinelli method. Increasing the circulation rate increases the liquid mass flow and therefore also the frictional pressure determined with Friedel.

Freezing processes based on the refrigerant CO$_2$
8. SIMULATIONS IN MATLAB™ AND RESULTS

<table>
<thead>
<tr>
<th></th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Friedel</strong></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Accelerational</td>
<td>7</td>
<td>20</td>
<td>10</td>
</tr>
<tr>
<td>Frictional</td>
<td>8</td>
<td>370</td>
<td>1,150</td>
</tr>
<tr>
<td>Total</td>
<td>15</td>
<td>390</td>
<td>1,160</td>
</tr>
<tr>
<td><strong>L &amp; M</strong></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Accelerational</td>
<td>10</td>
<td>10</td>
<td>20</td>
</tr>
<tr>
<td>Frictional</td>
<td>110</td>
<td>590</td>
<td>1,050</td>
</tr>
<tr>
<td>Total</td>
<td>120</td>
<td>600</td>
<td>1,070</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th></th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Friedel</strong></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Accelerational</td>
<td>8</td>
<td>22</td>
<td>12</td>
</tr>
<tr>
<td>Frictional</td>
<td>8</td>
<td>380</td>
<td>1,180</td>
</tr>
<tr>
<td>Total</td>
<td>16</td>
<td>402</td>
<td>1,192</td>
</tr>
<tr>
<td><strong>L &amp; M</strong></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Accelerational</td>
<td>12</td>
<td>13</td>
<td>23</td>
</tr>
<tr>
<td>Frictional</td>
<td>120</td>
<td>600</td>
<td>1,090</td>
</tr>
<tr>
<td>Total</td>
<td>132</td>
<td>613</td>
<td>1,113</td>
</tr>
</tbody>
</table>

Table 8.5: Results of simulations for the pressure drop due to two-phase flow at -30°C and -40°C for different circulation rates.

8.3.5 PRESSURE DROP SUCTION SIDE

Knowing that the CO₂ will not evaporate any more at the suction side (see section 8.2.2), the pressure drop at the suction side will be of the same order as the frictional pressure drop inside the channels (accelerational pressure drop is zero because no evaporation takes place). The dimension of this header is hardly any different accept for the length. The header is much shorter than a single channel. Therefore the frictional pressure drop will be smaller than inside the channels. Due to this, the pressure drop at the suction side can also be neglected. This implies that the only dominant factor is the pressure drop created by the orifice. This is exactly what is desired otherwise the system would become instable.

Freezing processes based on the refrigerant CO₂
8. SIMULATIONS IN MATLAB™ AND RESULTS

8.3.6 HEAT-TRANSFER COEFFICIENT

The heat-transfer coefficient (α) can be determined with different methods. The used methods can be found in Eq. (5.7), (5.16a), and (5.17). Results of the simulations for different conditions can be seen in Figure 8.7 and Table 8.6.

![Heat-transfer coefficient for different models](image)

**Figure 8.7:** Heat-transfer coefficient at different circulation rates (CR) and evaporation temperatures for different models. At the x-axis the conditions are shown, and on the y-axis the heat-transfer coefficient, in [W/m²K].

<table>
<thead>
<tr>
<th>$T_{ev}$</th>
<th>Circulation Rate</th>
<th>$\alpha_L$ [W/m²K]</th>
<th>$\alpha_{SG}$ [W/m²K]</th>
<th>$\alpha_{GW}$ [W/m²K]</th>
</tr>
</thead>
<tbody>
<tr>
<td>-30°C</td>
<td>1.2</td>
<td>40</td>
<td>400</td>
<td>770</td>
</tr>
<tr>
<td></td>
<td>3</td>
<td>160</td>
<td>440</td>
<td>800</td>
</tr>
<tr>
<td></td>
<td>5</td>
<td>210</td>
<td>680</td>
<td>1,500</td>
</tr>
<tr>
<td>-40°C</td>
<td>1.2</td>
<td>50</td>
<td>410</td>
<td>7800</td>
</tr>
<tr>
<td></td>
<td>3</td>
<td>170</td>
<td>450</td>
<td>820</td>
</tr>
<tr>
<td></td>
<td>5</td>
<td>220</td>
<td>700</td>
<td>1,600</td>
</tr>
</tbody>
</table>

**Table 8.6:** Results of simulations for the heat-transfer coefficient inside the channels for different conditions and different methods (single phase flow (only liquid), two-phase flow (Schrock-Grossman, Guger-Wingerton.))

