Energy Analysis of the Drying Hood in Paper Machine 1 at Stora Enso, Nymölla Mill

by

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Preface

This report is the result of my Master Thesis that has been carried out from January to June 2016 at Stora Enso, Nymölla, Sweden. It is also the end of my education to Master of Science in Engineering, chemical, with specialization process design, at Lund's university.

First of all I would like to express my gratitude to my supervisor Professor Stig Stenström at Lund University (department of chemical engineering) for your great advices and your support during the Master Thesis. I would also like to thank my supervisor at the company, Per-Arne Olsson, for giving me the opportunity to perform my Master Thesis at Stora Enso and for sharing your knowledge. It has both been an instructively and an exciting work. Thanks to Malin Öberg and Linnea Eriksson for taking your time for discussions as a help to solving my problems.

Last but not least, a big gratitude to everyone at the mill who has been helping me during the Master Thesis and contributes to knowledge or a helping hand for the experimental measurements.

Emmy Lam

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Sammanfattning

Den 1 juni 2014 trädde lagen om energikartläggning i stora företag (2014:266) i kraft från energimyndigheten. Lagen innebär att alla företag har en skyldighet att göra en energikartläggning för deras årliga energianvändning. Detta för att få ökad kunskap om vad energin används till i företaget. Med stora företag menas att man anställer minst 250 personer eller har en årsomsättning över 50 miljoner EUR eller en balansomslutning som överstiger 43 miljoner EUR per år.

Detta examensarbete har haft i syfte att göra en detaljerad energikartläggning för torkpartiet av pappersmaskin 1 på Stora Enso, Nymölla bruk. Bruket har 520 anställda och omfattas därmed av den nya lagen om energikartläggning i stora företag. Med den detaljerade energikartläggningen av torkpartiet tas förslag av kostnadseffektiva åtgärder fram för att spara energi eller för att öka energieffektiviseringen.

Pappersmaskinen på Nymölla bruk består utav 8 olika processteg som är; inloppslåda, viraparti, pressparti, förtorkning, limning, eftertorkning, kalandrering och upprullning. Torkpartiet (både förtorken och eftertorken) studeras i detalj, där in- och utströmmar definieras. De definierade in och utströmmarna i för- och eftertorken utgörs utav:

- Papper
- Vatten
- Ånga
- Luft

Med dessa in- och utströmmar beräknas mass- och energibalanser för både förtorken och eftertorken och illustreras till sist i ett Sankeydiagram.

Den totala massan in och ut från förtorken är 126 kg/s och motsvarar ett energiflöde in och ut från förtorken på 40 MW. Vilket ger en årlig energianvändning på 310 GWh i förtorken. För eftertorken erhölls den totala massan in och ut till 76 kg/s med ett totalt energiflöde in och ut på 14 MW. Detta är räknat på en produktionstid på 7500 h/år. Den specifika ångförbrukningen i förtorken beräknas till $\Delta H_{vap} = 3\ 701.4\ kJ/kg$ och för eftertorken beräknas den till $\Delta H_{vap} = 3\ 116.4\ kJ/kg$. Detta värde jämförs med det teoretiska värdet för förångningsentalpin vid 70 °C som är $\Delta H_{vap}(70^{\circ}C) = 2333\ kJ/kg$. Vilket visar på det finns energibesparingsmöjligheter.

Åtgärdsförslagen för att bli energieffektivare är att installera en extra tilluftsfläkt i förtorken och nollnivåreglering i eftertorken. Detta reducerar ner läckluftsmängden. Kostnaden för tilluftsfläkten är 600 000 SEK. Detta ger en årlig besparing på 2 miljoner SEK samtidigt som det tillkommer en årlig driftskostnad på 40 000 SEK. Detta skulle ge en återbetalningstid på 4 månader för tilluftsfläkten. Installation av nollnivåreglering kostar 80 000 SEK och skulle ge en årlig besparing på 1,8 miljoner SEK. Återbetalningstiden för denna installation är 0,5 månader.

Båda installationerna anses som en lönsam investering då återbetalningstiden är mindre än 1 år.

Abstract

The new law about energy mapping in large companies (2014:266) from energy agency was valid from 1 June 2014. The law requires that every large company has a responsibility to make an energy mapping for their yearly energy usage. This is to get an increased knowledge of what the energy is used for in the company. Definition of a large company is if the company employs least 250 persons and have yearly revenue over 50 million EUR or a balance sheet total over 43 million per year.

This master thesis has an aim to do a detailed energy napping for the drying section of paper machine 1 at Stora Enso, Nymölla mill. The mill employs 520 persons and is enforced to follow the new regulation about energy mapping in large companies. The cost-effective suggestions for energy savings or increasing the energy efficiency are presented in the detailed energy mapping for the drying section.

The paper machine at Nymölla mill contains 8 different stages; head box section, forming section, pressing section, pre-drying section, sizing section, after-drying section, calendaring and wind up. The drying section (both pre-drying- and after-drying section) is studied in detail, where the inlet- and outlet streams are defined. The defined inlet- and outlet streams in drying section are:

- Paper
- Water
- Steam
- Air

With these inlet- and outlet streams the mass- and energy balances are calculated in both the pre-drying section and the after-drying section. They are illustrated in a Sankey diagram.

The total mass in- and out from the pre-dryer is 126 kg/s and correspond to an energy flow of 40 MW. This gives a yearly energy usage of 310 GWh in the pre-dryer. For the after-dryer the total mass in- and out are 76 kg/s with a total energy flow of 14 MW. This is calculated with a production time of 7500 h/year. The specific heat consumption for the pre-dryer is calculated to $\Delta H_{vap} = 3701.4\ kJ/kg$ and to $\Delta H_{vap} = 3\ 116.4\ kJ/kg$ for the after-dryer. Comparing these values with the theoretical value for the latent heat of evaporation at 70 °C which is $\Delta H_{vap}(70^{\circ}C) = 2333\ kJ/kg$ indicates there is potential of energy savings.

The suggestions for increasing energy efficiency are to install an extra supply air fan in the pre-dryer and a zero level control in the after-dryer. This will reduce the leakage air. The cost for the supply air fan is 600 000 SEK. This gives a yearly saving of 2 million SEK and resides an operating cost of 40 000 SEK. The pay-back time is 4 months for the supply air. Installation for the zero level control is 80 000 SEK and would give a yearly saving of 1.8 million SEK. The pay-back time is 0.5 months.
Both installations are seemed as a profitable investment since the pay-back time is less than a year.

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1. Introduction

1.1 The company

Nymölla Mill is a part of the Swedish-Finish concern Stora Enso and had its opening during 1962 [1] . The company is a leading provider for a sustainable wood industry, which means that the company refines the woods raw material in a sustainable way for the environment. Their aim is to replace the fossil based materials with renewable materials [2]. In Nymölla lies the production of paper for office use and for print media, for example the well-known *MultiCopy* is one of their products, but beside this also more special kind of papers for photography, calendars and envelopes are produced [3, 4]. The amount employees at Nymölla Mill are around 520. The production process at Nymölla Mill is a TCF (totally chlorine free) process, which means that there's no use of chlorine in the bleaching of pulp instead hydrogen peroxide, oxygen, sodium hydroxide and per-acetic acid is used for the bleaching [4]. Stora Enso has an amount of 6 mills in Sweden where each mill produces different kind of products from the wood, everything from packaging to biomaterials for hygiene [3].

For the paper production in Nymölla the pulp is produced in a pulpmill beside the paper factory and therefore the mill is an integrated pulp- and paper mill [5]. With this in advantage, the produced pulp could be used directly in the paper factory and avoid emissions caused by the transport of the pulp to the paper factory. The delivered wood is transported from south Sweden where it comes from beech or birch, but also some wood is imported, like eucalyptus or birch. The pulp also contains recycled paper that would make the paper more solid and also to utilize the raw materials [1].

1.2 Aim of the master thesis

The production of paper is an energy intensive process and with rising energy prices the energy question about reducing the energy usage becomes more serious and there is a large demand of increasing energy efficiencies. The purpose of this master thesis is to do a detailed energy mapping for one of the process at Nymölla Mill and put up a Sankey diagram for the energy usage. This will give knowledge of where energy efficiencies can be made and then come up with suggestions to save more energy. With this also comes an economic analysis for the suggested solutions for reducing the use of energy.

The measurement uncertainty for the mill is also discussed and what conclusion could be decided from this.

Nymölla mill has an aim to do energy savings and find energy efficiencies according to following cases:

- Save 5 MW electricity as an average over the year compared to the case in 2008
- No use of fossil fuels
- To yearly decrease the specific energy use (kWh/ton paper) by 7.5 % compared to the case in 2010 [4]

1.3 Limitation for the master thesis

The company contains lots of energy consuming processes that could be surveyed, and the time for the master thesis is 20 weeks. Because of the time limit it is here chosen to put focus on the drying section in paper machine 1 and make a detailed energy survey for this process. The functions of paper machine 1 are also briefly studied from when the pulp comes to the head box section till the paper is rolled on tambours. The hall where the paper machine stands in is excluded from this study.

1.4 Outline

This report starts with a description for the process of paper making together with how this process is controlled. In this chapter the different steps in the paper process are briefly described. The general demands and the new law from the energy agency are presented in chapter 3 together with a theoretical description of the thermodynamics first law. What has been found in the literature, and what has been done earlier is also presented here. The equations for the mass- and energy balances and the mathematical formulas used in this work are also included. How an energy survey should be made is presented in chapter 4 together with the experimental set up to find the parameters needed for the mass- and energy balances. In chapter 5 the results of the experimental measurements that have been obtained are presented with flow sheets for the measured points. Calculations for the mass- and energy balances are also done here with values from the experimental measurements and are finally presented in a Sankey diagram. The report is then round off with chapter 6 with a discussion for the measurements uncertainty and what can be concluded. To the last, in chapter 7 the futures work are presented.

2. The process

The process of making paper, from pulp to fine paper contains of 8 different stages; head box section, forming section, pressing section, pre-drying section, sizing section, after-drying section, calendaring and wind up. The different stages in the paper making process are briefly presented in this chapter together with how the process is controlled. Figure 1 *below gives an overview of the paper making process.*

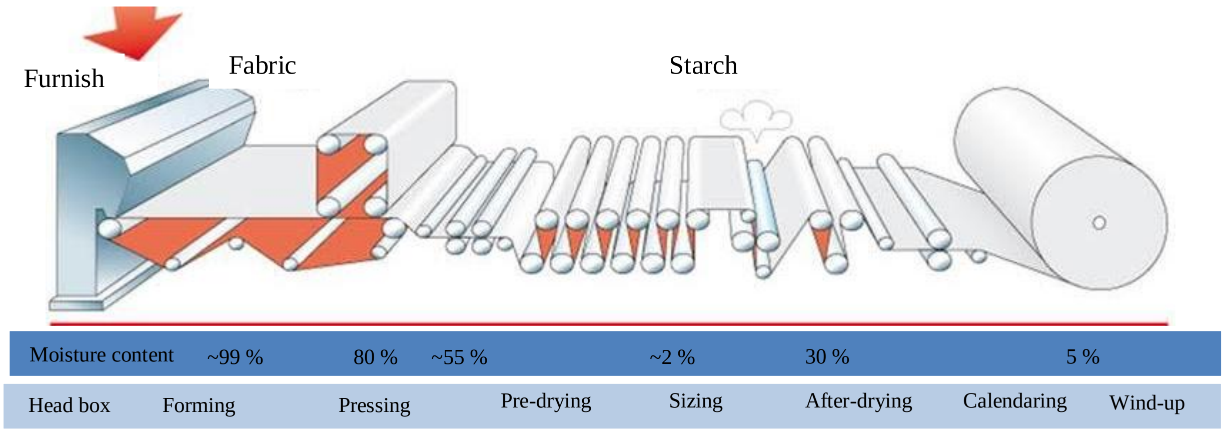

Figure 1 - Overview for the process of papermaking [6]

2.1 Head box section

The first stage in this process is the head box that contains the mixture of chemicals and the pulp of soft- and hardwoods that is diluted with water. The task that the box has is to spread this mixture out to the fine forming fabric where the most of the water from the mixture is rinsed of [7]. The moisture content in this step is 99 % [6]. When the pulp is at the forming fabric the process is called forming section.

There are different kinds of boxes that the mixture of pulp flows in. Dependent on the quality of the paper the mixture is spread out with different velocities. The velocity can be 100 m/min or 400 m/min and is adjusted on the forming fabrics velocity so there is an even turbulence in the box during the flow out of mixture [7]. The box used at Nymölla Mill is a lip box where the opening of the box could be regulated dependent on the decided basis weight of the paper [8].

2.2 Forming section

The pulp mixture is now on the forming fabric and the most of the water (free water) is dripping of with the gravity. After the box there is a shaker that will help the pulp mixture to draw by dewatering. This water is then reused as white water for the head box [8]. Among the paper web in the beginning at the sides, there is water that is sprayed out to form the paper sheet so that it wouldn't spread out from the fabric.

The shakers task is also to give an even distribution of the pulp for the paper sheet, this will give a more homogeneous surface of the paper or else there will be dark spots. There are also splines that will prevent this dark spots because of uneven distribution [9]. Later the forming fabric meets an over fabric that will help to press the paper web free from water and give a symmetrically paper [9]. For the dewatering there are also 6 vacuum pumps that will contribute to higher dryness in the paper web by absorbing the water [8]. The aim of the forming part is to give enough dryness for the paper so it can handle the press part where larger forces are performed at the sensitive web [7]. Before the paper web goes to next part of the process, it passes a press clothed with a fabric with high absorption capacity. This will give an effective dewatering of the paper web [7, 8].

Because of the high absorption capacity this could later give a decrease of the under pressure from the vacuum boxes in the pressing section since less water needs to be dewatered. This will contribute to a lower use of electricity and less wear at the forming fabric [7]. This press will give the paper web an increased dryness of 3-4 %-units before it comes to the pressing section [7] and the moisture content is 80 % when the paper web comes to the pressing section [6].

2.3 Pressing section

The transferring from the forming section to the pressing section is done with a pick-up roll that takes up the paper web with vacuum to a new sheet in to the pressing section. This phenomenon is seen in Figure 2 below. At the same time the paper web is picked up the sides are also cutted [10].

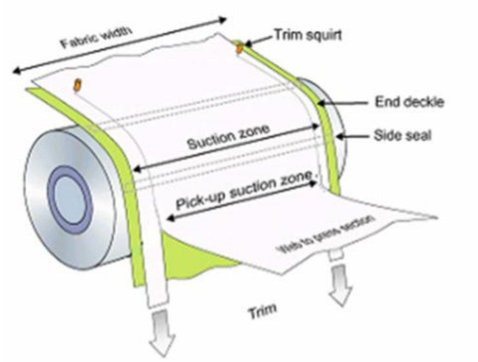

Figure 2 - The pick-up that transfer the paper web to the press section [10]

The paper web is then transferred through 3 big clothed pressing rolls to increase the dryness of the paper web and the first free draw occurs during this part. The clothing of the rolls absorbs the water from the web and the water in the cloths is then absorbed with vacuum [7, 10] .

When the clothing is absorbing water also other chemicals than water is absorbed and this will cause stowing for the clothing and give bad absorption capacity. The cloth is therefore cleaned with high pressure water in the process [7]. Before the paper web goes to next part of the process it passes a shoe press with longer contact area between the rolls and the paper web than the previous rolls. This enlarged contact area will increase the dewatering and less water needs to be evaporated in the pre-drying section [9].

2.4 Pre-drying section

When the paper web comes out from the pressing section the dryness is about 45 % [6] and the dewatering of the paper web takes place on multi-cylinder section alternated with steam and vacuum. The vacuum cylinders are placed in the lower row and the steam cylinders are place at the upper row in the beginning. This is seen in Figure 3 [11]. It is the steam heated cylinders that gives the paper web a higher dryness while the vacuum cylinders is just there to keep the paper web on place because of the high velocity of the sheet [12]. Beside this, there is also an air-system in the hood that helps the paper web to increase it dryness by taking in heated air to the hood, that aims to take up the evaporated water from the paper web [13] .

The design of an upper row of steam filled cylinders and vacuum cylinders at the lower row allow the paper web to be drawn freely. This occurs when the paper web alternately pass the steam cylinders and vacuum cylinders. In the free draw the drying rate is higher because the paper web is allowed to be evaporated from both sides with the ingoing drying air.

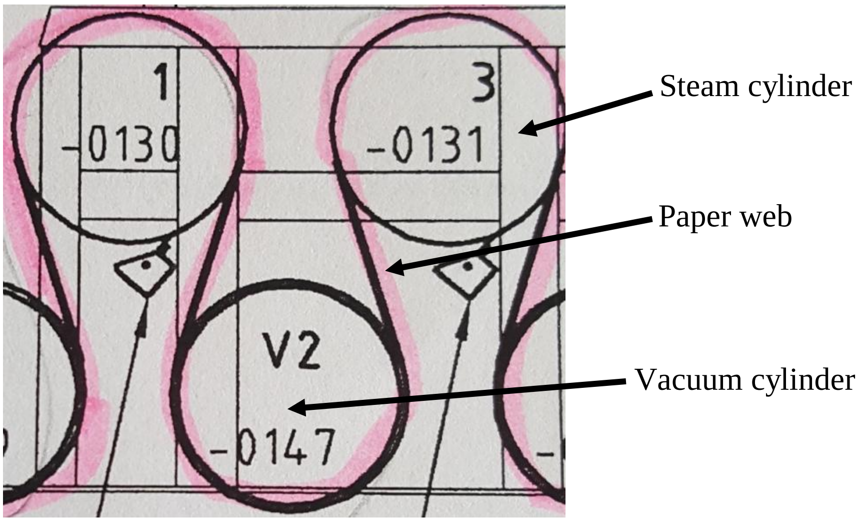

Figure 3 - Alternated steam- and vacuum cylinders in the drying section [9]

2.4.1 Steam-condensate

The paper web passes cylinders with different steam groups (different steam pressures e.g different temperatures) where the steam condenses inside the cylinders. This condensing steam will transfer heat to the paper web to evaporate the water. The dryness in the paper will increase and the evaporated water is then ventilated away with supply air [14].

Before the heat from the condensing steam reaches the paper it is transferred through a condensate layer inside the cylinder, the wall of the cylinder and the air film between the paper and the cylinder [15, 14]. To set up the heat transferring through every layer, the total heat flux to the paper web can be decided, see Appendix A. The Figure 4 below shows how the heat flux is transferred through the different layers before it comes to the paper web.

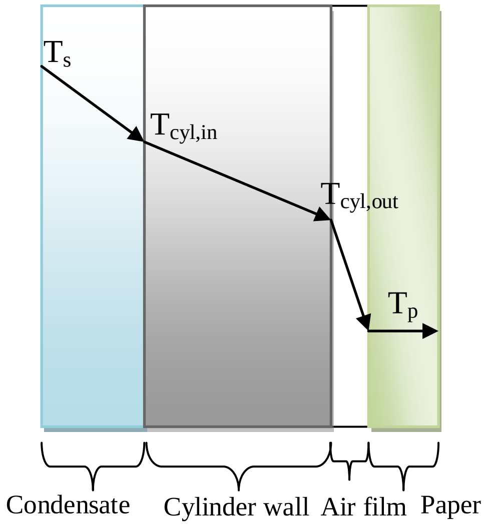

Figure 4 - The heat flux from the steam transferred to the paper web

The steam that doesn't condense comes out from the cylinder on the other side and is around 10-15 %, so called the blow-through steam. The blow-through steam brings out the condensate inside the cylinder and has a lower pressure than the ingoing fresh steam [15].

The mixture of blow-through steam and condensate is pumped to a flash drum. The flash drum has a lower pressure than the blow-through steam and will separate the secondary steam from the condensate. This will create flash steam. The flash steam is sent to a cylinder group with a lower steam pressure together with a little fresh steam and the condensate is sent to a condensate tank. This phenomenon will cause a cascade system and the blow-through steam out from the last cylinder group is too low to fit a cylinder group. The blow-through steam from the last group is therefore sent to a surface condenser. All the condensate from all the cylinder groups is then sent back to the boiler house to be heated up to steam [15, 14]. An illustration of the cascade system is shown in Figure 5.

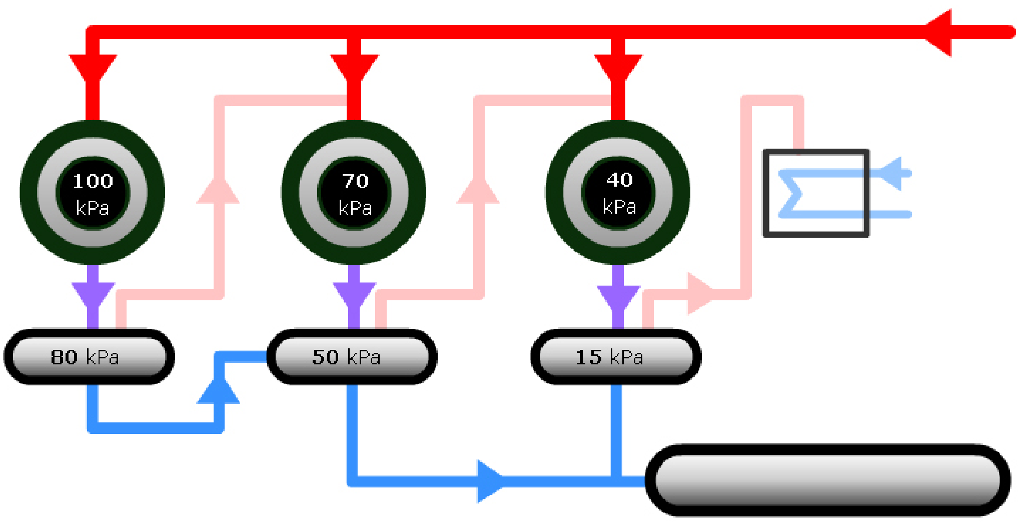

Figure 5-Illustration of the cascade system [14], pink=flash steam, purple=blow through steam, red=fresh steam, blue=condensate.

The pre-drying section at Nymölla Mill contains 30 steam filled cylinders and they are divided into 6 different steam groups with different numbers of cylinders. Each steam groups then works with their steam pressure dependent on which kind of paper that is produced. In the beginning the cylinders are alternated with steam and vacuum cylinders; later on in the drying process when the paper is dry enough only steam cylinders are used in the upper and lower row [16].

The amount of steam cylinders in each steam group for the pre drying section are presented in Table 1 below:

Table 1 – The steam groups in the pre-dryer section at Nymölla Mill [16]

Steam group	Amount cylinders
01	1
1A	2
1B	1
2	3
3A	13
3B	10

2.4.2 Air-system

The air condition in the drying hood is important for the paper drying process, since taking out the humid air enable the evaporation of water in the paper web [15]. The exhaust air from the drying hood is used as heat recovering since almost all of the added energy comes out with the exhaust air [17]. This recovered heat contents large amounts of energy and this energy are reused for preheating the ingoing fresh air from the machine room, heating water for ventilation and processing stuff which would give savings for steam usage [18]. After the fresh air is preheated with the exhaust air it is heated with steam in an air-preheater till it reach the required temperature of the supply air. Then it is blown to the paper web and gives an increase of the dryness [13].

The preheated supply air in to the drying hood that is added for command is 80 % of the total outgoing air from the drying hood. The remaining 20 % of the outgoing air make up the leakage air. This leakage air is not preheated before it goes in to the drying hood [13].

2.5 Sizing

When the paper web comes out from the pre-drying section it passes the sizing step with a dryness of 98 % [6]. The sizing aims to give the paper strength by applying starch to the surface of the paper web. This starch will give the paper better surface strength, reduced dusted and better rigidity [19]. It is important that the paper have right dryness before the sizing takes place or else the sizing would give the paper an uneven surface. Therefore the sizing have to be placed in a sufficiently distance from the pre-drying section to make sure the paper web is dry [7].

The sizing contains two rubber clothed rolls that is cooled with water inside to give the paper web correct temperature when applying the hot starch. On these two rubber-clothed rolls there is starch and when the paper web passes through these rolls the starch cover the surface of the paper [19]. See Figure 6. Because of the starch applying in the sizer, the paper is rewet again and therefore needs to be dried again. This is done in the after-drying section [15].

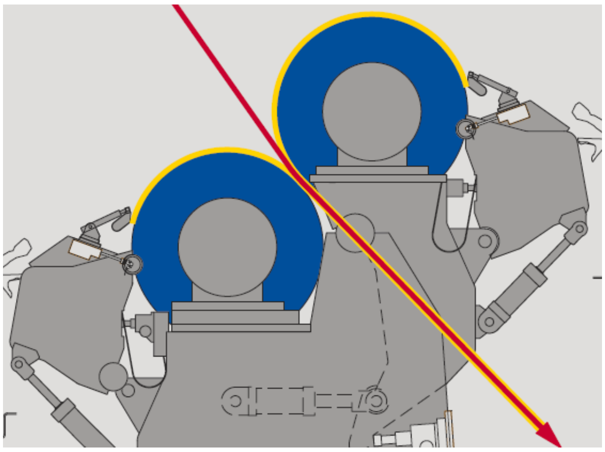

Figure 6 - Coating of the paper with starch [19]

2.6 After-drying section

The water at the surface of the paper web contains most free water and therefore the drying is relative easy in this section. The absorbed water from the sizing should be evaporated for increased dryness. This is done in the same way as the pre-drying section but with fewer cylinders since less water needs to be evaporated. Also the steam pressure is lower here and is not so energy intensive as in the pre-drying section [7].