According to the results, the heat-transfer coefficient in single phase flows rises when the circulation rate increases. This is very good to understand because the mass flow increases at a higher circulation rate, so will the velocity, which improves the heat-transfer. Looking at the two-phase flow heat-transfer coefficient, one can see a similar effect. The value increases at a higher circulation rate. This is because, the lower the gas fraction (circulation rate high), the two-phase flow pattern will change from stratified flow into plug flow, which has a positive effect on the heat-transfer [14] (see Figure 4.1 for the flow profile).

*Freezing processes based on the refrigerant CO₂*
EXPERIMENTAL RESULTS

9.1 Introduction

As discussed in section 7.2.3.1, 7 experiments were performed under various conditions in the HPF pilot plant. During these experiments several parameters were monitored. In order to put these results in perspective, the results of the reference measurement at Polskamp will be described first. After this the elements monitored in the experiments (pressure drops, refrigeration capacity of the HPF, freezing time of the product, energy consumption and the heat-transfer coefficient) will be shown. These will be compared with the simulations and the reference measurement if possible.

9.2 Reference measurement R-22

FIGURE 9.1: Freezing curves of meat at Polskamp. The sensor numbers are the barcodes of the data loggers (for position see Figure 7.2).

To say something about CO₂ in a HPF compared to other refrigerants a reference measurement is necessary. This was performed at Polskamp Meat Industry B.V. Ermelo. The HPF
there is cooled with R-22 at an evaporation temperature of \(-48^\circ C\). The way the tests were performed can be found in section 7.1.2. The only parameter that was determined is the freezing curve. The results can be seen in Figure 9.1.

The freezing time of these packages is determined as described in section 7.3.3.2. The starting time is when the first package crosses the \(0^\circ C\) line. The end is at \(-18^\circ C\). This gives an average freezing time of (60 ± 10) minutes.

9.3 Experimental results at HPF pilot plant

9.3.1 Pressure drop header/plate

During the experiments the pressure drop of the header as well as of the complete plate were monitored. This was done by placing two pressure drop sensors on specific places (see right in Figure 7.8 for the position). In Figure 9.2 the pressure drop of the header and the complete plate during an experiment can be seen.

![Pressure drop header](image1)

![Pressure drop complete plate](image2)

**Figure 9.2:** Pressure drop of the header (top) and the complete plate (bottom) at \(-30^\circ C\) and CR 3.

Looking at the pressure drop of the complete plate (bottom), the first 10 minutes of the experiment no pressure drop is measured. This can be understood knowing that the CO\(_2\) pump was started after 10 minutes, which started the mass flow over the HPF. On the other hand the pressure drop sensor of the header monitors a value from the beginning and hardly
changes during the experiment. It seems the sensor is measuring an off-set. This phenomenon is investigated for all experiments and it seemed that only during the meat experiments the sensor measured a useful value. For these two experiments the maximum and minimum value will be given (see Table 9.1).

The pressure drop of the complete plate has three interesting points. These points, for the experiment in Figure 9.2, are when the CO₂ pump is switched on (t=10 min), approximately halfway the experiment (t=90 min), and when the experiment is almost at the end (t=180 min). During the experiment the amount of CO₂ that evaporates decreases. To measure at a constant circulation rate, the mass flow has to be decreased, and causes the change in pressure drop.

The pressure drop of the plate, for the experiment in Figure 9.2, decreases from approximately 1.5 bar till 0.7 bar and in the end 0.5 bar. The pressure drop in the header (maximum and minimum) as well as the pressure drop in the complete plate at the three points, for all experiments, can be found in Figure 9.3 and Table 9.1.