In the after-drying section at Nymölla Mill, there are 9 steam filled cylinders that are divided into 3 different steam groups with 3 cylinders in each group which are shown in Table 2 [16]. The steam groups in the after-drying section also make up a small cascade system like the pre-drying section.

Table 2-The steam groups in the after-drying section

Steam group	Amount cylinders
5	3
6A	3
6B	3

2.7 Calendaring

When the paper is dry enough it passes a surface processing step for increased shine and better sleekness for the surfaces of the paper. The paper web passes two pinches alternated with a soft clothed roll made of cast iron and a hot thermos-cylinder with hot water inside the roll. It is the hot roll that gives the paper its shine. The soft clothed roll is sensitive against heat that comes from the hot thermo-cylinder and therefore needs to be cooled [20]. But before the paper web passes these two pinches it is humidified with steam that condense at the paper web. The heat from the condensed water steam is then transported to the paper and also gives the paper web increased humidity before passing the rolls. This humidifying of the paper would give the paper less different sided paper, better shine and sleekness [7].

2.8 Wind-up

The last step in the paper making process is the rolling of the paper web on large tambours for later converting to correct sizes for paper sheets. Each roll of paper gives 5 million sheets A4 and it takes around 1 h to finish a paper roll [21].

2.9 Controlling of the process

The process is regulated with computers in the control hall where parameter settings are set from the operators. In the process there are indicators that measure the temperature, velocity, moisture content, steam pressures and other parameters. All the instantaneous parameters are observed in the software “Metso DNA” and it is also here the settings for the process are controlled. Also in the control hall there are screens on parts of the process that is used for observing defects on the paper, like paper break and holes. Some parts of the process are controlled manually from the operators in the fabric where the operators can go out and change the flow path by closing a port or by changing the opening grade of the port to change the velocity of the flow.

3. Theoretical frameworks

This chapter presents the theory that are found for the master thesis, like journal articles of what has been done before and literature that are used. The chapter includes the general demands from energy agency, the thermodynamics first law and formulas that are used for the energy survey in the drying section of the paper machine.

3.1 Energy agency

Due to the new law (2014:266) about energy mapping in large companies from the energy agency that was valid from 1 June 2014. Every large company should make an energy survey for their yearly use of energy to get knowledge about the use of energy in the company. With the increasing knowledge of the energy usage efficiencies can be made [22]. Definition of a large company is if the company employ least 250 persons and have a yearly revenue over 50 million EUR or a balance sheet total that is over 43 million EUR each year [23].

The energy mapping should be updated every 4th year [23] and should be done within a certified framework. A certified framework systemically describes how the energy usage can be provided in a company. A standard certified framework is ISO 50 0001 [24]. The energy mapping should contain a describing yearly overview of the company's total use of energy and a detailed mapping for the processes that consume energy. Suggestions of how to use the energy in a more efficient way for both decreasing the costs and energy usage should also be included in the energy mapping [23]. The aim for the suggestions is to produce more products per energy unit [22] and the energy added to the company should be in MWh and in SEK per kind of energy used per year [25].

3.2 The law of conservation of energy

According to the first law of thermodynamic, the total energy in an isolated system is constant. The energy can't disappear or be produced, it is only transformed to another form of energy but the total amount of energy in the system still remains [26]. This means that what goes in to a system comes out again and mass- and energy balances can be set up. The general formula for the balances is set up as following [27] :

$$Ackumulated = Inlet - Outlet + Production - Consumption \tag{3.1}$$

For the paper machine, there is no reaction and only mass- and energy in different kinds is added. The production and consumption terms in equation 3.1 are therefore not included in the paper machine. When the accumulation term is zero all the inlet streams are equal to the outlet streams. A boundary line describes what makes up what's inside and outside the system. The streams that cross the boundary line for the system decide which mass- and energy balances that should be set up and from here it is known what kind of data that is needed to be collected [27].

3.3 Literature search

Journal articles

Database: Engineering village

Keywords: Energy saving, paper machine, energy analysis, energy survey

3.3.1 The paper industry

The papermaking process is an energy intense process and because of increasing threats against the environment it is crucial to improve the usage of energy in these industries [28]. It is known that the pulp- and paper industry are the fourth largest industry in the world because of its high usage of electricity and heat. By doing an energy survey of the pulp and paper industry, where the energy flows are analyzed, potential of energy savings can be identified [29]. The largest costs in a paper mill are the raw material and the energy. It is estimated that 20-30 % of the total operation costs is constituted of the energy usage in form of steam and electricity [30]. Also with the rising prices and environmental issues many paper mills are motivated to reduce their energy usage by finding energy efficient solutions. Except that the paper making process is an energy intense process it is also a huge user of fossil fuels which contribute to the emissions of greenhouse gases [29].

The general principles for a pulp- and paper production consist of five steps as following [29]:

- Raw material production
- Pulping
- Bleaching
- Chemical recovery
- Papermaking

By improving the energy usage of the paper industries it will contribute to minimizing the output of greenhouse gases, e.g., the fossil fuel. The analyzation of the energy flow consumption includes the energy supply, energy distribution, energy conversion and energy-end use in the pulp- and paper industry [31]. As a result when the energy flow for a pulp and paper industry in Taiwan was studied, there were large potentials for energy conservation, most in the energy-end use where energy could be used in other manufacturing [31].

3.3.2 Drying theory

The primary task in the paper making process is dewatering from the paper web. The different dewatering step consists of forming section, pressing section and drying section. The drying section that is the most energy intensive step [32]. The drying of water in the drying section is done by evaporation and the high energy usage in this step is because of the waters high vaporization heat [13].

When the paper web is transferred from the pressing section to the drying section, the moisture content in the paper web is in general around 35-55 %. This water is evaporated in the drying section till the paper web reach a general final moisture content of 6-9 %. Even if the drying section takes away the least portion of water in absolute terms, it is still the most energy consuming step and the step with highest costs in the paper making process [28]. The drying section removes 1 % of the original water and the energy required for this is more than

65 % of the total energy required for the whole paper making process [32]. The cost for the drying section is 40 % of the total capital investment [30]. Therefore an optimization of the drying section in the paper machine could decrease the energy usage of the paper machine significantly since the drying section is the largest energy user. There is a high priority of finding efficiencies for the drying section [30].

The process parameters are the one that affects the energy usage in the paper machine. With different settings on the process parameters for the same drying section, the energy usage for the drying section is varied. There are therefore optimal settings for the energy usage in the drying section [30].

The driving force for the water evaporation is the difference of the partial pressure for the steam at the paper web and the partial pressure for the water in the surrounding air. It is desirable to have a high driving force to evaporate as much water as possible. The supply air in to the hood shouldn't be too moisture because this will give a small driving force for the evaporation and decrease the airs ability of taking up water. As a result, more air is needed. The dryer air the air is, with higher temperature gives higher drying rate for the paper [13]. It is an advantage to use low air flows in to the drying hood because less energy is needed to heat this air. Low air flows will also decrease the drying capacity since more humid air is stack up in the drying hood. If this humid air is not ventilated away it will be saturated and the drying stops to occur [13]. The high humid air will give a higher dew point and the advantage is that it contains more energy. This energy is used for heat recovering to preheat the ingoing fresh air from the halls. The warmer the preheated fresh air is the less steam is needed to heat the supply air and will give energy savings, both for the fans and the steam usage [18]. Therefore there should be a balance of the air flows that will give a high dew point at the same time as the drying of paper is not reduced.

The paper that should be dried is seemed like a solid material with an overall wet surface. The drying process of paper with air is treated in the same way as the system for air-water at evaporating. Since the heated air evaporates the water from the paper, the condition of the air is going to change. The airs condition changes along the adiabatic line, which is the same as the wet-bulb temperature in the Mollier chart for humid air. The air flow and water content in an air stream is constant during cooling or heating. The changing parameters in the air stream are the relative humidity and the enthalpy. When heating the air stream the enthalpy is increased and when cooling the enthalpy is decreased. If the air is cool down below the dew point the water in the stream condense and therefore this will decrease the humidity of the air [33].

3.3.3 Different methods of paper drying

The whole paper drying process is accomplished with a steam-and condensate system, air system and heat recovery system [30]. Today there are different methods for drying paper where all methods have the same basic task to transfer heat to the paper web. The different methods that are used for the drying section are cylinder drying, infrared drying and convection drying. [28]. The multi-cylinder drying is today the most used method for paper drying where it's dominating in the industries [15] and stands for a part of 85-90 % [28].

The multi-cylinders that are used in the paper machine consist of a number of hollow cast iron cylinders where the paper web is passing on the surface. Inside the cylinders steams is going through and evaporate water from the paper web by condensing. Except the steam that evaporates the water there is an air-system that heat the supply air that will give the paper web increased dryness and ventilate away the humid air [28]. The whole drying section takes place in a drying hood. The drying hood is designed to ventilate away the humid air. This allows the drying of the paper web to continue and also to recover as much heat as possible from the humid air [15].

3.3.4 Mathematical formulas

The elements that are included in the drying section are fiber, steam, water and air [32]. Mass- and energy balances are set up for these elements with respect to the thermodynamics first law where the energy is conserved in the drying section.

The mass balances for the elements are [32]:

$$\sum G_s = \sum G_c + \sum G_{bt} \tag{3.2}$$

$$\sum G_{sup} + \sum G_{leak} = \sum G_{exh} \tag{3.3}$$

$$\sum G_{sup} \cdot X_{sup} + \sum G_{leak} \cdot X_{leak} + \sum m_{evap} = \sum G_{exh} \cdot X_{exh} \tag{3.4}$$

$$m_{fib,in} = m_{fib,out} \tag{3.5}$$

And the energy balances for the same elements are set up in to: [28, 32]

$$Q_s = G_s \cdot H_s \tag{3.6}$$

$$Q_{sup} = G_{sup} \cdot H_{sup} \tag{3.7}$$

$$Q_{exh} = G_{exh} \cdot H_{exh} \tag{3.8}$$

$$Q_{leak} = G_{leak} \cdot H_{leak} \tag{3.9}$$

$$Q_{p,in} = \left(m_{fib\ in} \cdot C_{P,fib} + m_{H_2O(l),in} \cdot C_{P,H_2O(l)}\right) \cdot T_{p,in} \tag{3.10}$$

$$Q_{p,out} = \left(m_{fib,out} \cdot C_{P,fib} + m_{H_2O(l),out} \cdot C_{P,H_2O(l)}\right) \cdot T_{p,out} \tag{3.11}$$

$$Q_c = G_c \cdot C_{p,H_2O(l)} \cdot T_c \tag{3.12}$$

The energy content for the humid air is dependent on the humidity and the enthalpy is defined at the 0-point as follows [34]:

$H_{dry\ air,0\ {}^0C} = 0\ kJ/kg\ dry\ air$

$h_{H_2O(l),0^0C} = 0\ kJ/kg\ H_2O$

To calculate the enthalpy for the water-air mixing with a moisture content X at a temperature T ^{0}C it is done in following three steps:

- Heating of dry air from 0 ^{0}C to T ^{0}C→ $C_{P_{air}}(T-0) = C_{P_{air}} \cdot T$
- Evaporating of water at 0 ^{0}C → $\Delta H_{vap} \cdot X$
- Heating of steam from 0 ^{0}C to T ^{0}C → $x \cdot C_{P,H_2O(g)} \cdot (T-0) = X \cdot C_{P,H_2O(g)} \cdot T$

When summing up the three steps they will together give the enthalpy for the water-air mixing to:

$$H = C_{P_{air}} \cdot T + X \cdot C_{P,H_2O(g)} \cdot T + \Delta H_{vap} \cdot X = \\ = \left(C_{P_{air}} + x \cdot C_{P,H_2O(g)}\right) \cdot T + \Delta H_{vap} \cdot X \quad (3.13)$$

The parameters for air and water at 0 ^{0}C are tabulated as following:

$$C_{P,air} = 1.0 \ \frac{kJ}{kg\ dry\ air,^0C}$$

$$C_{P,H_2O(g)} = 1.88 \ \frac{kJ}{kg\ water\ steam,^0C}$$

$$\Delta H_{vap} = 2500 \ \frac{kJ}{kg\ water}$$

Equation 3.13 above can therefore be rewritten to [34]:

$$H = (1.0 + 1.88X) \cdot T + 2500 \cdot X \quad (3.14)$$

To calculate the amount water that evaporates from the paper web the moisture content before and after the drying section of the paper web needs to be known. The fiber content in the paper web is constant through the whole drying section and it is only the water content that changes. The amount water that evaporates is calculated as following [15]:

$$m_{evap} = b_p \cdot v_{mach} \cdot w_p \cdot \left(MR_{before} - MR_{after}\right) \quad (3.15)$$

Where the moisture ratio (MR) for the paper web is calculated as the formulas below shows:

$$DMC\ \% = \frac{kg\ dry\ matter}{kg\ dry\ matter + kg\ water} \cdot 100 \quad (3.16)$$

$$MC\ \% = \frac{kg\ water}{kg\ dry\ matter + kg\ water} \cdot 100 = 100 - DMC \quad (3.17)$$

With this the MR can be decided according to formula 3.18 below:

$$MR = \frac{MC}{DMC} = \frac{kg\ water}{kg\ dry\ matter} \quad (3.18)$$

By setting up and calculate the balances for the elements in the drying section, it is known how much that flows in and out from the system. The results can be presented in a Sankey diagram that describes the energy flow [32]. The mass- and energy balances are set up due to the instantaneous settings for the parameters. When values for the mass-and energy balances are received a new mass-and energy balance could be set up for new parameter settings to see how the values for the balances change.

The air condition, especially the condition for the supply air and exhaust air affect the air consumption significantly. With higher air humidity for the supply air, the more air is needed to evaporate right amount water. If the exhaust air is too humid more supply air is needed to ventilate away the humid air [30].

3.3.4 What has been observed in earlier studies

In an earlier study of the paper machine it was observed that the steam- and electricity consumption were reduced without compromising the quality of the paper [28]. This was done by decreasing the velocity of the air fans and gave a decreasing of energy usage of 8 % in the drying section. The same results were observed in another study where the steam flow to the air-preheater in the drying section was decreased. This gave a temperature of the supply air in to the drying hood decreasing to 100 ºC from 112 ºC [32]. This means that there were no obvious increased evaporation rate with higher supply air temperature and would therefore only be waste of energy. The supply air temperature can't be too low either because this would give condensation of the water vapor in the drying hood. There exists an optimum temperature of how low the temperature of the supply air can be before it affects the heat transfer from the surrounding air to the paper web. This would increase the energy consumption in the cylinders instead [32]. With high temperature of the supply air more heat can be transferred to the paper web and more water is allowed to evaporate. If the heat transferring is to low then the steam in the cylinder has to evaporate more water, which leads to higher steam consumption. By comparing the specific heat consumption per kg evaporated water for different temperature of the supply air it could be seen how much steam that could be saved [32].

The air humidity in the exhaust air is also an important factor that affects the energy usage. The air humidity is adjusted by changing the velocity of the air fans with dew point controlling [35]. This means that if the velocity of the air fans is decreased this will give an increase of the humidity for the exhaust air and a lower air flow.

To fulfill the mass balance of air a decrease of the exhaust air will also give a decrease of the supply air flow [32]. By decreasing the air flow this will decrease the energy usage for the air fans. Except this electricity saving, more heat can be recovered from the exhaust air, which then would lead to less steam usage for air heating and an energy saving is also done for the steam. But there is a limit of how high the humidity for the exhaust air is allowed to be before the steam usage is increased. This is because too high air humidity decreases the driving force of mass transferring from the paper web to the surrounding air and would require more heat in form of steam in the cylinders to evaporate water from the paper web. By comparing the specific steam consumption for water with different air humidity for the exhaust air it shows how the air humidity affects the steam usage [32]. A study of how the air humidity for the exhaust air affects the steam consumption, showed that by increasing the humidity of the exhaust air from 0.10 kg water/ kg dry air to 0.14 kg water/kg dry air, gave a lower specific heat consumption of 4.6 %. This also gave a power decreasing for the air fans of 32 % [32]. It is desirable to have a low value of the specific heat consumption per kg water, near the value of ΔH_{vap}=3000 kJ/kg.

Another way to reduce the steam consumption is with controlling of the zero level. The zero level is the point where the pressure inside and outside the drying hood is equal to each other [35]. Dependent on where the zero level is set different amount leakage air in to the drying hood is received. By controlling the zero level the amount leakage air in could be minimized and also leakage of humid air out from the system is reduced and would result in more heat recovering according to [35].

Above the zero level there is overpressure and below the zero level there is under pressure. With too high zero level, an overpressure is observed and this overpressure means that the supply air is too low and would lead to the amount leakage air in to the system is increased [17]. This leakage air takes power from the system for heating and would therefore give higher steam consumption. If the zero level is too low an under pressure is observed. This will give high supply air flow and humid air is leaked out from the system and less heat is recovered [35].

When the drying section is run with dew- point and zero level controlling, a higher dew point without risk for condensation can be set for the drying section. This could be compared to the case when no controlling was included [35]. When no controlling is included a lower dew point needs to be set with respect to condensation and dripping of water on the paper web due to variations of the running. A controlled running would therefore give a lower power consumption [35].

In this master thesis the drying section (pre-dryer and after dryer) is evaluated where mass- and energy balances are set up according to the literature researching. To fulfill the balances values need to be found or measured. To the last, the flows are presented in a Sankey diagram and then analyzed to find energy efficiencies or see if there is potential for energy savings.

The key numbers for the specific heat consumption per kg evaporated water are calculated to see if it’s able to decrease the steam consumption for the drying section. This could be done with dew-point and zero level controlling and recover more heat or decrease the leakage air.

4. Method

In this chapter it is described how the measurements are made to fulfill the mass and energy balances and how to do the energy mapping and the detailed energy mapping

4.1 Energy mapping

The first step in an energy mapping is to identify the energy use units in the company and make a limitation of what should be included in the mapping. This could be done by studying flow sheets for the company that will give an overview of how the buildings and processes are located. This will also show the streams of what flows in and out from each section [24].

The next step of the energy mapping is to make an overview mapping of how much energy each unit consumes and with which kind of energy carrier. This could be done by collecting historical energy data, studying old invoice or by calculations for the latest year of how much energy each section consumes [24]. The data for usage of energy should be the actual values, measured and historical sampled for tracking in the future and give a reliable knowledge for the energy usage [36].

In this step an overview of the total energy usage for the company, divided on different energy units in the form of energy carriers is known and could be illustrated in Figure 7.

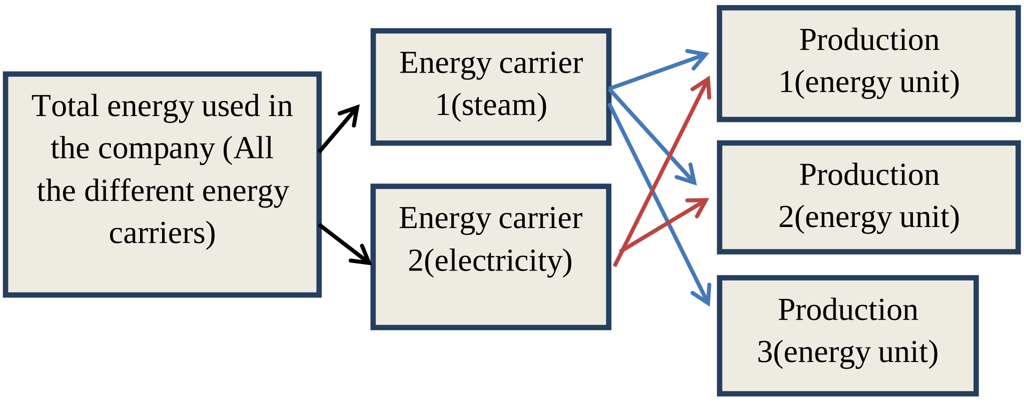

Figure 7 – The energy distributed on different energy carriers for different energy units

The distribution has to be good enough for determining which section that consumes most energy and that section is the ground for the detailed energy mapping in the later step. With this distribution of the total energy for the company it gives an overview of which sections consume most energy and with which kind of energy. It is easy to see which sections that are a useful energy unit for the company. The sections that are high consuming energy units and has a high potential of improvement of energy savings should be mapped in detailed first [22]. The cost-effectives suggestions of how to save energy and use energy efficiently should be included in the detailed energy mapping, where the cost for the suggestions is based on a LCC or the pay-back method [36].

By looking up the machines power and estimate the uptime, the energy used for one machine could be known and then by doing this for all machines that are included in each section, the total energy use for the machines are known [37]. Also by inventing the power for a stream and estimate the uptime for this stream, the yearly energy use for can also be calculated in the same way as for the machines. The yearly energy use for a specific machine or a stream could be calculated with following formula [37]:

$$Yearly\ energy\ use\ (kWh/year) = Power(kW) \cdot uptime(hours/year) \tag{4.1}$$

4.2 Detailed energy mapping

In the detailed energy mapping it could be seen how the energy carriers is used for that section it goes into. In the over viewing energy mapping it is only seen how the total energy of the company is distributed for the different sections. For the section that should be energy mapped in detail the flow sheet should be studied and from the flow sheet the different flows in the process could be defined. The flows could be material flows, product flows, byproducts and energy related flows like steam or fuels. By identifying these flows, mass- and energy balances could be set up with the thermodynamics first law and this will give knowledge of how the energy carriers are distributed in the section [38].

When the mass- and energy balances are set up it is known which kind of data that is needed to calculate this set up of balances. These data could be historical sampled or collected from measurements and calculations as complements. The mapping should be done at the place for the section that should be mapped. A visit should also be included to give a representative picture of the energy using. When all the balances are calculated the energy flows are known and they could be presented in a flow sheet that will give a good illustrative picture of how the energy is used [38].

In this master thesis a detailed energy mapping of the drying hood in the paper machine 1 should be done and the different sections for the paper machine are:

- Forming section
- Pressing section
- Pre-dryer section
- Sizing
- After-dryer section

For the drying section that is studied in detailed following balances is set up according to the energy usage:

- Water balance
- Steam balance
- Paper balance
- Air balance

The steam that is used in the drying section is used for heating cylinders and heating the supply air that goes in to the drying hood. How much steam is consumed is dependent on different kind of paper and therefore the energy use is different [11]. An overview for the whole drying section is shown in Figure 8.

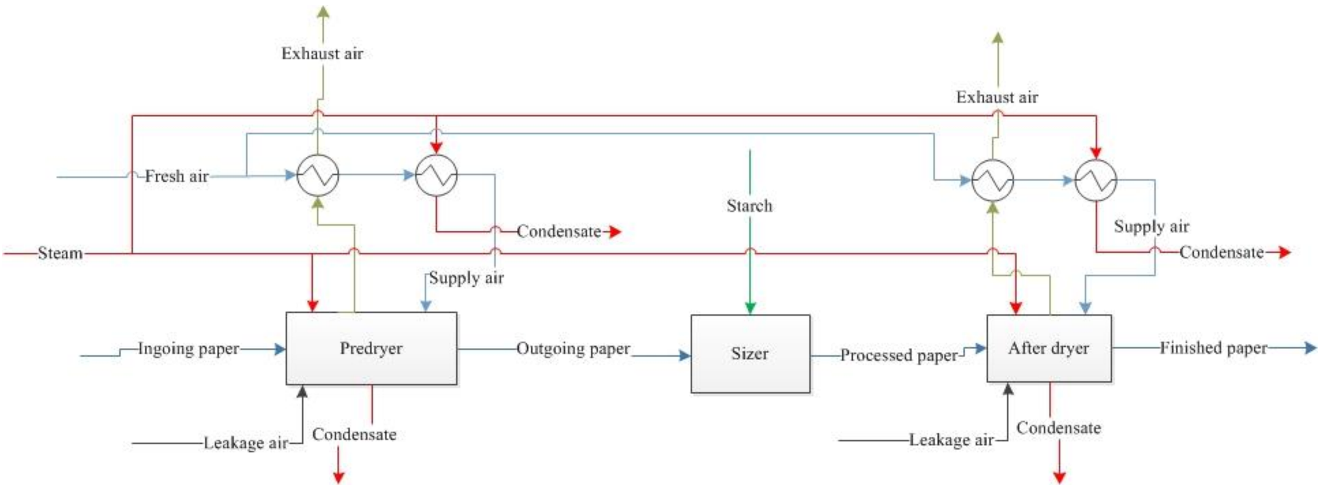

Figure 8 - Overview for the whole drying section

4.3 Measurements of airflows

To fulfill the mass- and energy balances, the different airflows in and out from the drying hood needs to be measured. This is done with a pitot tube where the difference between dynamic and static pressure is measured. A pitot tube (2) is a curved tube with the opening against the air flow. At the main tube where the flow is measured there is also a reference tube (1) that will measure the static pressure. See Figure 9. In these two tubes no flowing of air occurs and the air is stagnant. The difference between these two pressures gives the velocity of the flows [34].