The pressure drop in the header, in the meat experiments, shows a maximum value of 10 mbar. This is in accordance with the simulations (section 8.3.1).

9.3.2 Refrigeration capacity plate freezer

The refrigeration capacity of the HPF pilot plant can be determined using Eq. 7.1. In Figure 9.4 the capacity during an experiment can be seen. In this figure three important points can be seen. These three points correspond with the same points as described in the previous section (Pressure drop header/plate). At these points the refrigeration capacity changes. For
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<table>
<thead>
<tr>
<th>Product</th>
<th>$T_{ev}$</th>
<th>CR</th>
<th>$\Delta p$ header [mbar]</th>
<th>$\Delta p$ plate [bar]</th>
</tr>
</thead>
<tbody>
<tr>
<td>-25°C</td>
<td>3</td>
<td></td>
<td>off-set</td>
<td>1.3 - 0.8 - 0.5</td>
</tr>
<tr>
<td>TYLOSE</td>
<td>-30°C</td>
<td>1.2</td>
<td>off-set</td>
<td>1.5 - 0.5 - 0.1</td>
</tr>
<tr>
<td></td>
<td></td>
<td>3</td>
<td>off-set</td>
<td>1.5 - 0.7 - 0.5</td>
</tr>
<tr>
<td></td>
<td>-37°C</td>
<td>3</td>
<td>off-set</td>
<td>1.5 - 1.2 - 0.5</td>
</tr>
<tr>
<td>MEAT</td>
<td>-37°C</td>
<td>3</td>
<td>4 - 10</td>
<td>1.7 - 0.5 - 0.3</td>
</tr>
<tr>
<td></td>
<td></td>
<td>5</td>
<td>4 - 10</td>
<td>1.5 - 1.0 - 0.8</td>
</tr>
</tbody>
</table>

Table 9.1: Results of the experiments for the pressure drop in the header and the complete plate for different conditions.

Figure 9.4: Refrigeration capacity of the plate freezer during an experiment with $CR=3$, and evaporation temperature -30°C.

In this experiment these values are 80 kW, 40 kW, and 25 kW. Results for the other experiments can be found in Figure 9.5 and Table 9.2.

Comparing the experiments of meat (CR=5) and tylose (CR=3) at -37°C, the refrigeration capacity is lower in the meat experiment. The mass of meat on the plate was 330 kg compared to 400 kg of tylose in the tylose experiments. This implies a difference of 20% which can be found in the results.

Another difference can be seen in the refrigeration capacity for the meat experiments at CR=3 and CR=5. In the experiment with CR=3 the tylose packages (present on plate 1 and 3) were already at -18°C when starting the experiment, but with CR=5 all packages needed to be frozen (see section 7.2.3.4, Meat). This results in a lower refrigeration capacity of the plate freezer at CR=3 compared with CR=5.
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Figure 9.5: Refrigeration capacity of the plate freezer for all experiments. At the x-axis the conditions are shown, and on the y-axis the refrigeration capacity, in [kW].

<table>
<thead>
<tr>
<th>Product</th>
<th>$T_e$</th>
<th>CR</th>
<th>Refr. capacity [kW]</th>
</tr>
</thead>
<tbody>
<tr>
<td>TYLOSE</td>
<td>-25°C</td>
<td>3</td>
<td>80 - 40 - 25</td>
</tr>
<tr>
<td></td>
<td>-30°C</td>
<td>1.2</td>
<td>80 - 30 - 20</td>
</tr>
<tr>
<td></td>
<td>-30°C</td>
<td>3</td>
<td>80 - 30 - 20</td>
</tr>
<tr>
<td></td>
<td>-30°C</td>
<td>5</td>
<td>85 - 30 - 20</td>
</tr>
<tr>
<td>MEAT</td>
<td>-37°C</td>
<td>3</td>
<td>80 - 50 - 30</td>
</tr>
</tbody>
</table>

Table 9.2: Results of the experiments for the refrigeration capacity of the HPF pilot plant for different conditions.