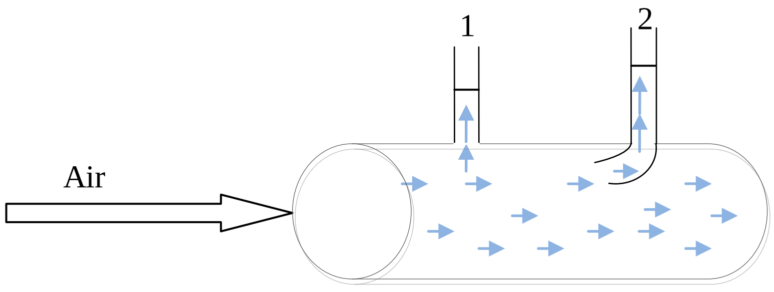

Figure 9 - Illustration for measurement of airflow with a pitot tube

At the stagnations point, the fluid inside the pitot tube is stagnant and this gives the velocity at this point to zero [34]. With Bernoullis equation the flow can be calculated to:

$$p_1 + \frac{\rho {v_1}^2}{2} = p_2 + \frac{\rho {v_2}^2}{2}, v_2 = 0 \tag{4.2}$$

$$\rightarrow v_1 = \sqrt{\frac{2\Delta p}{\rho}} \tag{4.3}$$

The density ρ is for the humid air and is calculated as follows [34]:

$$\rho = \rho_t(1 + X) \tag{4.4}$$

The density for the humid air can also be decided from the Mollier chart if the relative air humidity and the wet bulb temperature are known.

To get the volume flow the velocity is multiplied with the cross-sectional area where the flow is going through.

$$F = v \cdot A \tag{4.5}$$

From this the mass flow of air can be calculated if the density for the dry air is known as following:

$$G = F \cdot \rho_t \tag{4.6}$$

Putting formula 4.5 and 4.6 together gives directly the mass flow of dry air according to

$$G = v \cdot A \cdot \rho_t \tag{4.7}$$

Where ρ_t is decided with formula 4.4 above or with Mollier chart.

4.4 Measurement of air humidity

With a wet thermometer and a dry thermometer the air humidity and relative humidity can be determined. The wet thermometer is surrounded with a wet cloth and since the air flow isn't saturated there is a concentration gradient of water vapor in the air and the wet cloth. This driving force is determined with the partial pressure of the water vapor in the surface of the wet cloth and the air. The air isn't saturated water will evaporate from the wet cloth. For evaporating the water from the wet cloth energy is needed and this energy is taken from the water and little from the air and the temperature from the water will decrease. This decreasing temperature of water will continue till the temperature difference between air and water is large. The energy for evaporating water is then only taken from the air. Thermal equilibrium is reached, which means that the heat transferring from air to the wet cloth is equal to the energy that is needed for evaporating water from the cloth and this gives the wet-bulb temperature [34]. This is described in detail in Appendix B! In the Mollier chart there are constant lines for the wet bulb temperature T_v and by measuring the dry temperature and the wet temperature the air humidity and relative air humidity could be determined [34].

4.5 Measurement of condensate flow and condensate temperature

To fulfill the mass balance for the steam-and condensate according to equation 3.2, the condensate flow out from condensate tanks and the steam flow in drying section need to be decided. The condensate is taken out with pumps and the pumps are used as help for determine the condensate flow. For each pump there is a characteristic pump curve that describes how the increased pressure from the pump varies with different volume flows for different diameters of the pump wheel. The increased pressure from the pump is calculated as following [34]:

$$\Delta p_{pump} = p_t - p_s \tag{4.8}$$

The increased pressure from the pump gives a head that is calculated as [34]:

$$H_{pump} = \frac{\Delta p_{pump}}{\rho g} \tag{4.9}$$

This means that each volume flow of liquid will give a head dependent on which pump that is used. When measuring the increased pressure from the pump and transforming this pressure to a head with equation 4.9 the volume flow could be decided from the pump curve.

The axial power for the pump varies dependent on the volume flow and there is a relation between the axial power and the volume flow. These two properties are also presented in the pump curve and therefore by measuring the axial power for the pump, the volume flow can be determined. The theoretical axial power for the pump is calculated as follows [34]:

$$P_{axial} = \Delta p_{pump} \cdot F \tag{4.10}$$

But because of frictions and losses in the pump more power is required to fulfill the needed power for the pump and by taking the pumps efficiency in consideration the real power can be calculated to [34]:

$$P_{axial} = \frac{\Delta p_{pump} \cdot F}{\eta_{pump}} \tag{4.11}$$

For the engine that runs the pump; its power is dependent on the work from the pump. With an efficiency η_{engine} for the engine the electrical power that the engine needs to run the pump is [34]:

$$P_{engine} = \frac{\Delta p_{pump} \cdot F}{\eta_{pump} \cdot \eta_{engine}} \tag{4.12}$$

For a three-phase engine the electrical power is calculated as follows [39]:

$$P_{engine} = I \cdot U \cdot \sqrt{3} \cdot cos\varphi \tag{4.13}$$

The current is measured with an ammeter. The voltage and power factor is constant and is looked up in the data sheet for the pump and with this the electrical power for the engine is determined.

By rewriting formulas 4.11-4.13 a relation between the axial power and the electrical power for the motor is observed as follows:

$$P_{axial} = \frac{P_{engine} \cdot \eta_{pump} \cdot \eta_{engine}}{\eta_{pump}} = P_{engine} \cdot \eta_{engine} \tag{4.14}$$

In the condensate tank the steam and condensate is in equilibrium, which means that the temperature and pressure of the steam are the same as for the condensate [33]. They are in equilibrium since some steam from the condensate is used as flash steam and some is sent out from the condensate tank as condensate, both phases (liquid and steam) has the same pressure and temperature. Figure 10 shows the equilibrium between the steam and condensate phase.

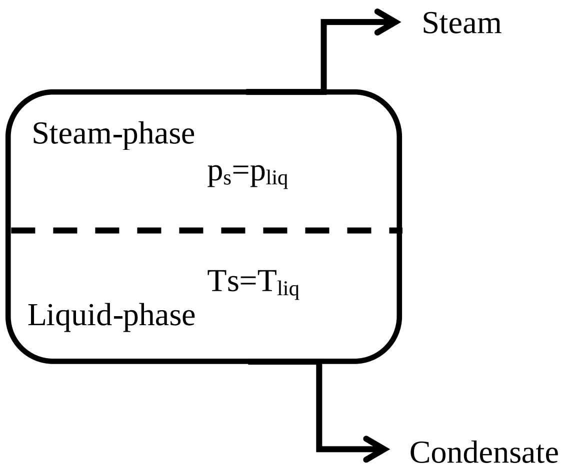

Figure 10– Condensate tank with both steam and liquid phase

Antione's equation describes how the steam-pressure for only one component in a liquid correlates with the temperature. If the steam-pressure in the condensate tank is known the condensate temperature can be calculated with Antoine's equation [40]:

$$\log P_i^{\,0} = A_i - \frac{B_i}{C_i + T} \tag{4.15}$$

Where the parameters for water is:

$A = 8.19$
$B = 1730.63$
$C = 233.43$

4.6 Measurement of blow through steam flow

The blow through steam that goes to the surface condenser is the blow through steam that leaves the drying section. To measure the blow through steam flow to the surface condenser, the cold water flow in to the surface condenser and temperature difference for the cold water flow is measured. This will give the amount of energy provided by the steam for water heating when the steam condenses. With the thermodynamic first law an energy balance could be set up to calculate the steam flow.

The cold water flow is measured with an ultrasonic flow meter that measures the velocity of a fluid with ultrasound and will give the volume flow since the area for the pipe is known. There are two sensors at this instrument, one transmitter and one receiver. The transmitter sends out a signal through the fluid and the signal are then detected at the receiver where the time is registered and the volume flow is then calculated. The Figure 11 below illustrates the phenomena for this instrument.

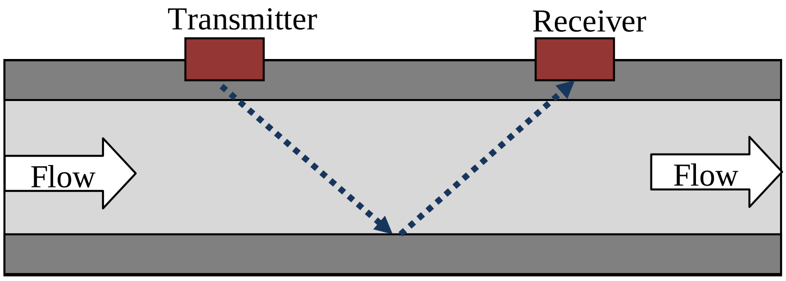

Figure 11-Measuring of water flow with an ultrasonic flow meter

The temperature for the cold water flow in and out from the surface condenser is measured with a contact thermometer outside of the pipes.

5. Results

This chapter presents the calculations for the amount evaporated water from estimated values, the results from the experimental measurements. The flow sheets for the processes where the measurements have been made are also included. The measured values are then used for calculating the mass-and energy balances which are also are presented in this chapter with Sankey diagrams.

5.1 Pre-drying section

The estimated moisture content for the paper web before it goes in to the pre-drying section is $MC_{before\ pre\text{-}drying}$=53 % and the moisture content for the paper web out from the pre-drying section is $MC_{after\ pre\text{-}drying}$=2 % [6]. It is here chosen to make calculations for paper with a produced grammage of 80 g/m^2 since it is the most common paper. With the given moisture content the amount evaporated water from the paper web is calculated with formula 3.15 where following the parameters is used:

$$b_p = 6790\ mm = 6.790\ m$$

$$v_{mach} = 915\ m/min = \frac{915}{60}\ m/s$$

$$Weight\ for\ finished\ paper = 80\ g/m^2$$

The variations for the paper web width in the drying section can be seen in Appendix C and it is here chosen to use the web width that the paper has at the inlet of drying section.

Before using equation 3.15 the dry material content that goes in to the pre-dryer section needs to be determined first were only the fiber that is seemed as the dry material inlet. With estimated moisture content of $MC_{after\ after\text{-}drying\ section}$=4.1 % [6] for the finished paper the amount dry material in the paper is:

$$w_{fib+starch} = 80 \cdot (1 - 0.041) = 76.7\ g/m^2 \tag{5.1}$$

The amount dry starch that is added is 2.5 g/m^2 [41] and the fiber content for the paper is calculated to:

$$w_{fib} = 76.7\ g/m^2 - 2.5\ \text{g/m}^2 = 74.2\ g/m^2 \tag{5.2}$$

Inserted in equation 3.15 gives the amount evaporated water from the paper web in the pre-drying section to:

$$m_{evap} = 6.790\ m \cdot \frac{915}{60} m/s \cdot 74.2 g/m^2 \cdot \left(\frac{53}{47} - \frac{2}{98}\right) = 8.5\ kg/s$$

5.2 Sizer

In the sizer the paper is rewetted again with starch. The total water in to the sizer is composed of water from the paper web and the starch. The total inlet water to the sizer is equal to the total amount of water out from the sizer. This is also the amount of water in to the after-dryer and can be illustrated in Figure 12.

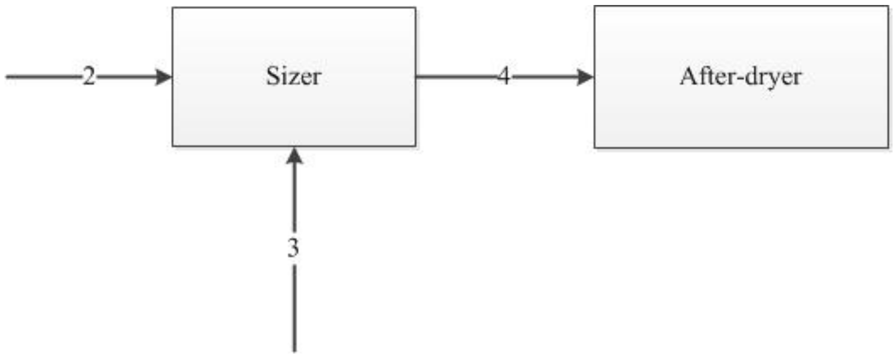

Figure 12-Paper (stream 2) and starch (stream 3) in to the sizer

A water balance over the sizer according to Figure 12 gives following mass balance:

$$m_{H_2O}(2) + m_{H_2O}(3) = m_{H_2O}(4) \tag{5.3}$$

Data for starch:

$2.5\ g\ DS/m^2$ – correspond to a dry concentration of 12.4 % of the starch [41].

$S = \frac{2.5\ g\ DS/m^2}{0.124} = 20.16\ g/m^2$- The starch that is added to the paper web.

The paper web that comes in to the sizer got a new width because of the shrinkage in the pre-drying and is now estimated to $b_{paper} = 6640\ mm = 6.64\ m$ see Appendix C. The amount water that is added from the starch to the paper in stream 3 according to Figure 12 above is calculated to:

$$m_{H_2O}(3) = b_p \cdot v_{mach} \cdot S \rightarrow \tag{5.4}$$

$$\rightarrow 6.64\ m \cdot \frac{915}{60} m/s \cdot 20.16\ g/m^2 \cdot (1 - 0.124) = 1.79\ kg/s$$

The amount water that comes in to the sizer with the paper in stream 2 is calculated to:

$$m_{H_2O}(2) = 6.64\ m \cdot \frac{915}{60}\ m/s \cdot 74.2\ g/m^2 \cdot \left(\frac{2}{98}\right) = 0.15\ kg/s$$

And the total amount water in the paper out from the sizer is calculated with equation 5.3 to:

$$m_{H_2O}(4) = 0.15\ kg/s + 1.79\ kg/s = 1.94\ kg/s$$

The fiber content in the paper is constant through the pre-drying section and when it comes to the sizer a dry layer of starch is added and the total dry material content increases. The moisture content for the finished paper is estimated to MC=4.1 % [6] . With this value the total dry material content in the paper web (fiber+starch) out from the sizer can be determined for the grammage of 80 g/m^2.

$$Total\ dry\ material\ content\ out = 76.7\ g/m^2 \cdot 6.64\ m \cdot \frac{915}{60}\ m/s\ \ = 7.77\ kg/s$$

This would give moisture content in the paper out from the sizer calculated with equation 3.17 to:

$$MC\ \% = \frac{1.94\ kg/s}{1.94\ kg/s\ + 7.77\ kg/s} \cdot 100 = 20\ \%$$

5.3 After-drying

The inlet moisture content for the paper web is based on the calculated value of 20 % and the paper web is dried to the estimated moisture content of 4.1 % [6]. The evaporated water in the after drying section is calculated in the same way as for the pre-drying section but with different moisture ratio and paper web width. The paper web width is the same as in the sizer, see Appendix C for the shrinkage of the paper.

$$m_{evap} = 6.64\ m \cdot \frac{915}{60} m/s \cdot 74.2 g/m^2 \cdot \left(\frac{20}{80} - \frac{4.1}{95.9}\right) = 1.60\ kg/s$$

5.4 Measurements airflows and air humidity

The measured points for the different inlet and outlet airflows in the pre-drying section are shown below in Figure 13. Where the dry temperature, wet bulb temperature, area and pressure difference is measured. The mean values for these measured parameters are presented in Table 3.

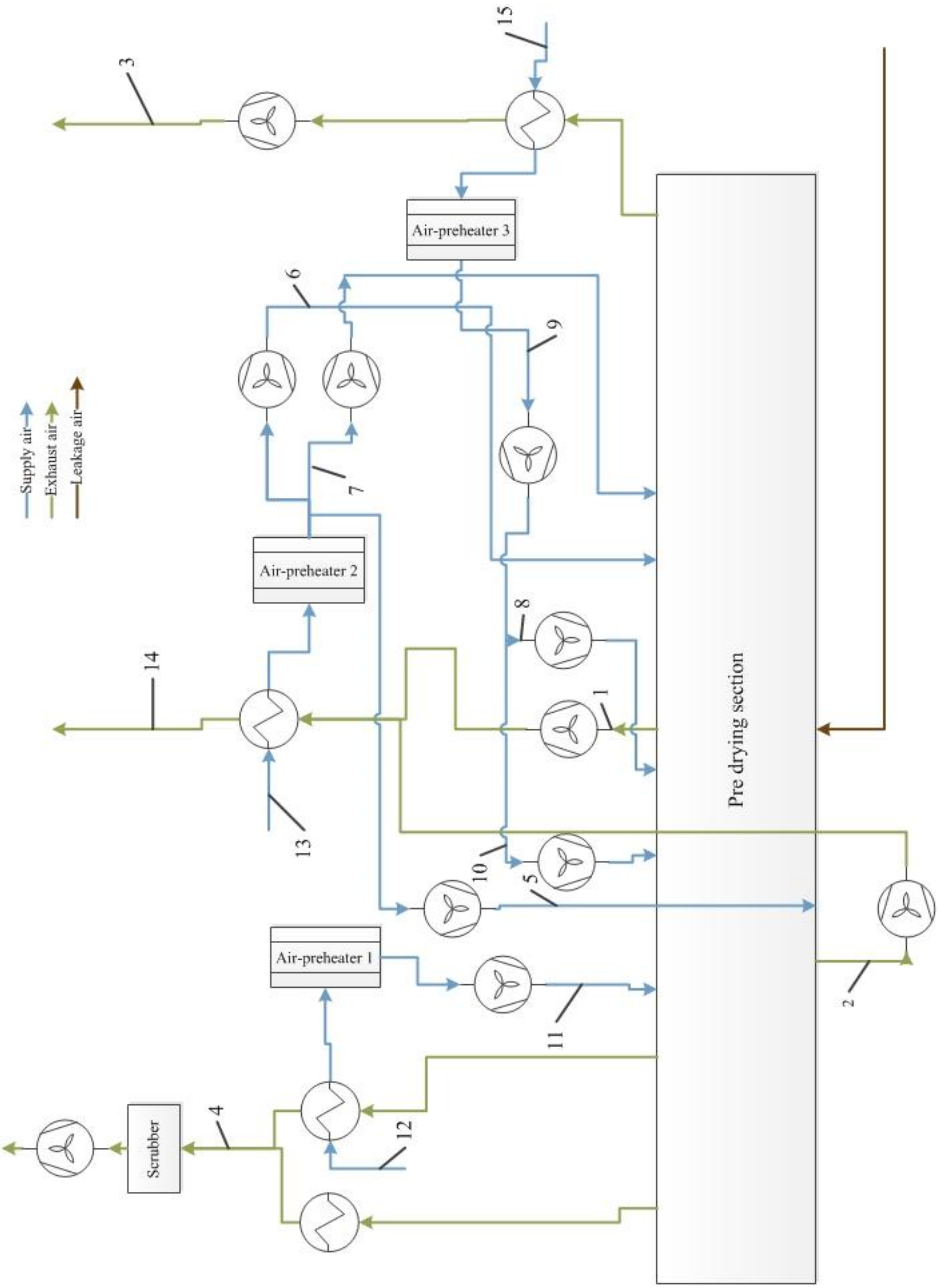

Figure 13- Air system in the pre-drying section

Table 3 – Measured values of the wet bulb temperature, the dry temperature and the pressure

Measure point	T [^{0}C]	T_v [^{0}C]	Δp [Pa]	Area A [m^2]
1	87.7	65.3	1.4	1.25
2	74.0	56.2	25.0	0.50
3	61.9	61.7	15.0	3.12
4	46.8	46.2	67.0	1.23
5	108.2	48.3	142.0	0.80
6	88.2	40.2	23.0	0.46
7	88.2	40.2	23.0	0.39
8	92.7	51.5	23.0	0.50
9	100.2	44.3	46.0	0.50
10	96.7	39.5	126.0	0.17
11	102.3	41.9	130.0	0.90

The wet-bulb temperature and the dry temperature in Table 3 are used the Mollier chart for humid air for deciding the air humidity, the relative air humidity and the wet density (see Appendix D). The results are presented in Table 4 below.

Table 4- The air humidity, relative air humidity and wet density decided from the measured values

Measure point	X [kg H_2O/kg dry air]	Relative air humidity ϕ	Wet density ρ [kg humid air /m^3 humid air]
1	0.19	0.35	1.05
2	0.11	0.40	1.05
3	0.17	1.0	0.92
4	0.06	1.0	1.60
5	0.05	0.06	1.10
6	0.03	0.07	1.10
7	0.03	0.07	1.10
8	0.07	0.15	1.05
9	0.06	0.06	1.15
10	0.02	0.04	1.10
11	0.03	0.04	1.10

Streams 12, 13 and 15 (Figure 13) are airflows that are taken from the hall and the temperature for these air flows are therefore the temperature for respective hall seen in Table 5. The air humidity for the air stream before and after heating is the same [33] and therefore the air humidity for these streams is the same as the values in Table 4.

Table 5-The temperature of the halls

Air flow [pre-dryer]	T_{hall} [°C]
12	38.4
13	41.3
15	36.0

With formula 4.3 the velocity for the air flows is calculated with the pressure Δp and the wet density shown in Table 3 respectively Table 4. The results for the velocity of the air flows are presented in Table 6 for all the measured points in pre-drying section.

Table 6-The velocity for the air flows

Measure point	Velocity v_i [m/s]
1	1.63
2	6.90
3	5.71
4	9.15
5	16.10
6	6.47
7	6.47
8	6.62
9	8.94
10	15.10
11	15.40

To calculate the dry mass air flow G, the dry density is needed. The wet density ρ and the air humidity X is known the dry density can be rewritten with formula 4.4 to:

$$\rho_t = \frac{\rho}{(1+X)} \tag{5.5}$$

For measure point 1 this gives the dry density to:

$$\rho_t = \frac{1.05\ kg\ humid\ air/m^3 humid\ air}{(1+0.19\ kg\ water/kg\ dry\ air)} = 0.88\ kg\ dry\ air/\ m^3\ humid\ air$$

For the rest of the measure points the dry density is presented in Table 7.

Table 7-The dry density for the air flows

Measure point	Dry density ρ_t [kg dry air /m³ humid air]
1	0.88
2	0.95
3	0.80
4	0.15
5	1.05
6	1.07
7	1.07
8	0.98
9	1.08
10	1.08
11	1.07

The dry mass air flow is calculated with formula 4.7 and for measure point 1 the mass flow of dry air is:

$$G_1 = 1.63\ m/s \cdot 1.25\ m^2\ \cdot 0.88 \text{ kg dry air } /m^3\ humid\ air = 1.80\ kg\ dry\ air/s$$

For the rest of the measure points the mass flow of air are presented below in Table 8.

Table 8- The mass flow of air

Measure point	Mass flow of air G_i [kg dry air/s]
1	1.80
2	3.28
3	14.25
4	17.11
5	13.47
6	3.19
7	2.73
8	3.26
9	4.88
10	2.83
11	14.78

The same measurements and calculations were done for the after-drying section. The measured points for the airflows for the after-drying section are shown in Figure 14.

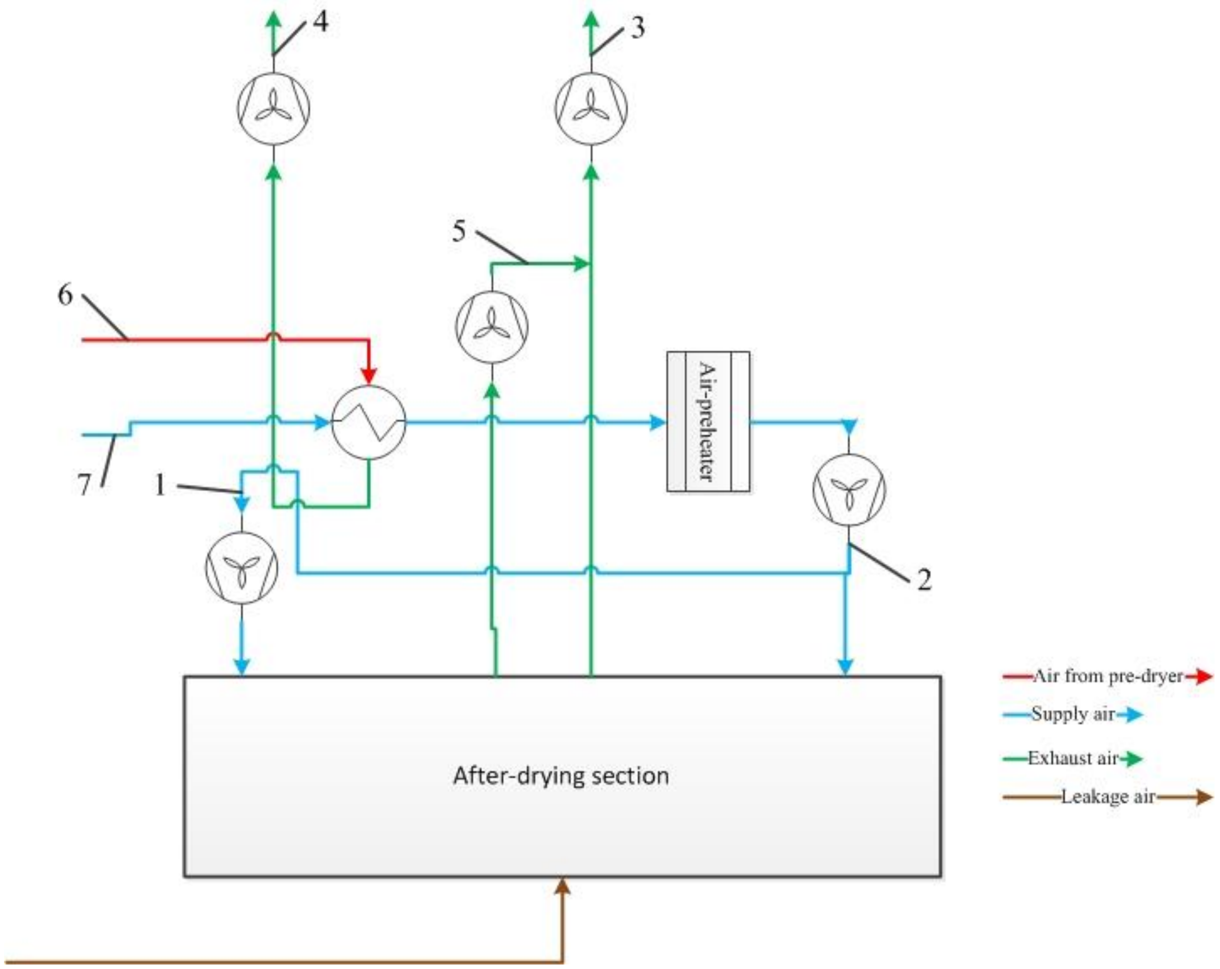

Figure 14- Air system in the after-drying section

Table 9 presents the mean values of the measured parameters in the after-drying section.

Table 9- Measured values of the wet bulb temperature, the dry temperature and the pressure

Measure point	T [^{0}C]	T_v [^{0}C]	Δp [Pa]	Area A [m²]
1	76.5	38.0	47.3	0.12
2	78.9	41.0	27.7	1.16
3	67.7	49.5	2.0	2.0
4	37.8	37.2	7.0	1.13
5	72.9	53.3	17.0	0.10
6	79.5	57.7	5.2	3.0

In the same way as for the pre-drying section, the air humidity, the relative air humidity, the wet density is decided from the Mollier chart found in Appendix D and the results are presented in Table 10.

Table 10- Values from Mollier chart from the measured values of wet bulb temperature and dry temperature

Measure point	X [kg H_2O/kg dry air]	Relative air humidity ϕ	Wet density ρ [kg humid air /m^3 humid air]
1	0.025	0.10	1.15
2	0.040	0.15	1.15
3	0.078	0.40	1.05
4	0.040	1.00	1.10
5	0.092	0.47	1.10
6	0.030	0.40	1.05

Stream 7 /Figure 14) is an airflow that is taken from the hall and the temperature in this hall is T_7=35.3 ºC. The air humidity for this air stream is the same as the air humidity for measure point 2 since the air humidity before and after heating should be the same [33].

The velocity and dry density calculated with equation 4.3 respectively 5.5 of the airflows in the after-drying section are presented in Table 11.

Table 11- The velocity of the measured airflows and the dry density

Measure point	Velocity v_i [m/s]	Dry density ρ_t [kg dry air /m^3 humid air]
1	9.07	1.12
2	6.94	1.10
3	1.95	0.97
4	3.57	1.06
5	5.56	1.00
6	3.15	0.95

The dry density is known and the mass flow of air can be calculated and are presented in Table 12.

Table 12-The dry mass air flow in the after-drying section

Measure point	Mass flow of air G [kg dry air/s]
1	1.22
2	9.22
3	3.82
4	4.27
5	0.54
6	8.93

5.5 Mass balances for the drying section

To set up the mass balances for the pre-drying section and after-drying section every inlet and outlet streams need to be defined. Besides the airflows that make up the air-system there is also a steam-condensate system. This is illustrated in Figure 15 for the pre-drying section and Figure 16 for the after-drying section, and these makes up a cascade-system.

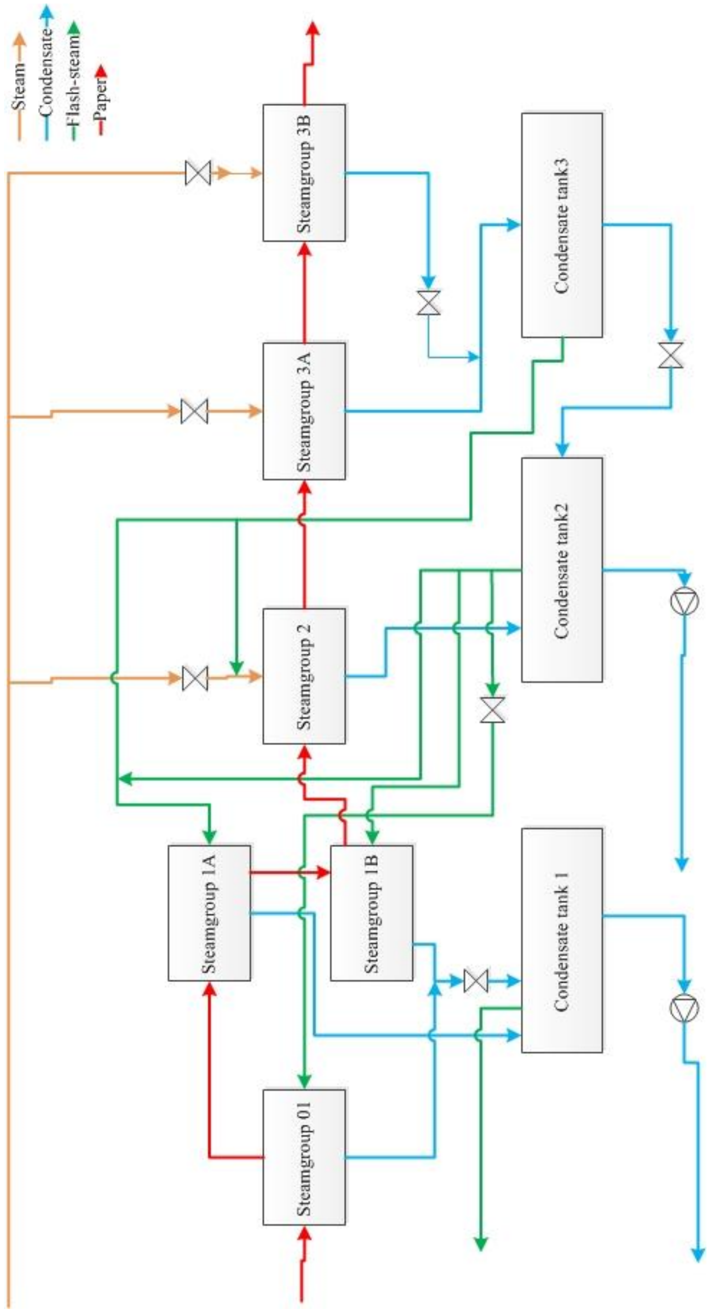

Figure 15- Cascade system for the pre-drying section with the different steam-groups

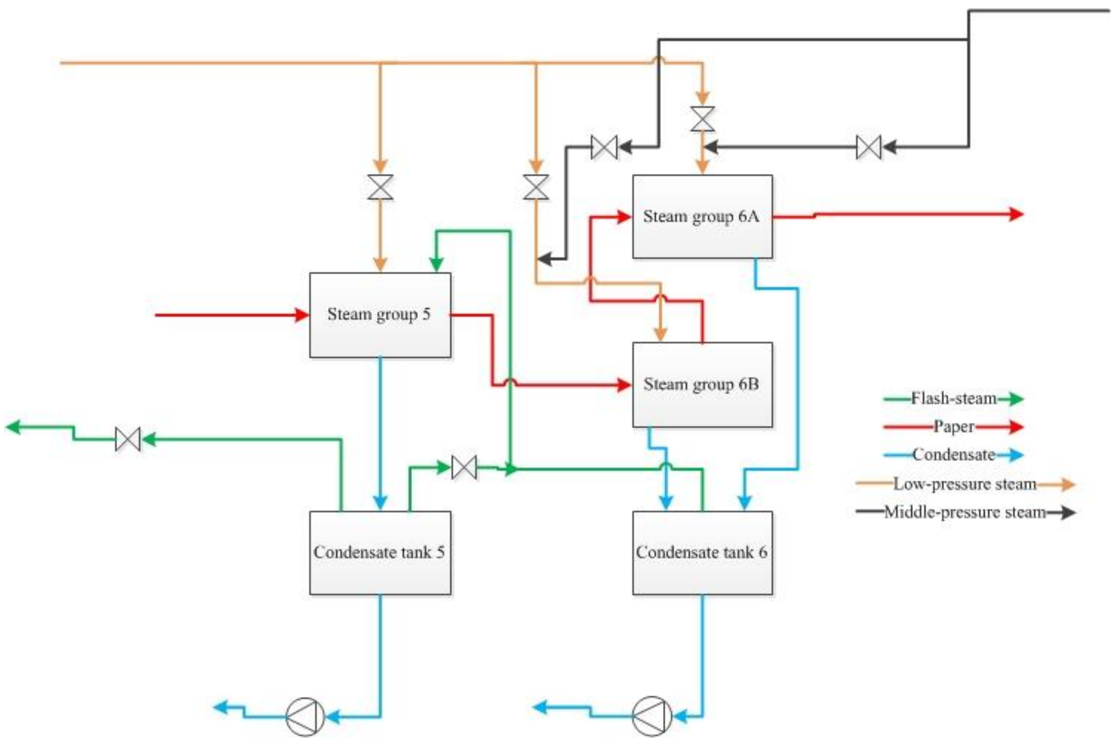

Figure 16- Cascade system for the after-drying section with the different steam-groups

Both the pre-drying section and after-drying section is composed of the cascade system and the air system. Putting together the air-system and the cascade system a flow chart for the whole pre-drying section and after-drying section are observed as Figure 17 respectively Figure 18 shows.

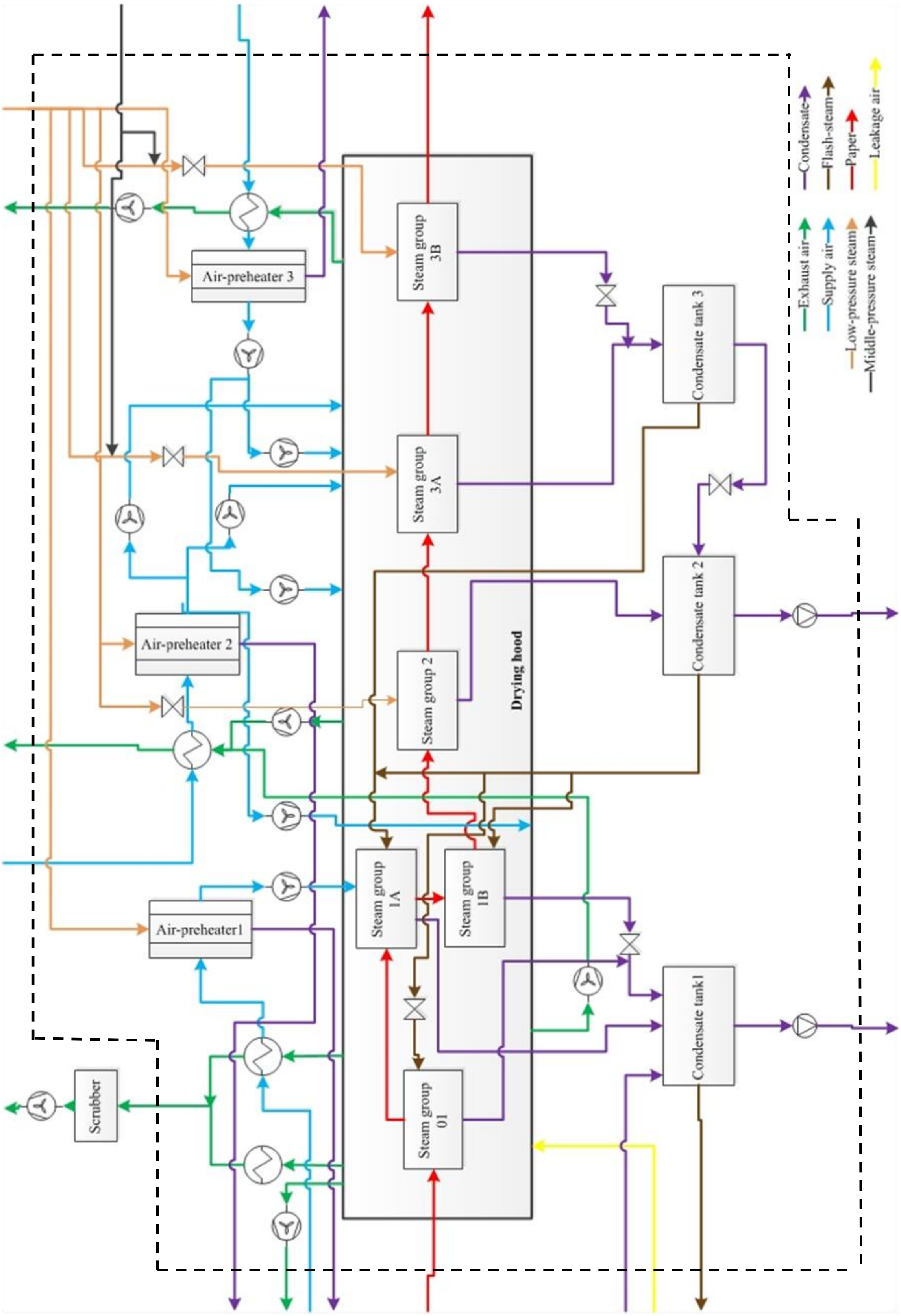

Figure 17-The pre-drying section with air-system and cascade system

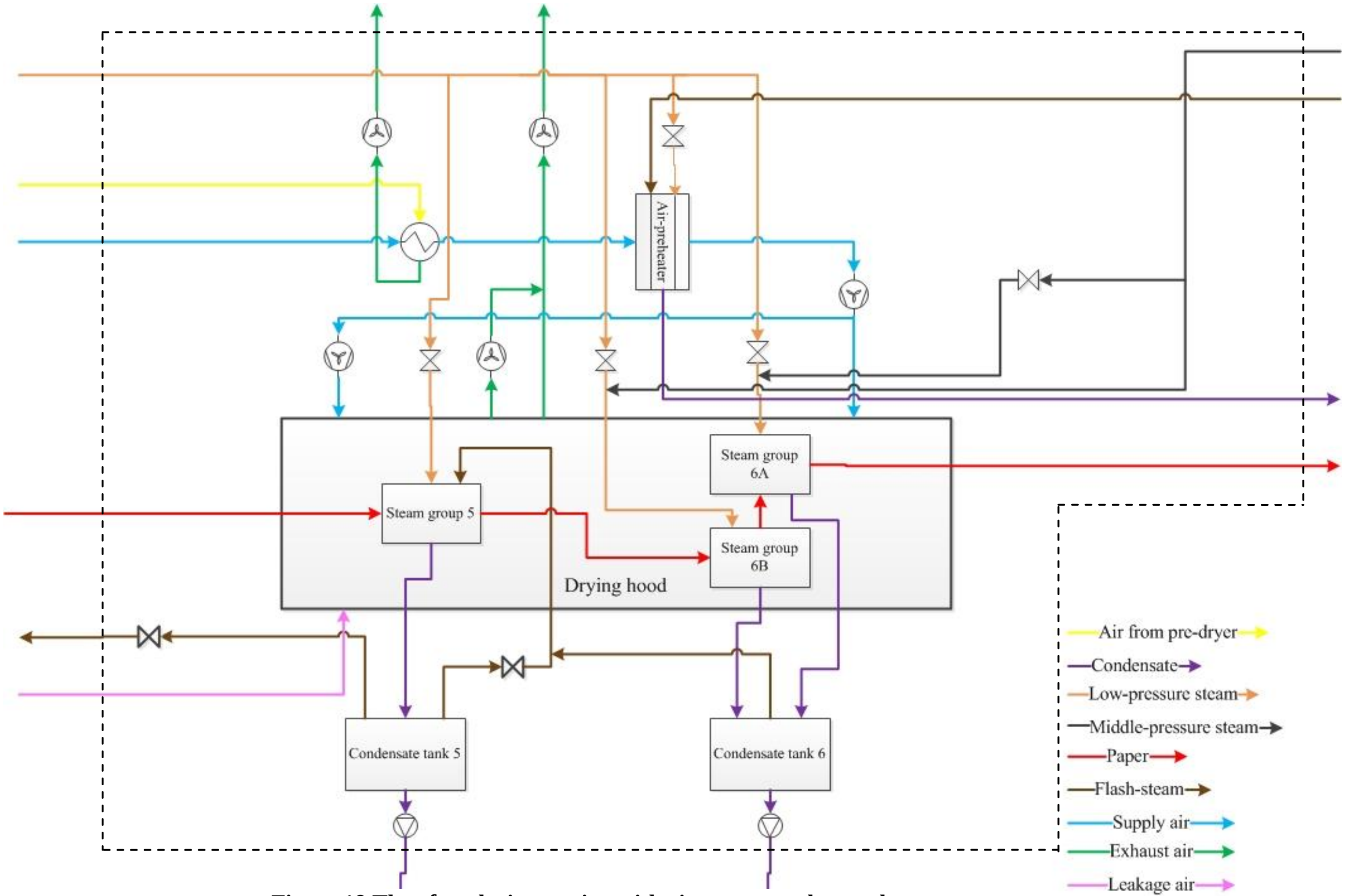

Figure 18 The after-drying section with air-system and cascade system

In Figure 17 and Figure 18 where boundary lines are drawn it is seen that the streams that cross the boundary line are:

- Condensate from condensate tanks
- Condensate from steam battery
- Low- and middle pressure steam
- Flash steam/blow through steam
- Supply air flow
- Exhaust air flow
- Paper in and out
- Air from pre-dryer section
- Leakage air

5.5.1 Air

The mass balance of air for the drying section is calculated with formula 3.3. The inlet airflows in the pre-drying section according to the flow chart in Figure 13 are 5, 6, 7, 8, 9, 10, and 11. The values for the mass flows are found in Table 8. As been seen in Figure 13, stream 9 is divided into 2 streams (8 and 10) and therefore the sum of the supply air is composed of:

$$\sum G_{sup,pre-dryer} = G_5 + G_6 + G_7 + G_9 + G_{11} = 39.0\ kg\ dry\ air/s$$

For the exhausted air, it is composed of stream 1, 2, 3 and 4 according to the measured points in Figure 13.

$$\sum G_{exh,pre-dryer} = G_1 + G_2 + G_3 + G_4 = 36.0\ kg\ dry\ air/s$$

For the mass balance of air in the after-drying section it is seen in Figure 14 that the inlet airflows are 2 and 6. The outlet air flows are 3 and 4 and the sum of the supply air and exhaust air can be calculated. The values of the air flows are found in Table 12.

$$\sum G_{sup,after-dryer} = G_2 + G_6 = 18.0\ kg\ dry\ air/s$$

$$\sum G_{exh,after-dryer} = G_3 + G_4 = 8.0\ kg\ dry\ air/s$$

5.5.2 Water

For the water balance where the water comes in with the paper web, the supply air and leakage air, the total amount water in to drying is calculated with equation 3.4. The values of X and G are presented in Table 4 and Table 8 for the pre-drying section and Table 10 and Table 12 for the after-drying section.

For the pre-drying section the left hand side of equation 3.4 (inlet) is calculated as follows:

$$G_{H_2O(g),in} = G_5 \cdot x_5 + G_6 \cdot x_6 + G_7 \cdot x_7 + G_9 \cdot x_9 + G_{11} \cdot x_{11} + m_{evap} + \sum G_{leak} \cdot X_{leak} = 10.10\ kg\ water/s + \sum G_{leakage\ air} \cdot X_{leakage\ air}$$

The right hand side of equation 3.4 (outlet) for the pre-drying section is calculated to:

$$G_{H_2O(g),out} = G_1 \cdot x_1 + G_2 \cdot x_2 + G_3 \cdot x_3 + G_4 \cdot x_4 = 4.10\ kg\ water/s$$

The same water balance was set up for the after-drying section were following results were observed:

$$G_{H_2O(g),in} = G_2 \cdot x_2 + G_6 \cdot x_6 + m_{evap} + \sum G_{leak} \cdot X_{leak} = 2.96\ kg\ water/s + \sum G_{leak} \cdot X_{leak}$$

$$G_{H_2O(g),out} = G_3 \cdot x_3 + G_4 \cdot x_4 = 0.470\ kg\ water/s$$

The mass flows of the outlet water and air are too small to fulfill the balances of what goes in is equal to what goes out. This is observed in both the pre-drying section and the after-drying section. This also gives that no leakage air comes in to the system. The outlet airflows are therefore adjusted to be larger in the way that each air flow still keeps its percent of the total outlet airflow at the same time as the water balance is fulfilled.

The percent of each airflow is calculated as following:

$$y_{airi} = \frac{G_{i,out}}{Total\ outflow} \cdot 100 \qquad (5.6)$$

The results for the percent of each outgoing airflow in the pre-drying section and after-drying section are presented in Table 13.

Table 13-The part that each outgoing airflow corresponds to in the pre-drying section and after-drying section

Air flow	Pre-dryer[%]	After-dryer[%]
1	5.0	-
2	9.1	-
3	38.9	47.3
4	47.0	52.7

With the mass balances for air-and water an equation system is set up as a help to calculate how much the total exhaust air flow should be to fulfill the water balance. By explicating these two balances and putting in the percent of the outgoing airflow following equation system is observed for the pre-drying section:

$$\sum G_{sup} \cdot X_{sup} + \sum m_{evap} + \sum G_{leak} \cdot X_{leak} =$$
$$y_{air1} \cdot \sum G_{exh} \cdot X_1 + y_{air2} \cdot \sum G_{exh} \cdot X_2 + y_{air3} \cdot \sum G_{exh} \cdot X_3 + y_{air4} \cdot \sum G_{exh} \cdot X_4$$
(water balance)

$$\sum G_{sup} + \sum G_{leak} = \sum G_{exh} \text{ (air balance)}$$

Putting in all the known numbers and an assumption for the humidity of the leakage air is $X_{leak} = 0.015 \ \frac{kg\ water}{kg\ air}$ [15] the equation system now looks like:

$$10.10\ kg\ water/s + \sum G_{leak} \cdot 0.015 \frac{kg\ water}{kg\ air} = 0.05 \cdot \sum G_{exh} \cdot 0.19 + 0.091 \cdot \sum G_{exh} \cdot 0.11 + 0.389 \cdot \sum G_{exh} \cdot 0.17 + 0.47 \cdot \sum G_{exh} \cdot 0.06$$

$$39.0\ kg/s + \sum G_{leak} = \sum G_{exh}$$

There are two equations and two unknowns, G_{exh} and G_{leak}, which gives a soluble system. This equation system is solved in the program Matlab and following results were observed for the pre-dryer:

$\sum G_{exh} = 96.3\ kg/s$

$\sum G_{leak} = 57.2\ kg/s$

The same equation system was solved for the after-drying section but with corresponding numbers for the after-dryer and the observed airflows are:

$\sum G_{exh} = 62.6\ kg/s$

$\sum G_{leak} = 44.5\ kg/s$

The new outgoing airflows based on the adjusted total outgoing air flow are calculated with equation 5.6 and the results are presented in Table 14.

Table 14-The new adjusted outgoing airflows

Air flow	Pre-dryer [kg/s]	After-dryer [kg/s]
1	4.8	-
2	8.8	-
3	37.4	29.6
4	45.3	33.0

With these outgoing airflows the water balance in the pre-drying section is now calculated to:

$$G_{water,in} = 10.1\ kg\ water/s + 57.2\ kg/s \cdot 0.015 \frac{kg\ water}{kg\ air} = 10.95\ kg\ water/s$$

$$G_{water,out} = G_1 \cdot x_1 + G_2 \cdot x_2 + G_3 \cdot x_3 + G_4 \cdot x_4 + G_{12} \cdot x_{12} = 10.95\ kg\ water/s$$

The water balance for the after-drying section with the new outgoing air-flows is:

$$G_{water,in} = 2.96\ kg\ water/s + 44.5\ kg/s \cdot 0.015 \frac{kg\ water}{kg\ air} = 3.63\ kg\ water/s$$

$$G_{water,out} = G_3 \cdot x_3 + G_4 \cdot x_4 = 3.630\ kg\ water/s$$

It is the adjusted outgoing airflows in Table 14 and the new water balance that are used in the future calculations.

5.5.3 Paper

The fiber content in the paper is the same through the whole drying section which was mentioned before. It is the moisture content that changes and the inlet fiber content in the pre-drying section is calculated as follow:

$$m_{fib,in} = w_{fib} \cdot b_p \cdot v_{mach} = 74.2 g/m^2 \cdot 6.64\ m \cdot \frac{915}{60} m/s = 7.68\ kg\ fiber/s$$

This is also the same value for the outlet fiber content.