9.3.3 FREEZING TIME

For economical reasons, the freezing time of the product is one of the most interesting issues in the food industry. The way the freezing time is determined is of essential importance to compare different measurements. In this report the freezing time is determined as told in section 7.3.3.2. An example of freezing curves for a couple of tylose packages can be seen in Figure 9.6.

A specified amount of tylose and meat packages is positioned on each channel. The freezing times of these packages are determined and averaged. This results in an average freezing time per channel, which can be seen in Figure 9.7.

The average freezing time over the complete plate is determined by averaging the freezing times of all the packages on the plate. This is done for different conditions and can be seen
9. EXPERIMENTAL RESULTS

**Figure 9.6:** Freezing curves of tylose packages at CR=3, -30°C. The time where packages cross the 0°C line, and reach a temperature of -18°C, are defined as $t_{\text{start}}$ and $t_{\text{end}}$.

**Figure 9.7:** Average freezing time of tylose packages or meat placed on a specific channel.

in Table 9.3.
9. EXPERIMENTAL RESULTS

<table>
<thead>
<tr>
<th>Product</th>
<th>$T_{ev}$</th>
<th>CR</th>
<th>Freezing time [min]</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>-25°C</td>
<td>3</td>
<td>$(105 \pm 20)$</td>
</tr>
<tr>
<td>TYLOSE</td>
<td>-30°C</td>
<td>1.2</td>
<td>$(110 \pm 35)$</td>
</tr>
<tr>
<td></td>
<td>-37°C</td>
<td>3</td>
<td>$(95 \pm 20)$</td>
</tr>
<tr>
<td></td>
<td></td>
<td>5</td>
<td>$(100 \pm 10)$</td>
</tr>
<tr>
<td>MEAT</td>
<td>-37°C</td>
<td>3</td>
<td>$(65 \pm 15)$</td>
</tr>
<tr>
<td></td>
<td>-37°C</td>
<td>5</td>
<td>$(120 \pm 30)$</td>
</tr>
</tbody>
</table>

Table 9.3: Results of the experiments for the freezing time including deviation for different conditions.

Due to a low evaporation temperature, the freezing time is the fastest at an evaporation temperature of -37°C, and a circulation rate of 3 (according to Table 9.3). The freezing time at this condition is $(65 \pm 15)$ minutes. The deviation in the freezing time is approximately 25%. According to the characteristics of the tylose packages there is a natural deviation of 15 - 20% (see section 7.2.3.2), which implies that the other 5% is a result of plate influences. The average freezing time in the reference measurement ($(60 \pm 10)$ minutes) is in accordance with this result.

Figure 9.7 shows there is one channel in the plate freezer that gave a significant deviation in the freezing time. A possibility is that there are damaged packages on this channel. However, replacing these packages still gave the same result. This implies that the temperature distribution over the plate or the distribution of liquid CO$_2$ over the plates was not uniform. Both elements are studied, and the results follow below.

**Temperature distribution** When the plate temperature is non-uniform over the plate, the packages will freeze also non-uniform. Therefore thermocouples are placed on the plate at the injection side and the suction side of the channels, to check if there is a difference in plate temperature. A result of the plate temperatures at these positions can be seen in Figure 9.8.

Looking at these figures it shows there is a slight difference in plate temperatures at the injection side and the suction side. But the difference is not visible at the specific channel. Sensor 1305 and 1310 are placed on this channel, and do not have a big deviation. This is investigated for each experiment, and gave no other results. Therefore it can be said that this cannot be the reason for the longer freezing time of the packages on this channel.

**Distribution of liquid CO$_2$** Another reason could be that there is a change of flow pattern in the header before this channel, or inside the channel which causes a different heat-transfer. According to the simulations there is a uniform distribution of liquid CO$_2$ over the channels. This means there will be no difference in distribution of liquid CO$_2$, and also in flow pattern before and inside the channels compared to each other. In other words, the first channel will have the same flow pattern as the final channel. But what if that specific channel has a con-
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**Figure 9.8:** Temperature of the plate at the injection side (top) and the suction side (bottom) of the channels.