For the after-drying section a dry layer of starch has been added and therefore gives a higher mass of dry material. The amount dry material in to the after drying is calculated before to:

$$m_{fib+starch,in} = m_{fib+starch,out} = 7.77\ kg/s$$

5.5.4 Steam-Condensate

The steam that comes in to the cylinders in the pre-dryer and after dryer section is both a middle-pressure steam and low-pressure steam. The low pressure steam is divided to the air-preheater and the steam cylinders in both the pre-dryer and after-drying section. From the software Metso DNA that controls the different parameters, the total flow for the steams (see Appendix E) in to the pre-drying section is observed as the values below shows:

$$G_{s,mp} = 6.70\ kg/s$$

$$G_{s,lp} = 4.30\ kg/s$$

The steam flows in to the after-drying section (see Appendix E) is also observed from the software Metso DNA as following values:

$$G_{s,lp} = 1.75 \; kg/s$$

$$G_{s,mp} = 0 \; kg/s$$

To calculate the amount steam that is needed for the air-preheaters, the temperature for the air streams before and after the air-preheater is measured. These temperatures are then used for calculating the enthalpy for the air caused from the increased temperature. Figure 19 shows the measured air flows (S1-S6) before and after the air-preheater for each air-preheater in the pre-drying section. The temperature of the air-flows before and after the air-preheaters is presented below in Table 15.

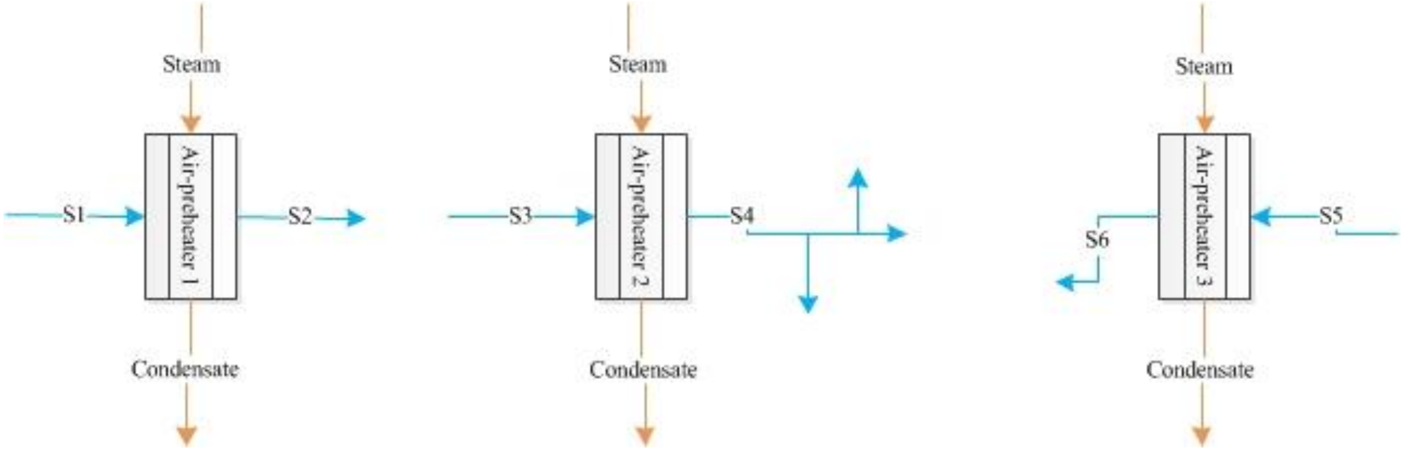

Figure 19-The measured points in and out for each air-preheater in the pre-drying section

Table 15-The measured parameters for the air-preheaters in the pre-drying section

Air-preheater	Dry temperature [^{0}C]		X [kg H_2O/kg dry air]	Air flow[kg/s]
	Before	After		
1	57	104	0.0275	14.78
2	58	98	0.0275	19.39
3	60	99	0.0275	4.88

With values from Table 15, the enthalpy for the air stream can be determined with formula 3.14 and are presented below in Table 16.

Table 16-Enthalpies for the air streams before and after the air-preheater

Air-preheater	**Enthalpy for air stream [kJ/kg]**	
	Before	**After**
1	128.9	178.6
2	130.5	171.6
3	132.3	173.0

The steam that goes in to the air-preheater is of a superheated low-pressure steam of 4 bar and the latent heat of evaporation for this steam is:

$\Delta H_{vap(4\,bar)} = 2134\ kJ/kg$ [42]

No blow through steam comes out from the air-preheater it means that all ingoing steam to the air-preheater comes out as condensate according equation 3.2. The blow-through term therefore disappears. The amount steam that is needed for each air-preheater is calculated with an energy balance as formula 5.7 below shows. The up-taken energy for the air is the same amount energy that leaves the steam for condensing.

$$G_c = G_s = \frac{G_{sup}(H_{aph,2} - H_{aph,1})}{\Delta H_{vap(4\,bar)}} \quad (5.7)$$

The results for the amount steam that is needed for the air-preheaters in the pre-drying section are presented in Table 17.

Table 17-Steam flow to each air-preheater in the pre-drying section

Air-preheater	**Steam flow $G_{steam,aph}$ [kg/s]**
1	0.34
2	0.37
3	0.09
	∑ 0.81

For the air-preheater in the after-drying section it is assumed that 50 % of the total steam in to the air-preheater is made up of flash steam [43]. There are no sensors that measure this flow. To calculate the total steam that goes in to the steam battery the condensate flow out is decided by measure the temperature of the air flow before and after the air-preheater. The observed results are presented in Table 18:

Table 18- The measured parameters for the air-preheater in the after-drying section

Air-preheater	Dry temperature [^{0}C]		X [kg H_2O/kg dry air]	Air flow[kg/s]
	Before	**After**		
1	53.4	75.2	0.013	9.2

The enthalpy before and after the air-preheater for the airflow is:

Table 19- The enthalpy for the air stream before and after the air-preheater in the after-drying section

Air-preheater	Enthalpy for air stream [kJ/kg]	
	Before	**After**
1	87.2	109.6

The condensate/steam flow for the air-preheater can now be calculated as following:

$$G_c = G_s = \frac{G_{supply\ air}(H_{aph,2} - H_{aph,1})}{\Delta H_{vap(4\ bar)}} = \frac{9.22(109.6 - 87.2)}{2134} = 0.097\ kg/s$$

This would give the flow for the flash steam to:

$$G_{flash} = 0.5 \cdot 0.097 = 0.05\ kg/s$$

The amount steam that goes to the cylinders in the pre-drying section can now be calculated as following:

$$G_{s,cyl,pre-dryer} = G_{s,mp} + G_{s,lp} - G_{s,aph} = 6.7\ kg/s + 4.3\ kg/s - 0.81 = 10.20\ kg/s$$

The amount steam that goes to the cylinders in the after-drying section is:

$$G_{s,cyl,after-dryer} = 0\ kg/s\ + 1.75\ kg/s\ - 0.097\ kg/s = 1.650\ kg/s$$

The condensate leaves the system from condensate tank 1 and 2 for the pre-dryer as can be seen in Figure 16. For the after-dryer the condensate leaves from condensate tank 5 and 6 shown in Figure 17. The condensate flow out from condensate tank 1 is calculated by measuring the pressure before and after the pump. Following pressures where observed:

$$p_t = 3.1 \cdot 10^5 Pa$$

$$p_s = 0.6 \cdot 10^5 Pa$$

This gives a pressure increase calculated with equation 4.8 to:

$$\Delta p_{pump} = 3.1 \cdot 10^5 Pa - 0.6 \cdot 10^5 Pa = 2.5 \cdot 10^5\ Pa$$

With formula 4.9 the head can be calculated to:

$$H_{pump} = \frac{2.5 \cdot 10^5\ Pa}{1000 \cdot 9.82} = 25\ m$$

With this value read in the pump curve for the pump at condensate tank 1with a wheel-diameter of 145 mm (see appendix F) the volume flow of the condensate is decided to:

$$F_{ct1} = 11.20\ m^3/h$$

Calculated to the mass flow with the density for water at 75 ºC ρ_{H2O}=975 kg/m^3 [40] the mass flow for the condensate is:

$$G_{ct1} = 11.20\ m^3/h \cdot 975.0\ \text{kg}/m^3 = 10.9\ \text{ton/h} = 3.03\ \text{kg}/s$$

For the pump at condensate tank 2 there were no openings to measure the pressures before and after the pump. Instead the power for the pump was measured and then this value was used in the pump curve to decide the volume flow.

The power for the engine is calculated with equation 4.13 and the current and voltage is measured to:

$$I = 4\ A$$

$$U = 510\ V$$

The power factor is $cos\varphi = 0.85$ and gives the power for the engine to:

$$P_{engine} = 4\ A \cdot 510\ V \cdot \sqrt{3} \cdot 0.85 = 3003\ W$$

With equation 4.14 this value can be transformed to the axial power for the pump with an efficiency of 0.85 for the engine.

$$P_{axial} = 3003\ W \cdot 0.85 = 2550\ kW$$

For the pump at condensate tank 2 with a wheel-diameter of 200 mm the volume flow for the condensate can be decided in the pump curve (see appendix F) to:

$$F_{ct2} = 16\ l/s = 16 \cdot 10^{-3}\ m^3/s$$

With the density ρ_{H2O}=958 kg/m^3 for water at 100 ºC gives a mass flow of:

$$G_{ct2} = 16 \cdot 10^{-3}\ m^3/s\ \cdot 958\ \text{kg}/m^3 = 15.3\ \text{kg/s}$$

Both of the condensate flow from condensate tank 5 and 6 were measured in the same way as for pump the pump at condensate tank 1.Where the pressures before and after the pump was measured. For the pump at condensate tank 5 following pressures where observed:

$$p_t = 4.5 \cdot 10^5 Pa$$

$$p_s = 2.0 \cdot 10^5 Pa$$

And this gives a pressure increase calculated with equation 4.8 to:

$$\Delta p_{pump} = 4.5 \cdot 10^5 Pa - 2.0 \cdot 10^5 Pa = 2.5 \cdot 10^5\ Pa$$

With formula 4.9 the head can be calculated to:

$H_{pump} = \frac{2.5 \cdot 10^5\, Pa}{1000 \cdot 9.82} = 25\ m$

With this value read in the pump curve for condensate tank 5 (Appendix F) with a wheel-diameter of 145 mm this gives the mass flow of the condensate to:

$G_{ct5} = 3.500\ kg/s$

For the pump at condensate tank 6 following pressures where observed:

$p_t = 4.0 \cdot 10^5 Pa$

$p_s = 2.0 \cdot 10^5 Pa$

This gives a pressure increase calculated with equation 4.8 to:

$\Delta p_{pump} = 4.0 \cdot 10^5 Pa - 2.0 \cdot 10^5 Pa = 2.0 \cdot 10^5\ Pa$

With formula 4.9 the head can be calculated to:

$H_{pump} = \frac{2.0 \cdot 10^5\, Pa}{1000 \cdot 9.82} = 20\ m$

With this value read in the pump curve for condensate tank 6 with a wheel-diameter of 145 mm this gives the mass flow of the condensate to:

$G_{ct6} = 5.80\ kg/s$

As can be seen the condensate flow out in both the pre-drying and after-drying section is much larger than the ingoing steam flow in to the systems. The steam-condensate balance is not fulfilled according to equation 3.2. The steam flows in to the cylinders are more reliable and therefore used in the future calculations instead of the condensate flows out from the condensate tanks. Therefore the condensate flow is adjusted in a way so they are not higher than the ingoing steam flows and still keeps the same percent of the total condensate flow. The percent of each condensate flow is calculated as following:

$$y_{cond} = \frac{G_{c,ctj}}{Total\ condensate\ from\ tanks} \cdot 100 - Percent\ condensate\ flow \qquad (5.8)$$

The percent of the condensate flow out from tank 1 in the pre-drying section is:

$y_{cond,ct1} = \frac{3.03\ \mathrm{kg/s}}{3.03\ \mathrm{kg/s} + 15.3\ \mathrm{kg/s}} \cdot 100 = 16.7\ \%$

The percent for the rest of the condensate flow are presented in Table 20.

Table 20-The percent for the condensate flows from the condensate tanks

Condensate flow-tanks	Pre-dryer[%]	After-dryer[%]
1	16.7	-
2	83.3	-
5	-	37.6
6	-	62.4

The blow through steam out from condensate tank 1 and 5 cross the boundary line in Figure 17 respectively Figure 18. These blow-through steams are calculated by measuring the cold water flow and temperature difference for the cold water in to the surface condenser. The results for the pre-drying section and after drying section are presented in Table 21.

Table 21-The measured parameters for the cold water flow in the surface condenser

Surface condenser	Cold water flow [m³/s]	Temperature in [°C]	Temperature out [°C]
Pre-dryer	$3.3 \cdot 10^{-3}$	30	30
After-dryer	$4.6 \cdot 10^{-3}$	33	46

With the values in Table 21 the flow for the blow through steam is calculated with an energy balance according to equation 5.9.

$$Q_{bt} = Q_{H_2O(l)} = m_{bt} \cdot \Delta H_{vap} = m_{H_2O(l)} \cdot C_{p,H_2O(l)} \cdot \Delta T \qquad (5.9)$$

The cold water flow has a pressure temperature of 30 ºC and this gives a density of:

$\rho_{H_2O} = 996\ kg/m^3$ [40]

The power that is provided from the blow through steam to the cold water can now be calculated with equation 5.9 as following:

$$Q_{H_2O(l)} = 3.3 \cdot 10^{-3} m^3/s \cdot 996\ kg/m^3 \cdot 4.18\ \frac{kJ}{kg^{\circ}C} \cdot (30 - 30)^{\circ}C = 0\ kW \text{ (Pre-dryer)}$$

$$Q_{H_2O(l)} = 4.6 \cdot 10^{-3} m^3/s \cdot 996\ kg/m^3 \cdot 4.18\ \frac{kJ}{kg^{\circ}C} \cdot (46 - 33)^{\circ}C = 249.0\ kW \text{ (After-dryer)}$$

The provided power from the blow through steam to the cold water flow in the pre-drying section is zero. This means that there is no blow through steam in to the surface condenser. This is because during normal operations there is no excess of heat, all heat is used for the system but during a paper break there is a lot of excess heat. To utilize this heat, the steam goes to the surface condenser to warm water that can be used for other purposes. But for the surface condenser in the after-drying section, the cold water flow is heated from the blow through steam.

The steam-pressure for the blow through steam read in Metso DNA is 59 kPa. This steam-pressure gives the latent heat of evaporation to $\Delta H_{vap,\,bt}$=2293 kJ/kg [42]. The mass flow for the blow through steam in the after-drying section can now be calculated with equation 5.9 since the power provided from the steam to the cold water flow is known.

$$m_{bt} = \frac{Q_{bt}}{\Delta H_{vap}} = \frac{249.0\ kW}{2293.2\ kJ/kg} = 0.11\ kg/s$$

Now when the blow through steam is known the steam and condensate balance according to equation 3.2 can be fulfilled. The condensate is distributed percental on the steam that doesn't condense. This give the condensate flow out form condensate tank 1 to:

$$G_{c,ct1} = y_{cond,ct1} \cdot (\sum G_{steam,cylinders,pre} - \sum G_{blow\ through\ steam,pre}) = 0.167 \cdot (10.2\ kg/s - 0\ kg/s) = 1.69\ kg/s$$

The rest of the new condensate flows from the condensate tanks are calculated in the same way and are presented in Table 22.

Table 22-The adjusted condensate flows for the condensate tanks

Condensate flow-tanks	Pre-dryer[kg/s]	After-dryer[kg/s]
1	1.69	-
2	8.45	-
5	-	0.58
6	-	0.97
	$\sum$ 10.2	$\sum$ 1.55

It is the condensate flows presented in Table 22 that is used in the future calculations.

5.6 Energy balances for the drying section

The masses for the different subjects that cross the boundary line are now known and energy balances can be set up. The process is shown in Figure 17 for the pre-dryer and Figure 18 for the after-dryer.

5.6.1 Air

Equation 3.7 is used for calculating the power for the ingoing air in to the system while equation 3.8 is used to calculate the power for the outgoing air from the system. The enthalpies for the airflows are calculated with equation 3.14.

For the measured point 1 in the pre-drying section with values from Table 3 and Table 4 the enthalpy is calculated as follows:

$$H_{exh1} = (1.0 + 1.88 \cdot 0.19) \cdot 87.7 + 2500 \cdot 0.19 = 594.07\ kJ/kg\ dry\ air$$

With mass flow of air in Table 14 this would give a power for point 1 to:

$$Q_{exh1} = 4.8 \cdot 594.07 = 2857.1\ kW$$

The enthalpies and the power for the rest of the air-streams are calculated in the same way as for point 1 and the results are presented in Table 23.

Table 23-The enthalpies and the powers for the air flows in the pre-dryer

Measure point	Enthalpy H_i [kJ/kg]	Power Q_i [kW]
1	594.07	2857.1
2	364.38	3192.6
3	503.68	18955.0
4	202.08	9147.3
5	243.37	3277.4
6	168.14	536.47
7	168.14	459.74
9	261.46	1594.1
11	183.03	2704.7
12	115.56	1707.7
13	118.63	2300.0
15	190.06	1158.8

The power for air stream 14 that leaves the system after it has been heat exchanging with air stream 13 (See Figure 13) is unknown. The temperature for stream 14 that comes in to the heat exchanger needs to be calculated first before the power that leaves the system with air stream 14 can be decided. This is calculated with mass- and energy balances that are set up according to Figure 20.

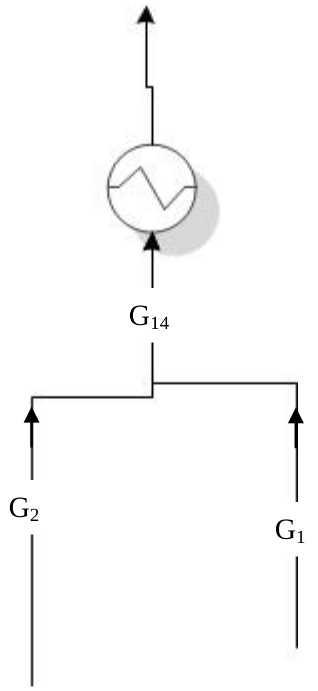

Figure 20- Air stream 1 and 2 mixed together to one new stream

The mass- and energy balances are as follows:

$$G_1 + G_2 = G_{14} - Total\ mass\ balance \qquad (5.10)$$

$$G_1 \cdot X_1 + G_2 \cdot X_2 = G_{14} \cdot X_{14} - Water\ balance \qquad (5.11)$$

$$G_1 \cdot H_1 + G_2 \cdot H_2 = G_{14} \cdot H_{14} - Energy\ balance \qquad (5.12)$$

With equation 5.10 and mass flows from Table 14 the new mass flow of air is calculated.

$G_{14} = 4.8\ kg/s\ + 8.8\ kg/s\ = 13.6\ kg/s$

And with equation 5.11 the air humidity for the mixed stream is calculated to:

$X_{14} = \frac{G_1 \cdot X_1 + G_2 \cdot X_2}{G_{14}} = \frac{4.8 \cdot 0.19 + 8.8 \cdot 0.11}{13.6} = 0.138\ kg\ H_2O/kg\ air$

The enthalpies for air stream 1 and 2 are calculated with equation 3.14 and are presented in Table 23. The enthalpy for air stream 14 can now be calculated with the energy balance equation 5.12

$H_{14} = \frac{G_1 \cdot H_1 + G_2 \cdot H_2}{G_{14}} = \frac{4.8 \cdot 594.1 + 8.8 \cdot 364.4}{13.6} = 445.9\ kJ/kg$

With the enthalpy and air humidity for air stream 14 inserted in equation 3.14 the temperature is calculated as followed:

$H_{14} = (1 + 1.88 \cdot X_{14}) \cdot T_{14} + 2500 \cdot X_{14} \rightarrow$
$T_{14,1} = 79.3\ ^0C$

To calculate the outgoing temperature of stream 14 from the heat exchanger a new energy balance for stream 13 and 14 over the heat exchanger is set up according to Figure 21:

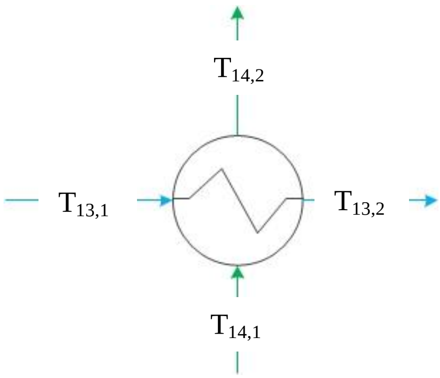

Figure 21- Heat exchanger for air stream 13 and 14 in the pre-drying section

The power that leaves stream 14 is equal to the up-taken energy to stream 13 and the energy balance can therefore be set up as follows:

$$Q_{13,2} - Q_{13,1} = Q_{14,1} - Q_{14,2} \qquad (5.13)$$

With mass flows from Table 8 and Table 14 and enthalpies from Table 23 and Table 16 the up taken power and the power that leaves can be calculated as follows:

$Q_{13,2} - Q_{13,1} = (13.5 + 3.2 + 2.7)\ kg/s \cdot (130.5\ kJ/kg - 118.6\ kJ/kg) = 231.3\ kW =$
$Q_{14,1} - Q_{14,2}$

The ingoing power to the heat exchanger from stream 14 is calculated as follows

$Q_{14,1} = (4.8 + 8.8)\ kg/s \cdot 445.8\ kJ/kg = 6050\ kW$

The power for the streams involved in the heat exchanger are known except $Q_{14,2}$ this power can be calculated with the known values inserted in equation 5.13 which gives following:

$Q_{14,2} = 6049.7\ kJ/s - 231.3\ \text{kJ/s} = 5819\ \text{kW}$

This power out for stream 14 corresponds to a temperature calculated to:

$5819 = (4.8 + 8.8)\ kg/s \cdot \left((1 + 1.88 \cdot 0.14) \cdot T_{14,2} + 2500 \cdot 0.14\right)$

$\rightarrow T_{14,2} = 65.8^{\circ}C$

The enthalpies and the power for the air flows in the after-drying section are set up and calculated in the same way as for the pre-drying section were the results are presented in Table 24.

Table 24-The enthalpies and power for the air flows in the after dryer

Measure point	Enthalpy H [kJ/kg]	Power Q [kW]
3	272.66	8071.2
4	140.64	4648.0
6	370.94	3312.9
7	137.95	1271.7

The power for the leakage air is calculated with equation 3.9 and the enthalpy is calculated in same way as before with equation 3.13. It is assumed that the air humidity of the leakage air is $X_{leak} = 0.015\ \frac{kg\ water}{kg\ air}$ and the temperature is assumed to be $T_{leak} = 20\ ^{\circ}C$ in both the pre-drying section and after-drying section. This would give the enthalpy of the leakage air to:

$H_{leak} = (1.0 + 1.88 \cdot 0.015) \cdot 20 + 2500 \cdot 0.015 = 58.06\ kJ/kg\ dry\ air$

The power the leakage air in the pre-drying section is now calculated to:

$Q_{leak} = 57.2\ kg\ dry\ air/s\ \cdot 58.06\ kJ/kg\ dry\ air = 3\ 321\ kW$

The total inlet power from the air in the pre-drying section is calculated to:

$\sum Q_{air,in\ ,pre-dryer} = Q_{12} + Q_{13} + Q_{15} + Q_{leak} = 8\ 488 kW$

The power for the leakage air in the after-drying section is calculated to:

$Q_{leak} = 44.5\ kg\ dry\ air/s\ \cdot 58.06\ kJ/kg\ dry\ air = 2\ 584\ kW$

This would give the total power from the air in to the after-drying section to:

$\sum Q_{air,in\ ,after-dryer} = Q_6 + Q_7 + Q_{leak} = 7\ 169\ kW$

The total power for the outlet airstreams in the pre-drying section is calculated to:

$\sum Q_{exh,pre-dryer} = Q_3 + Q_4 + Q_{14} = 34\ 150$ kW

The total power for the outlet airstreams in the after-drying section is calculated to:

$\sum Q_{exh,after-dryer} = Q_3 + Q_4 = 12\ 720$ kW

5.6.2 Paper

The power for the paper web in and out is calculated with formulas 3.10 and 3.11 and the inlet water and outlet water in the paper web for the pre-drying section needs to be known before the power can be determined.

The amount inlet and outlet water in the paper web in pre-drying section is calculated as follows:

$$m_{H_2O(l),in} = 6.79\ m \cdot \frac{915}{60}\ m/s \cdot\ 74.2\ g/m^2 \cdot \frac{53}{47} = 8.67\ kg/s$$

$$m_{H_2O(l),out} = 6.79\ m \cdot \frac{915}{60}\ m/s \cdot\ 74.2\ g/m^2 \cdot \frac{2}{98} = 0.157\ kg/s$$

The measured mean temperature for the paper web before it goes in to the pre-dryer is 41.5 ^{0}C and the outlet temperature for the paper web out from the pre-dryer is 92.4 ^{0}C. This was measured with an IR-camera.

The temperature and amount inlet water inserted in equation 3.10 gives the inlet power for the paper web to:

$$Q_{p,in} = \left(7.68\ kg/s \cdot 1.42\ \frac{kJ}{kg\ fiber,^0C} + 8.67\ kg/s \cdot 4.18\ \frac{kJ}{kg\ H_2O,^0C}\right) \cdot 41.5\ ^\circ C = 1949\ kW$$

The same calculation was done for the outlet power of the paper web with equation 3.11 and this gives the power to:

$$Q_{p,out} = \left(7.68\ kg\ fiber/s \cdot 1.42\ \frac{kJ}{kg\ fiber,^0C} + 0.157\ kg\ H_2O/s \cdot 4.18\ \frac{kJ}{kg\ H_2O,^0C}\right) \cdot 92.5\ ^\circ C = 955\ kW$$

For the after-drying section the inlet water and outlet water for the paper web are:

$$m_{H_2O(l),in} = 6.64 m \cdot \frac{915}{60}\ m/s \cdot\ 76.7\ g/m^2 \cdot \frac{20}{80} = 1.94\ kg/s$$

$$m_{H_2O(l),out} = 6.64 m \cdot \frac{915}{60}\ m/s \cdot\ 74.2\ g/m^2 \cdot \frac{4.1}{95.9} = 0.33\ kg/s$$

The measured mean temperature for the paper web in the after-drying section was measured in the same way as for the pre-drying section and following temperatures were observed:

$$T_{p,in} = 61.5\ ^\circ C$$

$$T_{p,out} = 83.8\ ^\circ C$$

The power for the paper web in to the after-dryer is calculated with formula 3.10 and gives following:

$$Q_{p,in} = \left(7.77\ kg/s \cdot 1.26\ \frac{kJ}{kg\ fiber,^0C} + 1.94\ kg/s \cdot 4.18\ \frac{kJ}{kg\ H_2O,^0C}\right) \cdot 61.5^0C = 1100.4\ kW$$

The power that comes out with the paper web is calculated with equation 3.11 to:

$$Q_{p,out} = \left(7.77\ kg/s \cdot 1.26\ \frac{kJ}{kg\ fiber,^0C} + 0.33\ kg/s \cdot 4.18\ \frac{kJ}{kg\ H_2O,^0C}\right) \cdot 83.8^0C = 936.2\ kW$$

5.6.3 Steam-Condensate

The steam used in the cylinders is an overheated middle- and low pressure steam of 13 bar and 200 ^{0}C respective 4 bar and 150 ^{0}C [8]. The power for the condensate is calculated with equation 3.12. With a pressure of 4 bar for the low pressure steam the condensation temperature is $T_{cond} = 144$°C [42]. This low pressure steam goes in to the air-preheater and with values of the condensate flow in Table 17; the power for the condensate out from air-preheater 1 in the pre-drying section is calculated as follows:

$$C_{p,H_2O(l)} = 4.18\ \frac{kJ}{kg\cdot^0C}$$

$$Q_{c,aph,1} = 0.34\ \text{kg/s} \cdot 4.18\ \frac{kJ}{kg\cdot^0C} \cdot 144^\circ C = 207.3\ kW$$

The power for the rest of the condensate flow out from the air-preheaters in the pre-drying section is presented in Table 25.

Table 25-The power for the condensate flows out from air preheater in the pre-dryer

Air-preheater	Power for condensate $Q_{c,aph,i}$ [kJ/s]
1	207.34
2	224.73
3	55.95
	∑ 488.02

The same calculation was done for the condensate flow out from the air-preheater in the after-drying section was following power was calculated for the condensate:

$$Q_{c,aph,after-drying} = 0.097\ kg/s \cdot 4.18\ \frac{kJ}{kg^\circ C} \cdot 144\ ^\circ C = 58.20\ kW$$

The condensate that comes from the steam-cylinders leaves the system from condensate tanks and the power for these condensates is also calculated with equation 3.12. Since there is a given steam-pressure in the condensate tanks from the software Metso DNA (see appendix G) and the condensate temperature is calculated with Antoine's equation 4.15. The results are presented in Table 26.

Table 26-The steam pressures and condensation temperatures for the condensate tanks

Condensate tank	Steam-pressure [mbar]	Condensation temperature [^{0}C]
1	370	74
2	1350	108
5	690	89.6
6	1740	115.8

The energy flows for the condensates are calculated with equation 3.12 with values from Table 22. For condensate tank 1 where the condensate flow is 1.69 kg/s the power is:

$Q_{ct1} = 1.69\ kg/s \cdot 4.18\ \frac{kJ}{kg \cdot {}^0C} \cdot 74^0C = 522.8\ kW$

The same calculation was made for the rest of the condensate flows from the condensate tanks and the results are seen in Table 27.

Table 27-The power for the condensate flows from the condensate tanks

Condensate tank	Power for condensate Q_{ctj} [kW]
1	522.8
2	3 814.7
5	218.9
6	468.8

It is assumed that all blow-through steam is condensed in the surface condenser. The condensate flow in to condensate tank 5 is therefore equal to the flow of the blow through steam. The temperature for blow-through steam is calculated with Antoine's equation 4.15 and gives a temperature of T=85.6 ºC where the steam-pressure is 59kPa. This is also the temperature for the condensate flow that goes in to condensate tank 5. The power for the condensate flow from the surface condenser in to condensate tank 5 is calculated with equation 3.12 to:

$Q_{c,surf} = 0.10\ kg/s \cdot 4.18\ \frac{kJ}{kg^{\circ}C} \cdot 85.6\ {}^{\circ}C = 35.90\ kW$

The flash steam that goes in to the air-preheater in the after-drying section is assumed to have a pressure of 20 kPa and a temperature of 100 ºC and are therefore an overheated steam [43]. The enthalpies for the overheated steams are found from the Mollier chart for steam [44] and are presented in Table 28.

Table 28-The enthalpies for the different steams

Steam	Enthalpy [kJ/kg]
Low-pressure	2775
Middle-pressure	2810
Flash	2690

With the enthalpies that are presented in Table 28 together with equation 3.6, the inlet power from the steams in to the pre-drying section and after-drying section can be calculated.

$Q_{s,lp} = 4.30\ kg/s \cdot 2775\ kJ/kg = 11\ 950\ kW$- Pre-dryer

$Q_{s,mp} = 6.70\ kg/s \cdot 2810\ kJ/kg = 18\ 730\ kW$ – Pre-dryer

$Q_{s,lp} = 1.75\ kg/s \cdot 2775\ kJ/kg = 4\ 856\ kW$ –After- dryer

$Q_{s,mp} = 0\ kg/s \cdot 2810\ kJ/kg = 0\ kW$ – After-dryer

$Q_{flash} = 0.05\ kg/s\ \cdot 2690$ kJ/kg $= 129.9\ kW$ –After-dryer

The power for the steam in the pre-drying section and after-drying section is known and the amount evaporated water. The specific heat consumption (ΔH_{vap}) could be calculated. The specific heat consumption for the pre-drying section is:

$$\Delta H_{vap} = \frac{11948\ kW + 19514\ kW}{8.5\ kg/s} = 3701.4\ \frac{kW \cdot s}{kg} = 3701.4\ kJ/kg$$

For the after-drying section the specific heat consumption is calculated to:

$$\Delta H_{vap} = \frac{4856.3\ kW + 129.95\ kW}{1.6\ kg/s} = 3116.4\ \frac{kW \cdot s}{kg} = 3116.4\ kJ/kg$$

Comparing these two values to the specific heat consumption $\Delta H_{vap}(70^{\circ}C) = 2333\ kJ/kg$ [42] shows that there is potential to decrease the steam usage. This is due to the specific heat consumption for both the pre-dryer and the after-dryer is higher than the theoretical value.

5.7 Total balance for the drying section

The total mass balance with all components for the pre-drying section according to Figure 17 is set up to:

$$\sum G_{sup} + \sum G_s + m_{fib,in} + \sum G_{leak} + \sum G_{water,in} = \sum G_{exh} + \sum G_{c,ctj} + \sum G_{c,aph} + \sum G_{water,out} + m_{fib,out} + G_{bt} \quad (5.14)$$

The sum of the inlet masses (left hand side of equation 5.14) is:

$39\ kg\ dry\ air/s + 6.7\ kg\ steam/s + 4.3\ kg\ steam/s + 57.2\ kg\ dry\ air/s + 10.95\ kg\ water/s + 7.7\ kg\ fiber/s = 125.9\ kg/s$

The sum of the outgoing masses (right hand side of equation 5.14) is:

$96.2\ kg\ dry\ air/s + 1.69$ kg $H_2O/s\ + 8.45$ kg $H_2O/s + 0.34$ kg $H_2O/s + 0.37$ kg $H_2O/s + 0.09$ kg $H_2O/s + 10.95\ kg\ H_2O/s + 7.7\ kg\ fiber/s = 125.8\ kg/s$

The inlet- and outlet masses in the pre-drying section are almost equal to each other. If equal to each other it means that everything that goes in is utilized and no losses of mass are observed. Here the outlet mass is 0.1 kg less than the inlet mass and therefore 0.1 kg is seen like a loss.

The total mass balance with all components for the after-drying section according to Figure 18 is set up in the same way as equation 5.14. The sum of the inlet masses is:

$$18.14\ kg\ air/s\ + 3.63\ kg\ water/s + 7.77\ kg\ fiber/s + 1.75\ kg\ steam/s + 0.05\ kg\ steam/s\ + 44.5\ kg\ air/s\ + 0.1\ kg\ water/s = 75.9\ kg/s$$

The sum of the outlet masses is:

$$62.65\ kg\ dry\ air/s + 3.63\ kg\ water/s + 7.77\ kg\ fiber/s + 0.1\ kg\ condensate/s + 0.58\ \text{kg condensate/s} + 0.97\ condensate\ /s + 0.1\ kg\ steam/s = 75.8\ kg/s$$

For the inlet masses and outlet masses in the after-drying section it is seen that the outlet masses is 0.1 kg less than the inlet masses. This means that 0.1 kg is not utilized and is seen like a loss.

The inlet and outgoing masses for the pre-drying section and after-drying section are sum up and presented in a mass flow diagram. This would give an overview of how every stream are related to each other. For the mass flows in the pre-drying section the diagram is shown in Figure 22 and the mass flows for the after-drying section is presented in Figure 23.

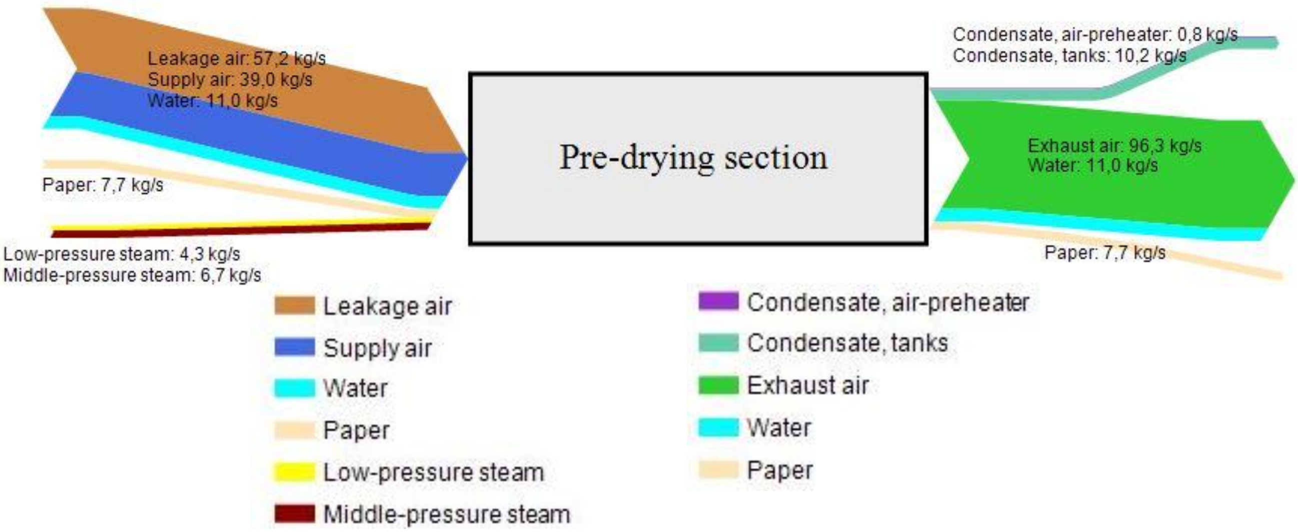

Figure 22-The mass flows in the pre-drying section

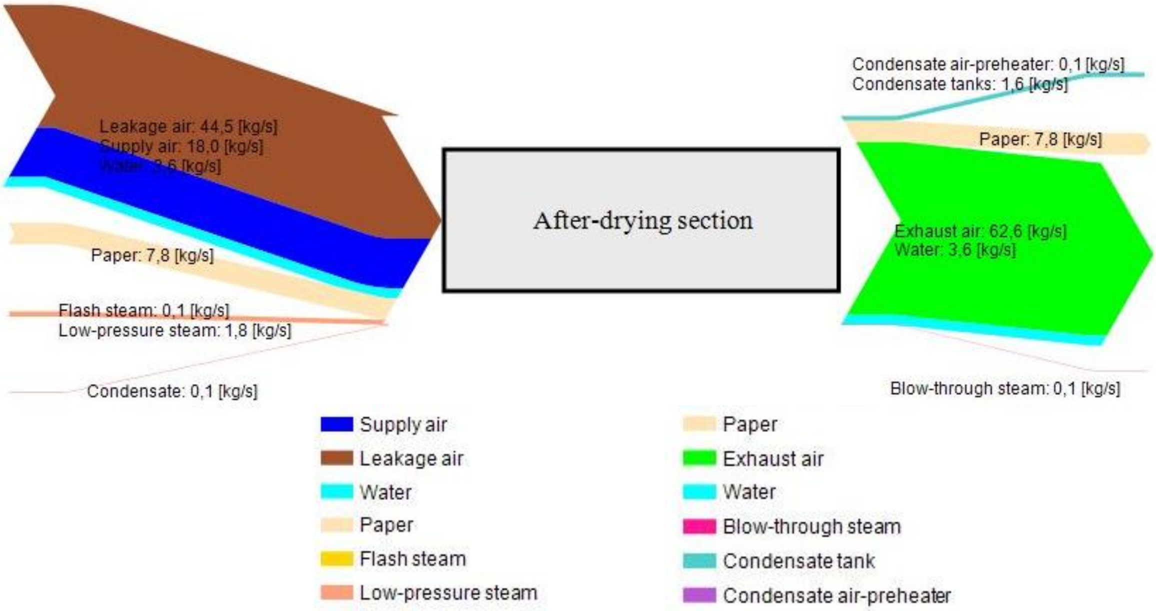

Figure 23-The mass flows in the after-drying section

The total energy balance for the system is set up as equation 5.15.

$$\sum Q_{sup} + \sum Q_{leak} + \sum Q_s + Q_{p,in} = \\ = \sum Q_{exh} + \sum Q_{c,aph} + \sum Q_{c,ctj} + Q_{p,out} + Q_{bt} \quad (5.15)$$

The left hand side of equation 5.15 makes up the inlet power and for the pre-drying section they are calculated to:

$8\ 488.1\ kW + 11\ 948\ kW + 18\ 733\ kW + 1\ 949.6\ kW = 41\ 120\ kW$

The right hand side of the equation makes up the outlet power and for the pre-drying section the outlet power is calculated to:

$34\ 153\ \text{kW} + 207.34\ \text{kW} + 224.73\ \text{kW} + 55.95\ \text{kW} + 522.8\ kW + 3\ 814.7\ kW + 955.45\ kW = 39\ 930\ kW$

The outlet power in the pre-drying section is 1185 kW less than the inlet power. This means that not all the ingoing power is utilized and disappears in form of heat or other losses.

The total inlet power for the after-drying section is calculated in the same way as for the pre-drying section. The total inlet power is:

$7168.5\ kW\ + 4856.3\ kW + 129.95\ kW + 1100.4\ kW + 35.9\ kW = 13\ 290\ kW$

The total power out is:

$12\ 719\ \text{kW} + 936.2\ kW + 58.2\ kW + 218.9\ kW + 468.8\ kW + 230.0\ kW\ = 14\ 680\ kW$

The outlet power in the after-drying section is 1385 kW higher than the inlet power. This phenomenon is not possible since energy can’t be created according to first law of thermodynamic [26].

The calculated energy flow for each component in the pre-drying section and the after-drying section are presented in a Sankey diagram. It is seen how large the energy flows are related to each other.

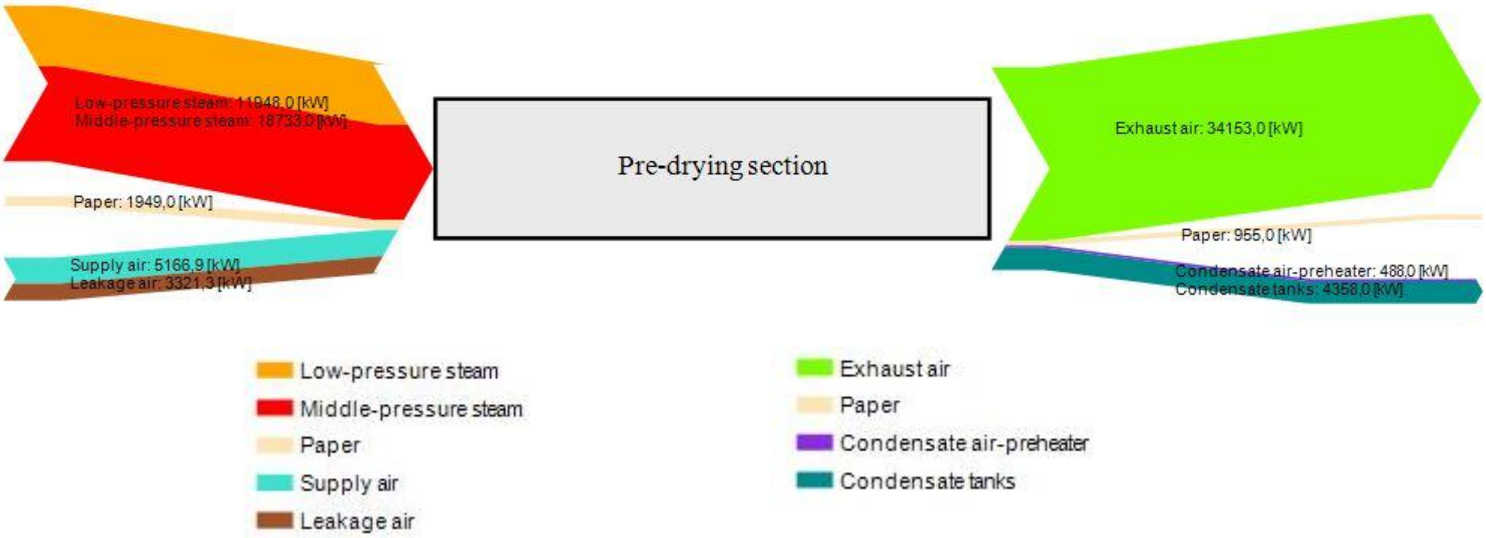

Figure 24-The Sankey diagram for the pre-drying section

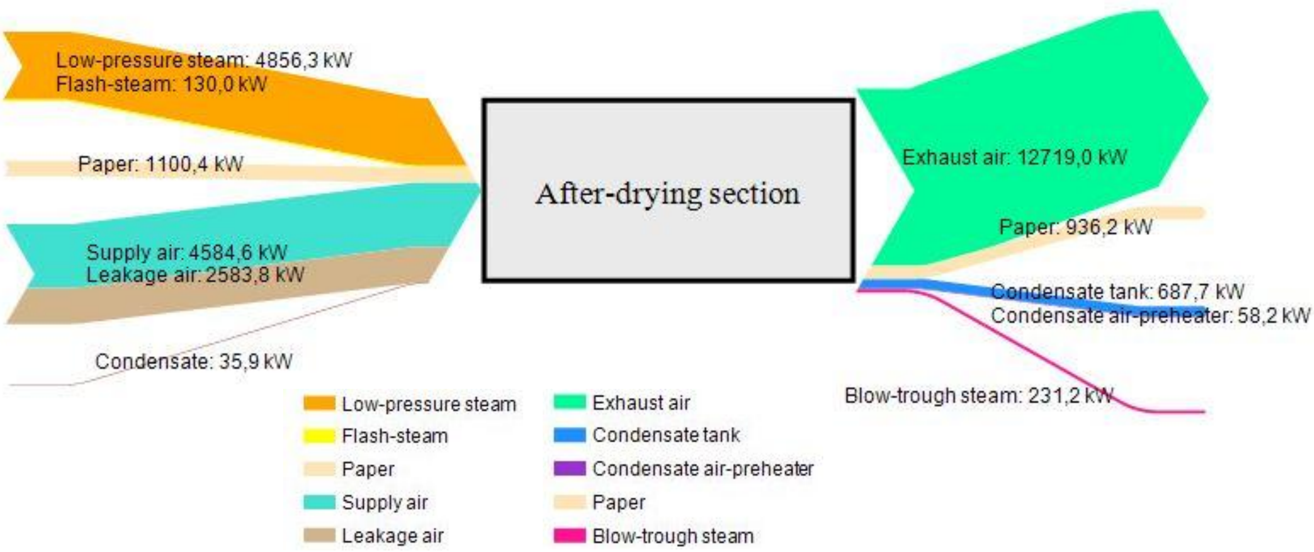

Figure 25-The Sankey diagram for the after-drying section

The yearly power consumption for the pre-dryer and after dryer calculated is calculated with equation 4.1 where it is approximated that the operation time is 7560 h/year [8]. This will give the yearly energy usage in the drying sections to:

$$Q_{year,pre-dryer} = 41\ 120\ kW \cdot 7560\ hours/year = 310.9\ GWh/year$$

$$Q_{year,afterdryer} = 13\ 290\ kW \cdot 7560\ hours/year = \ 100.5\ GWh/year$$

5.8 Economical analysis

The prices for producing the steam in the drying section are presented in Table 29 [8].

Table 29-Producing prices for the steam

Steam	SEK/MWh	SEK/ton
Low-pressure	195	115
Middle-pressure	205	125

The leakage air in to the drying hood takes needs to be heated up. This energy is then added in form of steam. The leakage air provides to a cost since more steam needs to be added and this cost is calculated for both the pre-drying section and the after-drying section.

The leakage air in the pre-drying section is heated to 80 ºC, which is the mean temperature of the outgoing exhaust air streams 1 and 2. The power needed for the leakage air-heating section is dependent on the enthalpy difference caused by the heating. The enthalpy is calculated with equation 3.9.

$$H_{leak,80^{\circ}C} = (1 + 1.88 \cdot 0.015) \cdot 80 + 2500 \cdot 0.015 = 119.8\ kJ/kg$$

The enthalpy for the leakage air at 20 ºC is calculated before to 58.0 kJ/kg and the power for leakage air heating is now calculated to:

$$Q_{leak} = 57.2 \cdot (119.8 - 58.0) = 3529\ kW$$

The leakage air in to the pre-drying section is approximately 60 % of the total ingoing air. Reducing the leakage air to 30 % (half of the leakage air) the power for heating the leakage air is reduced to the half and is $Q_{leak,red} = 1764.4\ kJ/kg$. This is equal to the power that comes from both the middle-pressure steam and the low-pressure steam and can be written to:

$$Q_s = x_{lp} \cdot G_s \cdot H_{lp} + x_{mp} \cdot G_s \cdot H_{mp} \tag{5.16}$$

Where the percentage for each steam is calculated with equation 5.17-5.18

$$x_{lp} = \frac{Amount\ low\ pressure\ steam}{Total\ amount\ steam} \tag{5.17}$$

$$x_{mp} = \frac{Amount\ middle\ pressure\ steam}{Total\ amount\ steam} \tag{5.18}$$

The percentage of each steam in the pre-drying section is calculated to:

$$x_{lp} = \frac{6.7\ kg/s}{6.7\ kg/s + 4.3\ kg/s} = 0.60$$

$$x_{mp} = \frac{4.3\ kg/s}{6.7\ kg/s + 4.3\ kg/s} = 0.40$$

The total amount steam that is saved in the pre-drying section when reducing the amount leakage air to the half can now be calculated with equation 5.16 to:

$$1794.4 = 0.6 \cdot G_s \cdot 2775 + 0.4 \cdot G_s \cdot 2810 \rightarrow G_s = 0.63\ kg/s$$

$$\rightarrow G_{s,lp} = 0.63\ kg/s \cdot 0.6 = 0.38\ kg/s = 1.37\ ton/h$$

$$\rightarrow G_{s,mp} = 0.63\ kg/s \cdot 0.4 = 0.25\ kg/s = 0.9\ ton/h$$

The yearly economical savings that can be made from this steam saving is calculated with the steam prices in Table 29. The same approximation of the operation time as before is made.

$$C_{lp} = 1.37\ ton/h \cdot 7560.3\ h/year \cdot 115\ SEK/ton = 1.20\ millions\ SEK/year$$

$$C_{mp} = 0.9\ ton/h \cdot 7560.3\ h/year \cdot 125\ SEK/ton = 0.84\ millions\ SEK/year$$

$$C_{tot,pre\ dryer} = 1.20\ millions\ SEK/year + 0.84\ millions\ SEK/year$$
$$= 2.05\ millions\ SEK/year$$

In the after-drying section the amount leakage air is 44.5 kg/s and corresponds to a percent of 71.2 % of the total ingoing air. This amount leakage air is reduced to 30 % of the total ingoing air and the new leakage air flow is now determined to:

$$G_{s,red} = 0.3 \cdot 62.7\ kg/s = 18.8\ kg/s$$

In the after-drying section the mean outlet temperature for the exhaust air is 67.7 ºC. This corresponds to measure point 3 in the after drying section. It is this temperature that the leakage air is heated to. The same assumptions for the leakage air in to the after-drying section are made as before. The enthalpy for the leakage air after heating is calculated to:

$$H_{leak,67.7^{\circ}C} = (1 + 1.88 \cdot 0.015) \cdot 67.7 + 2500 \cdot 0.015 = 107.1\ kJ/kg$$

The power needed for heating the leakage air before reducing is:

$$Q_{leak} = 44.5 \cdot (107.1 - 58.0) = 2183\ kW$$

When reducing the leakage air to 30 % of the total ingoing air the reduced power for heating this amount air is:

$$Q_{leak,red} = 18.8 \cdot (107.1 - 58.0) = 921.1\ kW$$

When reducing the leakage air to 30 % of the ingoing air it will give a power decreasing of:

$$\Delta Q_{leak} = 2182.5\ kJ/s - 921.1\ kJ/s\ = 1261\ kW$$

In the after drying section, only low-pressure steam and flash steam is used. It is only the low-pressure steam that contributes to increased cost and goes in to the drying hood. The power needed for heating the leakage air is only dependent on the amount low-pressure steam.