Construction error, or pollution, at the orifice, which effects the distribution of liquid CO₂? Will that change the flow pattern inside the channel, and also the heat-transfer? Each channel has its own 'local CR', which is the same for all channels when the distribution is uniform. When the distribution of liquid CO₂ changes, the 'local CR' also changes, because there will be more or less liquid inside each channel, but the same amount will still evaporate. To investigate this phenomenon the Reynolds number, for as well liquid as gas, was calculated at different circulation rates inside the channel. The result is plotted in Figure 9.9.

According to the plot, a transition takes place at a circulation rate of approximately 2. This implies e.g. if the 'local CR' inside a channel changes from 1 into 3, the liquid flow will become turbulent, and the gas flow laminar. Will this change also have an influence on the freezing processes based on the refrigerant CO₂.
Figure 9.9: Reynolds number, for liquid and gas inside the channel, at different circulation rates. Also shown is the transition line from laminar towards turbulent flow.

two-phase flow pattern? This is determined with Eq. (4.1), (4.2), and Figure 4.2. The results can be found in Table 9.4.

<table>
<thead>
<tr>
<th>'local CR'</th>
<th>Flow pattern</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.2</td>
<td>Stratified flow</td>
</tr>
<tr>
<td>3</td>
<td>Stratified flow</td>
</tr>
<tr>
<td>5</td>
<td>Plug flow</td>
</tr>
</tbody>
</table>

Table 9.4: Two-phase flow pattern inside the channels at different 'local CR'.

From the Table it shows another transition takes place at a 'local CR' of 5. According to [14], plug flow has a better heat-transfer than stratified flow. Now knowing that there is a difference in flow pattern, thus in heat-transfer, the plate was analyzed with X-ray photographs, to see if there were any errors, or pollution, at this orifice, which can cause a different mass flow through the channel. The photographs did not show anything peculiar, so the exact reason is still unknown. A possibility can be that the channel had a smaller manufactured orifice diameter than others, or down the channel there was some pollution that was not photographed, but this is only an assumption.

9.3.4 Energy Consumption

In this section the COP (Coefficient of Performance), Carnot efficiency and COP/ηcarnot are determined with Eq. (7.2) and (7.3) for every experiment. An example of these parameters,
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during an experiment with circulation rate 3 and evaporation temperature -30°C, is given in Figure 9.10.

![COP, Carnot efficiency, (COP/\eta_{carnot}) during experiment](image)

**Figure 9.10:** COP, Carnot efficiency and COP/\eta_{carnot} ratio during an experiment at circulation rate 3 and evaporation temperature -30°C.

To get an overview of these parameters during all experiments an average is taken when these parameters were relatively steady. The values can be found in Table 9.5.

<table>
<thead>
<tr>
<th>Product</th>
<th>T_{ev}</th>
<th>CR</th>
<th>COP</th>
<th>Carnot eff.</th>
<th>COP/\eta_{carnot}</th>
</tr>
</thead>
<tbody>
<tr>
<td>TYLOSE</td>
<td>-25°C</td>
<td>3</td>
<td>2.0</td>
<td>5.5</td>
<td>0.4</td>
</tr>
<tr>
<td></td>
<td>-30°C</td>
<td>1.2</td>
<td>2.5</td>
<td>4.5</td>
<td>0.5</td>
</tr>
<tr>
<td></td>
<td></td>
<td>3</td>
<td>2.2</td>
<td>4.5</td>
<td>0.55</td>
</tr>
<tr>
<td></td>
<td></td>
<td>5</td>
<td>2.5</td>
<td>4.5</td>
<td>0.55</td>
</tr>
<tr>
<td></td>
<td>-37°C</td>
<td>3</td>
<td>2.0</td>
<td>4.0</td>
<td>0.45</td>
</tr>
<tr>
<td>MEAT</td>
<td>-37°C</td>
<td>3</td>
<td>1.5</td>
<td>4.0</td>
<td>0.40</td>
</tr>
<tr>
<td></td>
<td></td>
<td>5</td>
<td>1.7</td>
<td>4.0</td>
<td>0.45</td>
</tr>
</tbody>
</table>

**Table 9.5:** Results of the experiments for COP, Carnot efficiency and COP/\eta_{carnot} ratio for different conditions.