The total amount steam that is saved in the after-drying section when reducing the amount leakage air to 30 % of the ingoing air is calculated with equation 5.16 as follows:

$$1261.4\frac{kJ}{s} = G_{s,lp}\frac{kg}{s}\cdot 2775\frac{kJ}{kg} \rightarrow G_{s,lp} = 0.45\ kg/s = 1.63\ ton/h$$

The yearly economical savings that can be made from this steam saving is calculated in the same way as for the pre-drying section which gives following results:

$$C_{lp} = 1.630\ ton/h \cdot 7560\ h/year \cdot 115\ SEK/ton = 1.42\ millions\ SEK/year$$

The total economical savings made by the steam in the whole drying section caused by reducing the leakage air is calculated to:

$$C_{tot,drying\ section} = 2.05\ millions\ SEK/year + 1.42\ millions\ SEK/year = 3.5\ millions\ SEK/year$$

To reduce the leakage air in the pre-drying section it could be done by installing an extra supply air fan. In this way the amount of supply air is increased. The investment cost for this supply air fan is 600 000 SEK and the yearly operating cost is 40 000 SEK [43]. This would give a yearly saving of 2.05 million SEK. With the pay-back method the payback time for installing the extra supply air fan can be decided with equation 5.19 [45] as follows:

$$PB = \frac{Gi}{a} \tag{5.19}$$

$$PB_{supply\ air\ fan} = \frac{600\ 000\ SEK}{2.05\ millions\ SEK - 40\ 000\ SEK} = 0.30\ years = 3.6\ months$$

The payback time for installing the extra air fan is approximately 4 months and is therefore seem like a profitable investment because the payback time is less than a year.

For the after-drying section no zero level controlling was installed. With installing, the leakage air could be reduced since the supply air is controlled for the zero level. The investment cost for installing the zero level control is 80 000 SEK [43]. With equation 5.19 the payback time can be calculated for installing the zero level control in the after-drying section to:

$$PB_{zero\ level\ control} = \frac{80\ 000\ SEK}{1.63\ millions\ SEK} = 0.05\ years = 0.5\ months$$

The payback time for installing the zero level control is less than a year and is also here seems to a profitable investment.

The high leakage air is caused by overpressure in the drying hood. Another way to reduce the leakage air is therefore to decrease the exhaust air. This will even out the pressure difference inside the hood and give less overpressure. In this way less leakage air is not allowed to be blown in to the drying hood.

6. Discussion and conclusion

In this chapter the observed results are discussed together with the measurements uncertainty that predict how reliable the results are and what conclusions could be made from the results in this master thesis.

6.1 Measurement uncertainty

As can be seen from the measurement of airflows it gives a high mass flow of leakage air in both the pre-drying section and the after-drying section. This result is based on the adjusted values from the measured values. Only the measured values gave that the amount air in to the system was higher than the amount out from the system. This is not correct for the air balance, since what comes in has to come out. The same result was observed for the water balance; the amount water in to the system was higher than the amount water out from the system. The quality of the paper is affected by the amount water that evaporates from the paper web. Therefore the amount water in to the system seemed more reliable because this parameter is controlled to give the paper its right properties. This was the reason why the outlet airflows were adjusted in a way that they will be equal to the inlet water.

Source of errors that has affected the results for the measurements of air-flows is the openings for the flow channels, which is regulated with damper, haven't been taken in consideration in the calculations. In the calculations the area for the different channels was used which correspond to full opening of the damper. The cross-sectional area for the air could have been less. Also when measuring the pressure in the flow channels with the Prandtl tube it is seen that the air is pulsing, this gave varying pressures all the time. This phenomenon makes it hard to determine the pressure and also the pressure in the channels were measured in the middle. The air pressure is varying in the whole channel and therefore the mean value when measuring at different places in the channel would give a better accuracy. Another source of error is that some measurements were made close to the air fan, or close to a curvature, which is not recommended since this will disturb the measuring by giving higher or lower values. There are recommended places to measure but sometimes the distances are too short to measure at these places.

Finding the air channels to be measured was done from the flow sheet and if the flow sheet is not correctly drawn, like missing of air channels, made that air channels aren't included in the calculations. If it is correct that there were missing of air channels this will affects the air- and water balance and could be a reason why they weren't equal to each other before adjusting the values. In this case it seemed like exhaust air fan/fans was missing and a quick tour in the mill was done to see if any channels were missing without any results. To be sure that there weren't any missing of air channels more knowledge and time is needed. Since this master thesis was carried out during 20 weeks there wasn't time to just spend on fulfilling the balances, other calculations were also needed.

Further measurements need to be made to be for sure if the results are reliable and to validate that the leakage air is too high. From this more accurate evaluations can be made if interventions are necessary for the leakage air. Hopefully the results of mass- and energy balances will fulfills each other better without adjusting the values as much as has been done now.

Andritz which is a company that designed the drying hood used at Nymölla Mill were at visit where the air balance was discussed. After discussion with personnel at the mill and with Andritz [46] about the high leakage air and comparing the results with earlier studies, it is indicated that the zero level is too high and need to be adjusted for decreasing the leakage air. Which means that the observed results about the high leakage air is pointed in correct direction, but still further accurate measurements need to be made that take the openings of the damper in consideration and make sure of that no air channels are missed.

Even when measuring the air humidity with a wet and a dry thermometer there are some factors that could have caused some errors in the results. Like the thermometers are not calibrated which could give wrong values of the temperatures when measuring and also reading of the temperatures when they are not stable. The measured dry- and wet temperature are then used for deciding the air humidity in the Mollier chart for humid air which will give a reading error of the humidity. This would then more or less affect the water balance since the humidity of the air could be higher or lower. To sum up, one error during measuring will give more error when deciding the air humidity in the Mollier chart. To be more accurate about the calibration of the thermometers, at least two thermometers could have been used to compare the measured temperatures.

For the steam- and condensate balance it is seen that the condensate flow from the condensate tanks is much higher than the inlet steams which is not correct for the balance according to equation 3.2. The steam flows are read from a steam indicator and for the condensate flows, the pressures and power are measured and the condensate flow is then decided from a pump curve. It is more reliable to use the steam flows therefore the condensate flows has been adjusted from the steam flow in a way that they keep the same percent of the total condensate flow. A factor of why the condensate flow is incorrect is because the information about the pump wheel is false. It is assumed here that the given information was correct; therefore the given diameter for the pump wheel was used for deciding the volume flow in the pump curve and this would then affect the results. Also if uncalibrated instruments are used when measuring the current or the pressure would give errors for the results. Further measurements for the condensate flows need to be made since they are now adjusted from the steam flows.

When calculating the specific steam consumption it is seen that the value is higher than the theoretical value for evaporation of 1 kg water at 70 ºC (ΔH_{vap}=2333 kJ/kg). It is seemed that there is potential for energy savings since this value could be lowered. This value is based on estimated moisture content for the paper web and read values from the steam indicators. To be really sure that the calculated specific steam consumption is correct it is need to be sure that the steam indicators are calibrated and shows correct values and that the estimated moisture content for the paper web is correct.

What also needs to be taken in consideration is that the steam flows varies over the time for different parameters which will give varying specific steam consumption. Overall what has affected the results is that there are many parameters that are dependent on each other, like the paper quality and the actual grammage for the paper.

In this master thesis the measurements has been done on paper with a weight basis of 80 g/m^2 but with different qualities since it was hard to limit to only one quality because of the time. The different qualities require different processing and would therefore give different results when measuring. Also during calculation the actual weight of the paper hasn't been taken in consideration were the set point for paper is used which then give some errors when calculating the amount evaporated water. Because of the different qualities of the paper during measuring the results are not completely correct and would give some errors since the measured parameters (air humidity, air flow, moisture content etc.) should be used for the same quality.

6.2 Conclusions

The set up mass- and energy balances fulfills each other pretty well of what goes in comes out after the outgoing air flow has been adjusted. The power that goes in to the pre-drying section is 40 119 kW and what comes out is 39 955 kW, this give a difference of 0.4 %. For the after-drying the inlet power is 13 291 kW and the outlet power is 14 676 kW. This corresponds to a different of 9.4 % for the energy balance in the after drying section. Since the energy balances differ less than 10 % it is seemed that there are no losses and all the inlet power is utilized. This is observed when the adjusted exhaust air flows were used to fulfill the water balance. The observed air balances from the measurements were compared to earlier reports for the air balances. Both the measured results and the results in earlier reports showed that the hood air balance wasn't optimal in relation to the supply- and exhaust air.

From the Sankey diagrams it is seen that the leakage air make up the highest mass flow in to the drying sections. This leakage air don't necessary need to be this high and by reducing the air from 57.2 kg/s to 28.6 kg/s in the pre-drying section and from 44.5 kg/s to 18.8 kg/s in the after-drying section, an economical saving of total 3.8 million SEK/year could be made. This economical saving is observed if a supply air fan is installed or by increasing the engine speed for the supply air fans that already exists in the pre-drying section, since this will reduce the amount of leakage air by decreasing the zero level. If the amount leakage air is reduced then the amount steam can also be decreased.

The mass- and energy balances are calculated from a script built in the software MATLAB, where the measured parameters are filled in and the answers from one section is dependent on the previous answer. Therefore changes of parameters would easy give new values for the mass- and energy balances. This script can be used when changing parameters to see if the changing gave better results for mass- and energy balances. Like investigate if an extra air fan would decrease the leakage air.

The drying hood is a complex process where the mass- and energy balances are dependent on every little tiny change. This makes it hard to come up with a clear solution of what can be done for minimizing the power consumption. However, according to the mass- and energy balances that were set up instantaneous it shows that reducing leakage air would give energy savings but further and more measurements over a longer period needs to be made for better accuracy.

But the observed results are still usable to give an insight of what suggestions could be done for energy savings, in this case, reduce the leakage air flow. But exactly how much this should be reduced need more evaluations.

7. Future work

The suggestions of how the work done in this master thesis can be continued are presented in this chapter, where focus has been on how to find more energy savings opportunities.

In order to find more energy saving opportunities further evaluations can be made for the air humidity of the exhaust air. There are set points for the air humidity's but if they are the optimal values need to be evaluated, since higher air humidity's allowed more heat to be recovered and less steam is needed. But there's a limit of how high this humidity is allowed to be before it turns and more steam is instead needed. To evaluate the optimal exhaust air humidity changes of the set point for the air humidity could be done and then see how much steam that is needed for different values.

Evaluation of need of isolation for the air channels is a way of finding energy savings, where it is seen that some air channels were only isolated half ways and some channels didn't have any isolation. By isolation the air channels, the air is prevented from condensing or cooling. When the supply air keeps its higher temperature less steam is needed for heating the supply air since isolation gives less loss of heat. Also the humid air condenses easier since the more water in the air gives less range down to the saturated line in the Mollier chart for humid air. At this point dew is observed. The benefit of using isolation in this process should be investigated and if it is really necessary.

More time can be spent on the mass- and energy balances, especially the air-, water, and steam- condensate balances and try to fulfill these better which would give less adjustments, where the balances differed a lot in these measurements.

The drying hood is dependent on the steps before the paper comes to the dryer. By optimizing the pressing section that would give lower moisture content before the paper web goes in to the pre-drying section. This would give an energy saving in the pre-drying section. A thumb of rule is that 1 % increase in the dry matter content would give an energy saving of 4-5 % in the pre-drying section [7, 23]. Also investigation can be made for the air in the hall since the supply air in to the drying hood is taken from the hall. Therefore it is important that the ventilation in the hall is optimal because this would affect the drying.

Since everything is dependent on each other it is important to evaluate the different steps in the process to get the optimal condition.

8. Table of abbreviations

Table 30-Symbols

Symbol	Description	Unit
T	Temperature	°C
G	Mass flow rate	kg/s
X	Air moisture content	kg water/kg dry air
Q	Power	W
Cp	Specific heat	kJ/kg °C
ΔH_{vap}	Latent heat of evaporation	kJ/kg
b	Paper web width	m
v	Velocity	m/s
W	Basis weight paper – Dry material	g/m^2
m	Mass flow rate	kg/s
MR	Moisture ratio	-
DMC	Dry matter content	-
MC	Moisture content	-
p	Pressure	Pa
Δp	Pressure difference	Pa
A	Area	m^2
F	Volume flow	m^3
ρ	Density humid air	kg humid air/m^3 humid air
ρ_t	Dry density	kg dry air/m^3 humid air
T_v	Wet bulb temperature	°C
p_t	Pressure after the pump	Pa
p_s	Pressure before the pump	Pa
H_{pump}	Head	m
g	Gravity constant 9.82	m/s^2
P_{axial}	Axial power	W
η	Efficiency	-
I	Current	Ampere
U	Voltage	Volt
$\cos \varphi$	Power factor	-
P_{engine}	Electrical power	W
P_i^0	Steam pressure component i	mbar
DS	Dry substance	-
S	Concentration starch	g/m^2
y_{air}	Percent air flow	%
ρ_{H2O}	Density liquid water	kg/m^3
y_{cond}	Percent condensate flow	%
ΔT	Temperature difference	°C
x	Percentage steam	-
C	Cost	SEK/year
ΔQ	Power difference	W
Gi	Investment cost	SEK
a	Cash surplus	SEK/year

Table 31-Subscripts

Symbol	Description
s	Steam
cyl	Cylinder
in	Inlet
out	Outlet
p	Paper
c	Condensate
bt	Blow through steam
sup	Supply air
leak	Leakage air
exh	Exhaust air
evap	Evaporation
fib	Fiber
i	Measure point index
$H_2O(l)$	Liquid water
air	Air
$H_2O(g)$	Water vapor
mach	Machine
pump	Pump
engine	Engine
liq	Liquid
starch	Starch
hall	Hall paper machine
ctj	Condensate tank number j
1	Before
2	After
lp	Low-pressure steam
mp	Middle-pressure steam
flash	Flash steam
tot	Total
red	Reduced
aph	Air-preheater
surf	Surface condenser

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Appendix A – Heat transferring from steam to paper web

When the steam hits the condensate inside the cylinder heat is transferred from the steam through the condensate with convection/conduction and the heat flux from the steam to the condensate is calculated as below:

$$q_s = \alpha_{cond}\left(T_s - T_{cyl,in}\right) \tag{A.1}$$

The heat from the condensate is then transferred through the cylinders wall by conduction and with Fourier's law the heat flux through the wall can be calculated if it is assumed that the media is homogenous. For this, the thermal conductivity coefficient λ_{cyl} is needed and the thickness for the cylinders wall b_{cyl}.

$$q_s = \frac{\lambda_{cyl}}{b_{cyl}}\left(T_{cyl,in} - T_{cyl,out}\right) \tag{A.2}$$

The heat transfer from the cylinder to the paper web occurs by conduction and is calculated with a contact heat transfer coefficient that takes the resistance of air between the paper web and cylinder in consideration. With this the heat flux can be decided as following:

$$q_s = \alpha_{cont}\left(T_{cyl,out} - T_p\right) \tag{A.3}$$

Rewriting of all the formulas A.1-A.3 above the total heat flux from the cylinder is [15]:

$$q_s = \frac{T_s - T_p}{\left(\frac{1}{\alpha_{cont}} + \frac{b_{cyl}}{\lambda_{cyl}} + \frac{1}{\alpha_{cond}}\right)} \tag{A.4}$$

The energy that is transferred to the paper from the steam must be equal to the energy that is needed for evaporating the water in the paper. This evaporating energy for the paper is the product of the rate of mass transfer from a wet paper surface and the latent heat of evaporation. With Stefan's equation the rate of mass transfer from a wet paper surface can be calculated as following equations A.5-A.8 below shows:

$$m_{evap} = \frac{k_g M_w P}{R(T_p + 273)} \ln\left[\frac{P - p_{air}}{P - p_p}\right] \tag{A.5}$$

$$k_g = \frac{\alpha_{air}}{\rho C_p} \text{ – Mass transfer coefficient} \tag{A.6}$$

$$p_{air} = \frac{X}{\frac{18}{29} + X} \cdot P \tag{A.7}$$

$$p_p = 133.322 \cdot \exp\left(18.3036 - \frac{3816.44}{T_p + 227,02}\right) \tag{A.8}$$

And with the equations A.5-A.8 the energy required for evaporating water from the paper can be written as equation A.9 below [15]:

$$q_{evap} = \frac{k_g M_w P \Delta H_{vap}}{R(T_p + 273)} \ln\left[\frac{P - p_{air}}{P - p_p}\right] \tag{A.9}$$

Appendix B - Derivation for the wet-bulb temperature

The heat transferring that is needed for evaporating the water is equal to the mass transferring of water multiplied with the enthalpy of evaporation and give following expression:

$$\frac{q}{A} = \frac{N}{A} \Delta H_{vap} \quad \text{(B.1)}$$

$$\alpha(T - T_w) = k_g(x_w - x)\Delta H_{vap} \quad \text{(B.2)}$$

$$\rightarrow T_w = T - \frac{k_g}{\alpha} \Delta H_{vap}(x_w - x) \quad \text{(B.3)}$$

The Reynolds analogy is used for the mass- and heat transferring for the system of air and water and with this analogy it is shown that

$$\frac{\alpha}{k_g} = C_f = C_{P,air} + x \cdot C_{P,H_2O,(g)} - \textit{The humid air heat capacity}$$

Rewriting formula B.3 with Reynolds analogy results in the expression B.4 below:

$$T_w = T - \frac{\Delta H_{vap}}{C_f}(x_w - x) \quad \text{(B.4)}$$

Appendix C – Shrinkage of paper

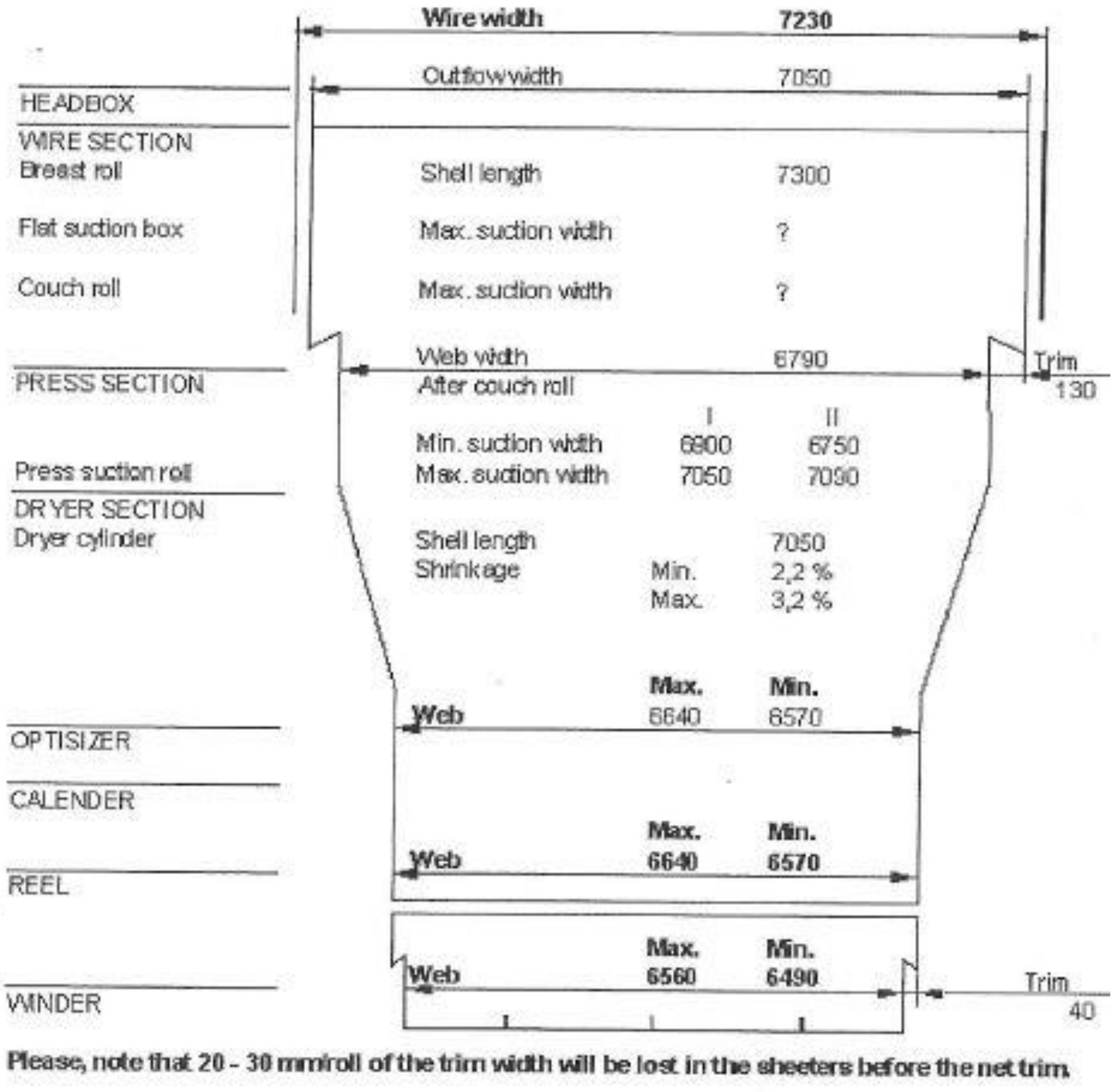


Figure C. 1 – The shrinkage of paper through the paper machine

Appendix D – Mollier chart for deciding air humidity

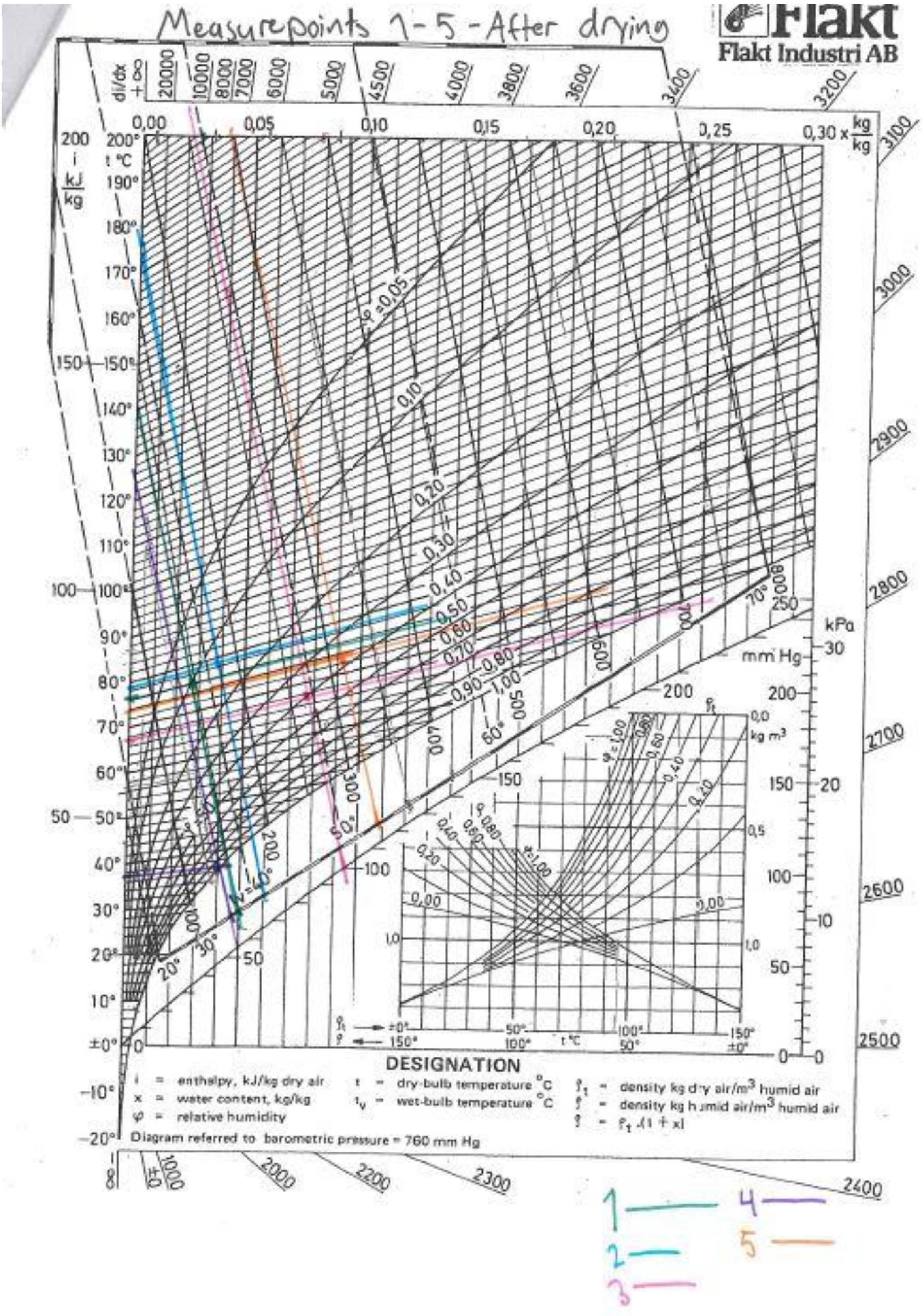


Figure D. 1-Mollier chart for humid air

Appendix E – Steam flows for drying section

Steam flows for both the pre-drying section and after-drying section are read from Metso DNA at page 8.1 PM1 and the read values are shown in Table E. 1 respectively Table E. 2.

Table E. 1- The steam flows in to the pre-drying section

Steam type	Pre-drying [t/h]	Date the values were used
Middle pressure	24	2016-04-19
Low pressure	15.3	2016-04-19

Table E. 2 The steam flows in to the after-drying section

Steam type	After-drying [t/h]	Date the values were used
Middle pressure	0	2016-04-29
Low pressure	6.3	2016-04-29

Where the middle-pressure steam should go in to is decided with valves. When reading the middle-pressure steam in Metso DNA at page 8.1 PM1 it gives the total value of the produced middle-pressure steam. How much of the middle-pressure steam that goes in to the pre-drying section, respectively the after-drying section needs to be check with the valves in each section. For the case 2016-04-19 the total production of middle-pressure steam was 24 t/h but the valves at the after-drying section where the middle-pressure steam comes in were closed. In the pre-drying section the valves where opened. This means that all the produced middle-pressure steam goes in to the pre-drying section this day.

Figure E. 1 shows a simplification for the valves for the middle-pressure steam in the pre-drying section and after-drying.

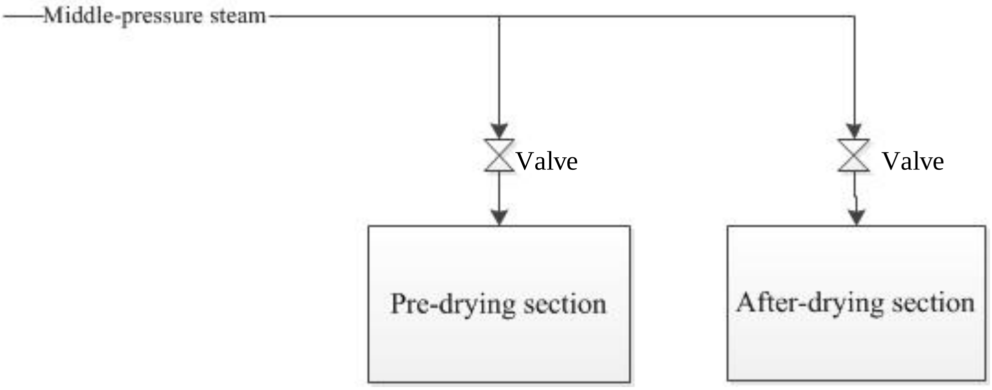
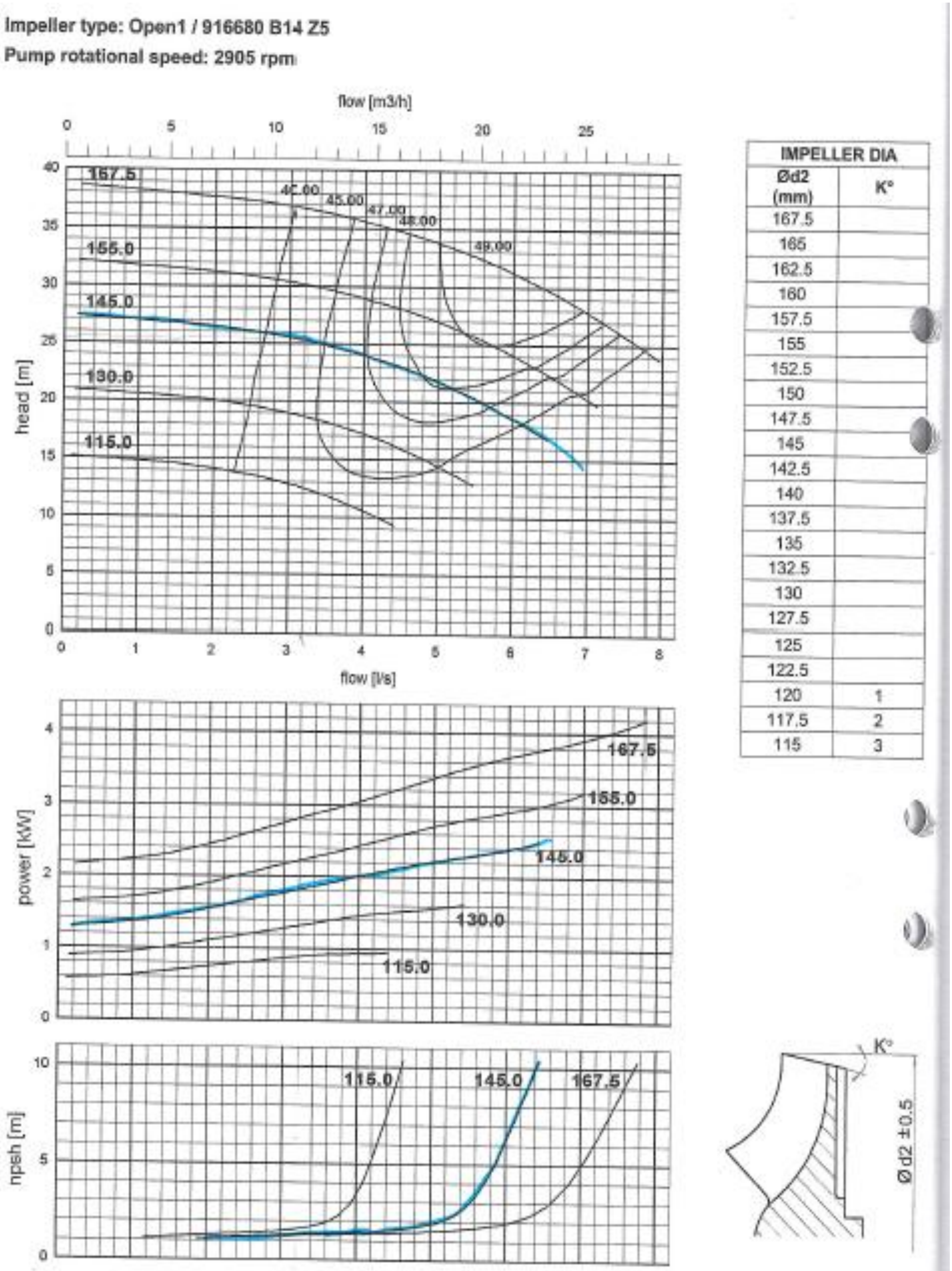


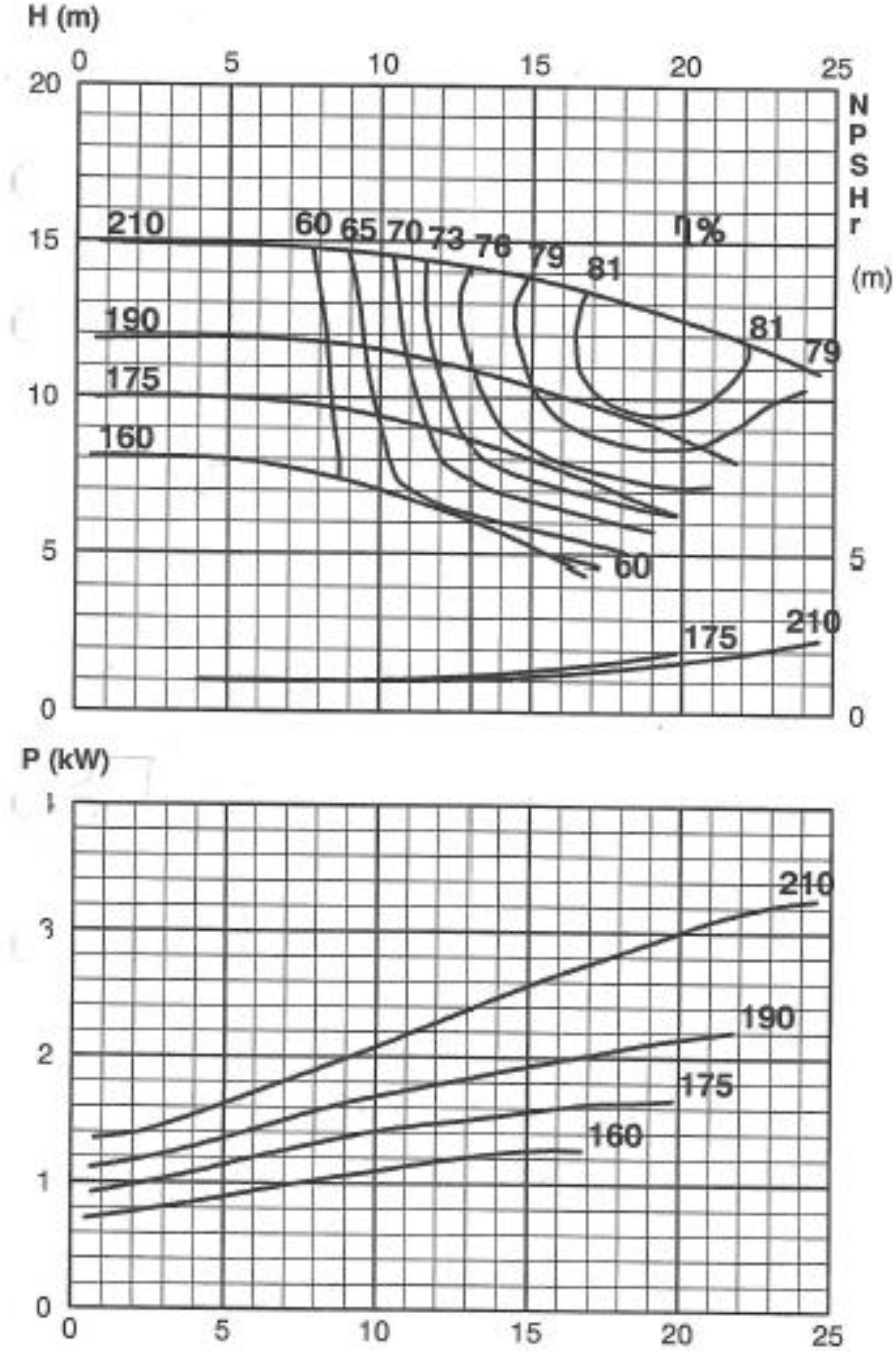
Figure E. 1- Simplification of distribution of the middle-pressure steam in the drying section

Appendix F – Pumpcurves



IMPELLER DIA	
Ød2 (mm)	K°
167.5	
165	
162.5	
160	
157.5	
155	
152.5	
150	
147.5	
145	
142.5	
140	
137.5	
135	
132.5	
130	
127.5	
125	
122.5	
120	1
117.5	2
115	3

Figure F. 1- Pumpcurve for the pump at condensate tank 1-Pre-drying section



IMPELLER DIA	
Ød$_2$ (mm)	Ød$_b$ (mm)
210	193
205	190
200	186
195	183
190	180
185	178
180	175
175	173
170	170
165	167
160	163

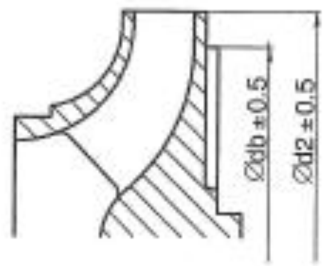
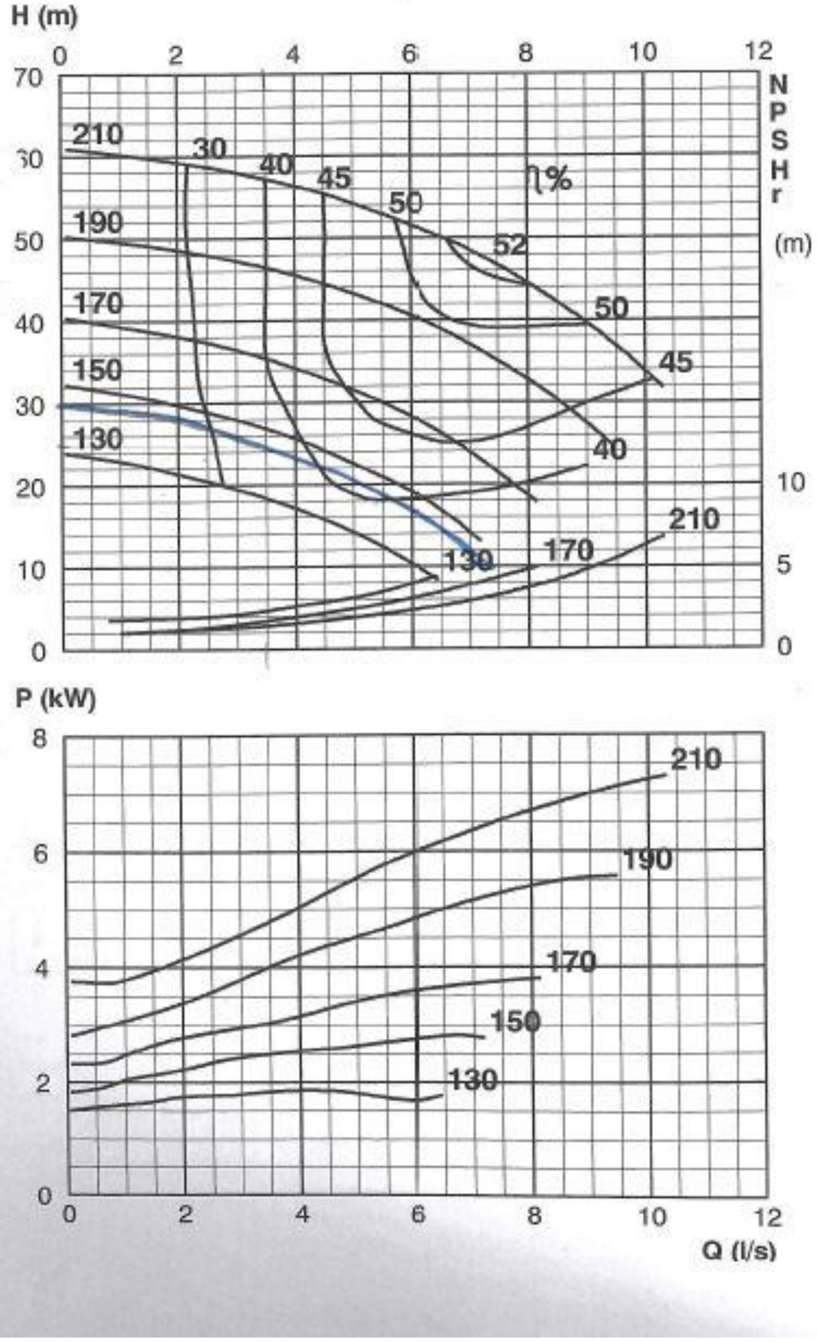


Figure F. 2- Pumpcurve for the pump at condensate tank 2-Pre-drying section



IMPELLER DIA	
Ød_2 (mm)	Ød_b (mm)
210	149.0
205	147.5
200	146.0
195	144.5
190	143.0
185	141.5
180	140.0
175	138.5
170	137.0
165	135.5
160	134.0
155	132.5
150	131.0
145	129.5
140	128.0
135	126.5
130	125.0

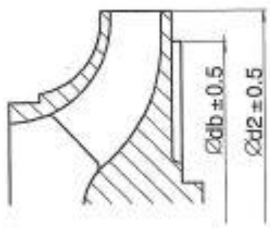
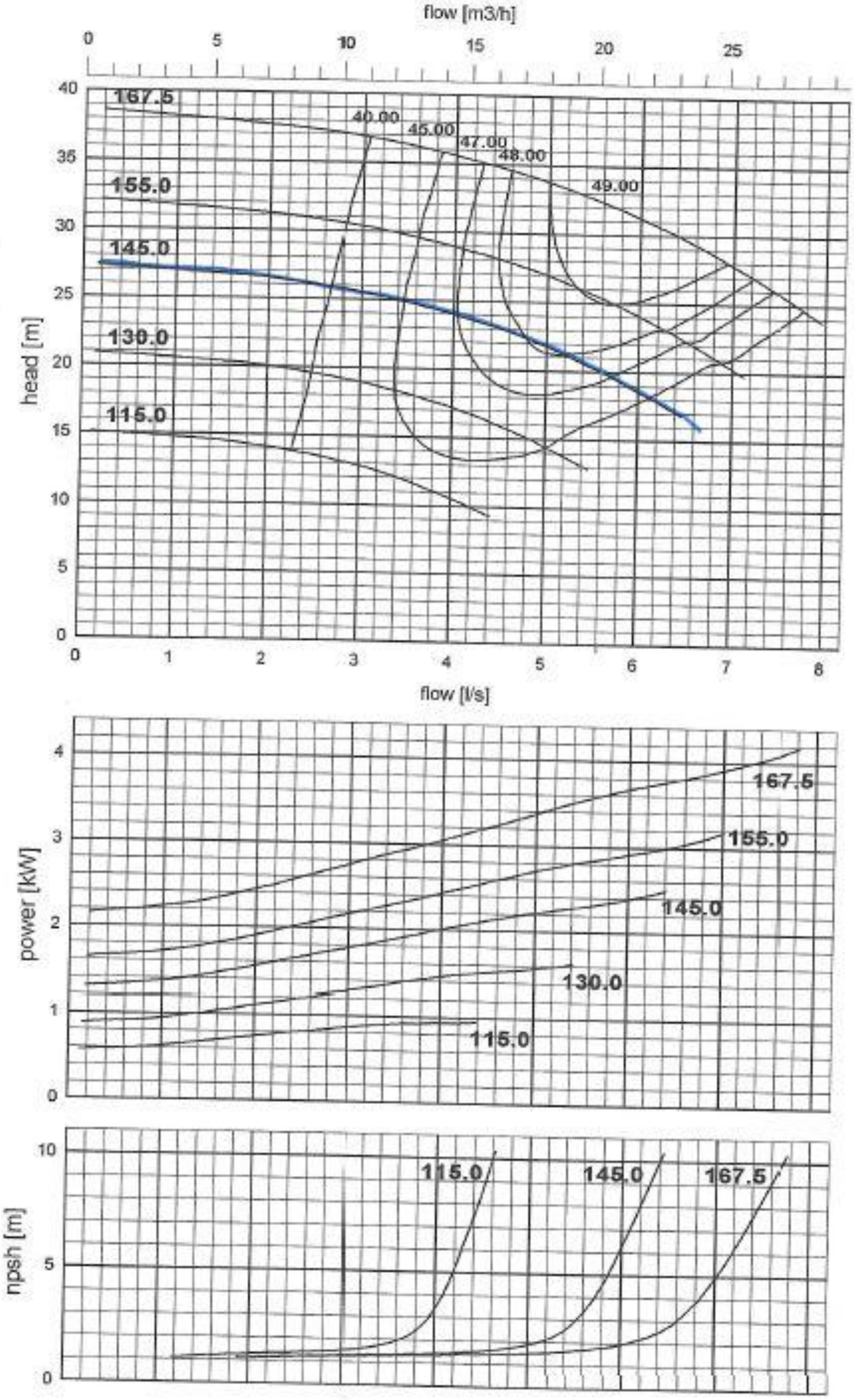


Figure F. 3- Pumpcurve for the pump at condensate tank 5-After-drying section



IMPELLER DIA Ød2 (mm)	K°
167.5	
165	
162.5	
160	
157.5	
155	
152.5	
150	
147.5	
145	
142.5	
140	
137.5	
135	
132.5	
130	
127.5	
125	
122.5	
120	1
117.5	2
115	3

Figure F. 4- Pumpcurve for the pump at condensate tank 6 – After-drying section

Appendix G - Pressures in the condensate tanks for drying section

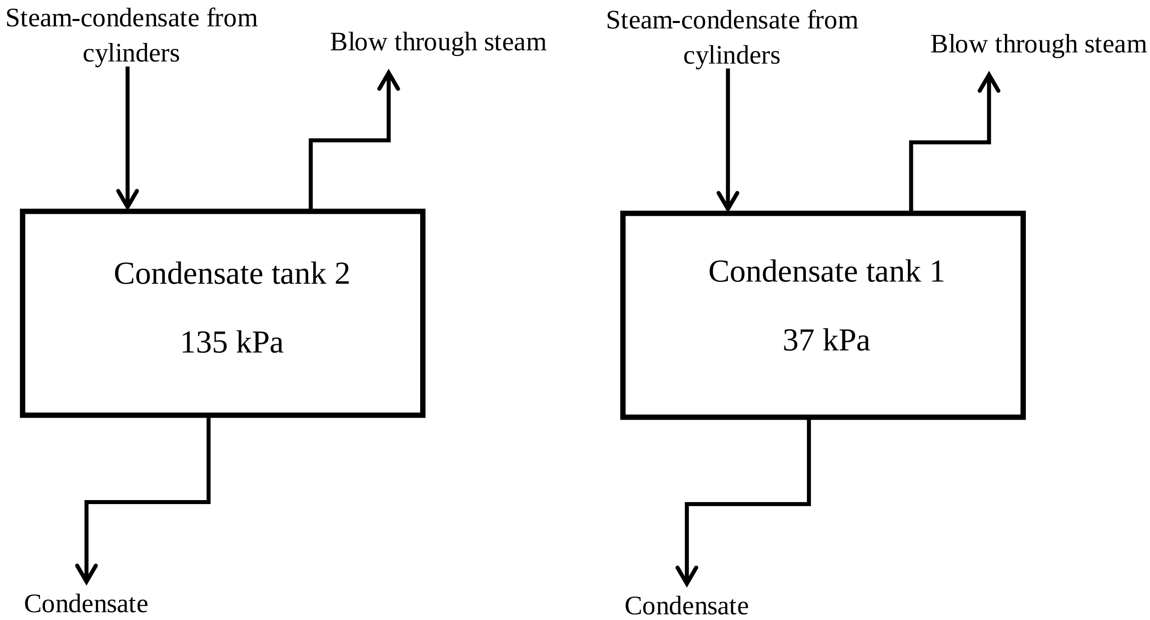


Figure G. 1- The steam pressures in the condensate tank for pre-drying section. Values are read 12/4-16 from Metso DNA page 8.2 PM1

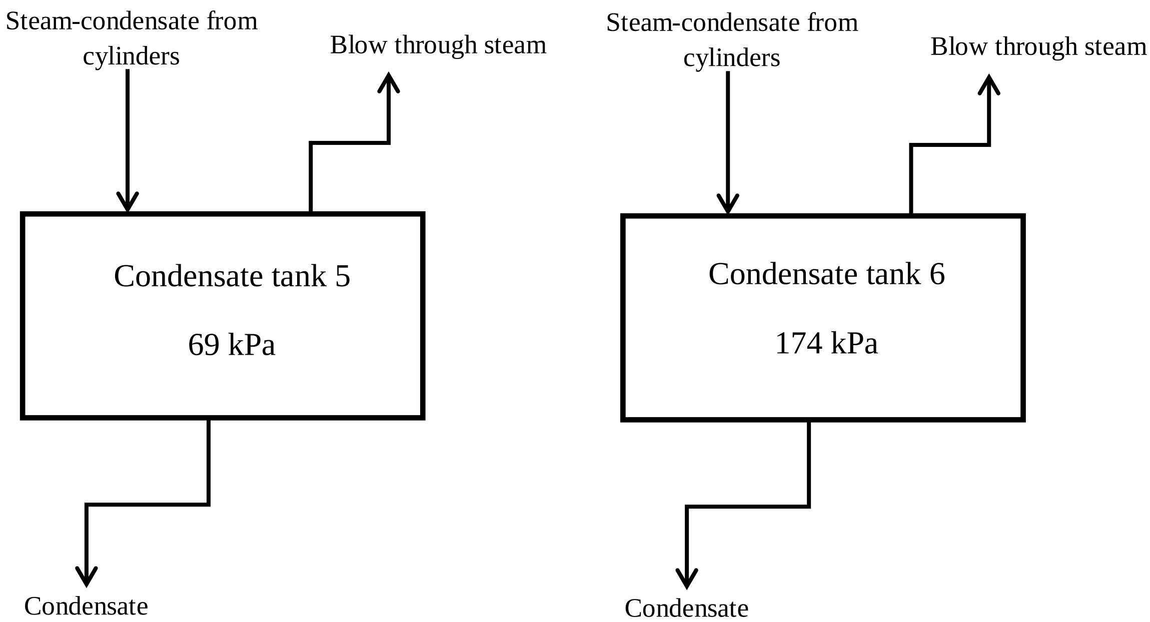


Figure G. 2- The steam pressures in the condensate tanks for after-drying section. Values are read 2(5-16 from Metso DNA page 8.3 PM1