The experiment performed at -37°C, CR=3 gives, in combination with the short freezing time, the best result. The other results do not show a big difference in efficiency, thus according to that, it is recommended to use the HPF at this condition.

9.3.5 HEAT-TRANSFER COEFFICIENT

The last parameters that are determined are the external and overall heat-transfer coefficient depending of the time. These are determined with Eq. (7.4) and (7.5). The lapse of these

*Freezing processes based on the refrigerant CO₂*
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Parameters, during an experiment, can be seen in Figure 9.11. The external and overall heat-transfer coefficient are almost constant during a certain phase of the freezing process. The refrigeration capacity of the plate freezer decreases in this phase, and the temperature difference between product and plate or product and CO₂ evaporation temperature also decreases. These things compensate each other, and therefore the coefficient nearly changes. The average external and overall heat-transfer coefficient are therefore determined for all experiments during this phase. The results can be seen in Table 9.6.

![Figure 9.11: External and overall heat-transfer coefficient during an experiment.](image)

<table>
<thead>
<tr>
<th>Product</th>
<th>Tev</th>
<th>CR</th>
<th>αext [W/m²K]</th>
<th>αov [W/m²K]</th>
</tr>
</thead>
<tbody>
<tr>
<td>-25°C</td>
<td>3</td>
<td>(90 ± 10)</td>
<td>(60 ± 10)</td>
<td></td>
</tr>
<tr>
<td>TYLOSE</td>
<td>-30°C</td>
<td>1.2</td>
<td>(60 ± 10)</td>
<td>(30 ± 10)</td>
</tr>
<tr>
<td></td>
<td>3</td>
<td>(70 ± 10)</td>
<td>(60 ± 10)</td>
<td></td>
</tr>
<tr>
<td></td>
<td>5</td>
<td>(80 ± 10)</td>
<td>(60 ± 10)</td>
<td></td>
</tr>
<tr>
<td>-37°C</td>
<td>3</td>
<td>(80 ± 10)</td>
<td>(70 ± 10)</td>
<td></td>
</tr>
<tr>
<td>MEAT</td>
<td>-37°C</td>
<td>3</td>
<td>(15 ± 5)</td>
<td>(10 ± 3)</td>
</tr>
<tr>
<td></td>
<td>5</td>
<td>(30 ± 5)</td>
<td>(20 ± 5)</td>
<td></td>
</tr>
</tbody>
</table>

Table 9.6: Results of the experiments for the external and overall heat-transfer coefficient for different conditions.

The highest overall heat-transfer coefficient is reached at an evaporation temperature of -37°C and CR=3. This implies that the absolute heat-transfer towards the product in this experiment is the highest, also because the difference in temperature between package and plate or evaporation temperature is the highest.

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CONCLUSIONS AND RECOMMENDATIONS

10.1 Conclusions

10.1.1 Half-open CO₂ systems

10.1.1.1 Suction process

The suction process is in fact similar to a one-dimensional (1-D) conduction problem with CO₂ as refrigerant. Solid CO₂ is formed instantaneously after injecting liquid CO₂ on a product in a cell at atmospheric pressure. Due to heat-transfer from the product towards the solid CO₂, it sublimes, and the gaseous CO₂ is pumped out to be used again. Thermodynamically the process seems to have no problems according to the process-cycle in the H-p-diagram.

The freezing time of a fish stick (68 grams), in the suction process, corresponds with a reference at a currently used freezing tunnel in Urk at Northseafood, which is 220 seconds compared to 162 seconds. Knowing the production capacity per year (2,000 tons), the energy consumption per year (504 MWh), and the required amount of CO₂ to freeze one kilogram of fish (0.68 kg), the savings on the operational costs per year can be calculated. The total savings are 87,833 EUR.

Possible bottlenecks which can influence the process feasibility are: The formation of solid CO₂, which can affect the product quality. Obstruction of injection-nozzles, when liquid CO₂ transforms into solid CO₂, while injected on the product. Equal distribution of solid CO₂ on the product that prevents non-uniform freezing times. Creation of a CO₂ gas layer, due to sublimation, between product and solid CO₂, that affects the heat-transfer in a negative way. Pumping out impure gaseous CO₂, which detoriates the process performance.

According to the results and the examined bottlenecks, it can be concluded that the suction process has a good chance to succeed.

10.1.1.2 Heat-pipe process

The heat-pipe process has a major difference compared to the suction process. The liquid CO₂ remains liquid after injection, due to the pressure inside the cell. It evaporates because of the heat-transfer (conduction and convection) from the product towards the liquid CO₂. The formed gaseous CO₂ rises and condenses in top of the cell. There the liquid CO₂ is
10. CONCLUSIONS AND RECOMMENDATIONS

captured on a drip-tray and used again. Thermodynamically the process also seems to have no problems according to the process-cycle in the $H-p$-diagram.

In the heat-pipe process, the freezing time of a fish stick is approximately 70 seconds. This is 3 times faster than the suction process and the reference, and is mainly caused by the heat-transfer due to convection. The savings on the operational costs per year can be determined using the yearly production capacity per year (2,000 tons), the energy consumption per year (400 MWh) and the required amount of CO$_2$ to freeze one kilogram of fish (0.64 kg), and results in 87,040 EUR.

Bottlenecks that could cause problems for the process performance are: The pressure inside the cell is too high, which causes the walls to break, and decreases the product quality. Equal distribution of liquid CO$_2$ that prevents non-uniform freezing times.

Compared with the suction process, the heat-pipe process freezes faster, but requires a more complex installation (high pressure inside cell). The savings on the operational costs are almost the same as in the suction process, and therefore it is preferred to develop and examine the suction process more thoroughly instead of the heat-pipe process.

10.1.2 HORIZONTAL PLATE FREEZER

10.1.2.1 Simulations

The horizontal plate freezer, used in the pilot plant, is modeled in MATLAB$^\text{TM}$. In this model four types of flow resistors, representing the pressure drop caused by the orifices, two-phase flow inside the channels, and the header at the injection and suction side, are present in the model (see Figure 8.1). With this model simulations are performed under different conditions.

The simulations showed that the flow resistor, representing the pressure drop generated by the orifice, is dominant. This is desired, otherwise the pressure drop caused by two-phase flow, injection or suction header 'determines' the amount of CO$_2$ that will flow through each channel. The dominance of the orifice implies each channel has an equal pressure drop, and therefore the same amount of liquid CO$_2$ enters each channel.

Now it is possible to determine the heat-transfer coefficient under different conditions. According to the results, the single phase flow heat-transfer coefficient increases at higher circulation rates, which is caused by a higher flow velocity that improves convection. Compared with the two-phase flow coefficient the single phase flow coefficient is much lower. At higher circulation rates this effect becomes bigger. This can be understood looking at the flow pattern of the two-phase flow, which changes from stratified flow into plug flow. Plug flow has a higher heat-transfer coefficient than stratified flow [14].

10.1.2.2 Experimental results

In the reference measurement at Polskamp, performed at a plate freezer with refrigerant R-22 at evaporation temperature of -48°C, data loggers are placed in meat packages, that enter the plate freezer, which monitor the temperature of the meat. After examining the data, the average freezing time of the meat is $(60 \pm 10)$ minutes.
10. CONCLUSIONS AND RECOMMENDATIONS

Different experiments are performed at the HPF pilot plant, with refrigerant CO₂, to compare with the simulations and the reference measurement. These experiments showed that the pressure drop inside the header was negligible (approximately 4-10 mbar), which is almost similar as in the simulations. From this it can be concluded that the liquid CO₂ distribution over the channels is uniform.

The refrigeration capacity of the plate freezer was monitored, and results showed the dependency of mass present on the HPF. In experiments with meat, there was 20% less mass present, than in the experiments with tylose. This resulted in a refrigeration capacity that was also 20% lower.

The freezing time of the product differed, dependent on the evaporation temperature of the CO₂, but independent of the circulation rate. The lowest average freezing time was reached at an evaporation temperature of -37°C and a circulation rate of 3. This was (65 ± 15) minutes, and is in the same range as the reference measurement at Polskamp.

However there was one channel, independent of the conditions, which had a significant longer freezing time. This was, after research, not caused by non-uniform plate temperatures or liquid distribution of CO₂. X-ray pictures of the orifice were taken, to see if there was an error, or pollution at this spot. On these pictures there was nothing special to see. According to that the exact reason is still unknown. A possibility can be that this channel had a smaller manufactured orifice diameter than others, or down the channel there was some pollution that was not photographed.

The energy efficiency was calculated during each experiment. According to the results, it showed that the HPF performed the best at an evaporation temperature of -37°C and a circulation rate of 3. The other conditions nearly differed.

Concluding it can be said that the HPF pilot plant with refrigerant CO₂ performed well, compared to the reference and simulations. The optimal condition seemed to be at an evaporation temperature of -37°C and a circulation rate of 3.

10.2 Recommendations

10.2.1 HALF-OPEN CO₂ SYSTEMS

In this study, a 1-D conduction problem (including convection at the heat-pipe process) is used to calculate the freezing time of a fish stick. A better estimation can be made using the program BERTIX™, which is especially developed for this. This probably results in a realistic freezing time, thus CO₂ usage.

The investigation of the costs for placing a NH₃-CO₂ installation instead of a "normal" NH₃ installation is also recommended. In the calculations on the operational costs this was not included. To compare both processes in a fair way with the reference, concerning the costs, this has to be done.
10. CONCLUSIONS AND RECOMMENDATIONS

Both processes are examined on efficiency and performance inside the cell. Therefore it is recommended to investigate the entire system for each process, in other words including the NH₃-CO₂ installation, to get a better view at the total efficiency and performance of the processes.

When these recommendations are executed, both processes can be better compared mutually and with the reference, which results in an all-embracing decision which process behaves the best.

10.2.2 HORIZONTAL PLATE FREEZER

Looking at the results of the simulations and the experiments/reference it is recommended to investigate the influence of changing the orifice diameter before the channels. This to optimize the distribution of liquid CO₂ over the channels, and hereby create a uniform plate temperature which, on its way, positively effects the freezing of the products (uniform, and the average freezing time decreases due to the elimination of "weak spots" on the plate.

Further, it is recommended to perform tests with higher circulation rates. The increase in circulation rate, thus mass flow, was in this thesis not possible, because the used CO₂ pump could not achieve a higher mass flow. It is recommended to increase the mass flow because after calculating the two-phase flow pattern inside the channels. It appeared that the two-phase flow pattern changes from stratified into plug flow, when the circulation rate is increased from 3 to 5, which contributes to a better heat-transfer. This two-phase flow pattern change will probably occur again when increasing the circulation rate further, thus improvement of the heat-transfer.

Another element that is useful to examine, is the influence of the height of the plate freezer. In this project there is a HPF used with 4 plates, so nothing can be said about the distribution of the liquid CO₂ over the height in these experiments. In reality the plate freezer consists of 29 plates. This induces an extra pressure drop element, namely a gravitational. Therefore the distribution of liquid CO₂ per plate can differ, and through that the freezing time of the product.

A last issue that is recommended to examine, is the influence of lubricant on the heat-transfer. Because a new way of decreasing the evaporation temperature is with the use of a CO₂ compressor instead of the two NH₃ compressors in the cooling installation, as now in Figure 7.3. This CO₂ compressor takes over the LT compressor in the installation. Because the liquid CO₂ is inside a compressor, lubricant will dissolve in CO₂. This changes the physical properties of CO₂, and probably the heat-transfer towards the product.
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