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IMPROVED ENERGY EFFICIENCY IN JUICE PRODUCTION THROUGH WASTE HEAT RECYCLING

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ABSTRACT

The global demand for Nordic wild berries has increased steadily, partly due to their healthy properties and their good taste. Juice concentrate is produced by pressing berries and heating up the juice. The by-products are berry skins and seeds. Traditionally, the by-products have been composted. Higher competiveness can be achieved by decreasing the production cost and increasing the product values. The berry skins and seeds have a commercial value since they are rich in vitamins and nutrients. To use and sell these by-products, they need to be separated from each other and dried to a moisture content of less than 10 %wt. A berry juice industry in the north of Sweden has been studied in order to increase the energy and resource efficiency and optimize the quality and yield of different berry fractions. This was done by means of process integration with thermodynamics and psychrometry along with measurements of the berry juice production processes. Our calculations show that the drying system could be operated at full without any external heat supply. This could be achieved by increasing the efficiency of the dryer by recirculating 80 % of the drying air and by heating the air with heat from the flue gases from the industrial boiler. This change would decrease the need for heat in the dryer with about 64 %. The total heat use for the plant could thereby be decreased from 1204 kW to 1039 kW. The proposed changes could be done without compromising the production quality or the lead time.

Keywords: renewable energy resources, pinch analys, drying, thermodynamic, psychrometrics

NOMENCLATURE

Symbols

$\dot{m}$	Mass flow	[kg/s]
x	Abs. moisture content	[kg water/kg dry air]
P	Pressure	[kPa]
h	Enthalpy	[kJ/kg]
θ	Relative humidity	[%RH]
t, T	Temperature	[°C]
r	Latent heat	[kJ/kg]
c_p	Specific heat	[kJ/kg.K]
$\dot{Q}$	Heat	[kJ/s]
β	drying utilization	[%RH]

Subscripts

i	*point i*
1	*out door, point 1*
2	*after the burner, point 2*
2´	*after the mixing point, point 2´*
3a	*exiting air, after air separation*
3b	*air recirculation*
w	*saturated conditions*
v	*vapor*
d	*dry air*
s	*saturated*
air	*air*
max	*maximum*

1. INTRODUCTION

The demand for Nordic wild berries (in particular bilberries, lingonberries and cranberries) has increased steadily since they are considered a healthy and tasty part of the diet. Currently, only the juice is sold, while the by-products, skins and seeds, are composted. Increased competition along with rising energy prices has prompted the industry to improve the resource and energy efficiency. It has been found that the skins and seeds contain antioxidants [1], making them an added-value product to the berry juice itself. In particular, the berry seeds consist of 20 %wt oil, which has a significant commercial value for health care and cosmetic use. Increased energy efficiency is also in the interest of the European Union, which can be seen in the 20-20-20 goals (20 % increased energy efficiency, 20 % increased renewables and 20 % reduced carbon dioxide emissions until 2020). The objective is to reduce the global temperature rise.

The aim of the paper is to propose how the berry industry processes can be developed into an energy and resource efficient integrated process, optimizing the energy usage while retaining and drying the berry skins and seeds. In particular, recirculation of heat and dry air will be investigated. However, in order to maintain a high quality of the seeds and skins, the maximum air temperature was set to 90 °C and the maximum moisture content of the skins and seeds was set to 10 %.

The juice factory that has been considered in this paper was owned by the dairy company Norrmejerier and is located in Hedenäset in the north of Sweden, close to the Finnish border.

The rotation dryer that was used in the experiments was provided by the company Aromtech, which uses dried berry seeds to produce dietary supplements. Hot air is circulated through the dryer, absorbing some of the moisture of the wet press cake. The air goes in one direction and the press cake in the other, while the drum is rotating in order to increase the surface area between the press cake and drying air. An overview of the effectiveness of different convective dryers can be found in [2].

Section 2 of the article contains a description of the dryer, which has been the focus of the research, and a description of the applied theoretical models; Pinch analysis [3-4], psychrometrics [5-6] and thermodynamics.

Section 3 contains the theory and calculations of the process integration tool Pinch analysis and psychrometrics. The focus in process integration is optimizing the system as whole instead of the individual units. Ambient conditions (surrounding air temperature, air moisture content, press cake moisture content and press cake temperature) will also affect the results to a certain degree but this has not yet been taken into account.

Section 4 presents the results and the discussion and the conclusions can be found in section 5.

2. MATERIAL AND METHODS

2.1 Process analysis and process integration

The present investigation is made partly through experiments on the dryer, partly through modeling of the juice plant and the drying process. To analyze the possible magnitude of recoverable heat, the process integration tool Pinch analysis was used. We analyzed the total energy usage for three different cases; Case A – the reference case, where the juice plant and the dryer has not been improved or changed; Case B – recirculation of drying air in the dryer, in order to better use the energy in the drying air. Case C – recirculation of drying air along with a heat exchanger between the exhaust gas of the steam boiler and the heating air for the dryer. In Case A, an oil burner is assumed to provide the necessary heat for the drying process. This is the current situation. Case B needs only a small investment (some pipes and air flow splitting devices). Case C requires an additional investment in the form of heat exchangers attached to the flue gas pipes. The hot and cold streams from the boiler and the drying system were used to find the possible pinch heat. The energy need for the drying system was given by the results from the experimental measurements.

2.2 Experimental measurements old drying process

The current drying process (Case A) is illustrated in Figure 1 with material and air flow. In Figure 2, the Mollier diagram of the process is schematically illustrated, showing the enthalpy, the absolute humidity and the temperature for the air state at the points 1-3. The drying process proceeds in the following order. The drying cycle starts with a flow of cold and dry outdoor air (point 1). Between points 1-2, the drying air is heated by a burner (a ThermoBetox 120 [7]) through a heat exchanger. During the heating step, the absolute moisture content is not changed, which will result in a horizontal line in the Mollier diagram (Figure 2). After point 2, a circulation fan transports the air through the rotating dryer drum, where the drying process takes place. After the drying drum, the skins are separated from the drying air by two particle separators. Point 3 is directly after the skin separation. Finally, the air is circulated to the outside. The dryer is a counter current dryer, which feeds the press cake in the counter current direction. The dryer partly separates the skins from the seeds. The remaining seeds and skins fuse together to form larger particles called pellets. The seeds and the pellets exit the dryer nearby the incoming air, while the skins follow the air out of the dryer.

The experimental sampling was done with an Intab PC-logger 3100 logger system [8] using the software LabView

[9]. The sampling interval was 2 seconds and the data was averaged over 6 seconds. The dry and wet bulb temperature was sampled in point 3 and the relative humidity and the dry bulb temperature were measured in point 1. The temperature of the air stream at the boundary layer close to the dryer walls was measured at the entrance and the exit point of the drying air (points 1 and 3). Between the burner and the dryer, two air iris valves were installed, one to control the pipe area into the dryer and one to control the amount of by-pass air evacuated before the dryer. Together, these two valves allow us to regulate the temperature and the flow of the incoming air. In point 2, the dry bulb temperature was sampled and also the dynamic pressure in order to calculate how the airflow changes depending on different settings. Each 30 minutes the product was examined by measuring the moisture content and total weight of the exiting seeds and skins to analyze the effect of the dryer. During the process, the press cake was fed into the dryer by a feeding screw.

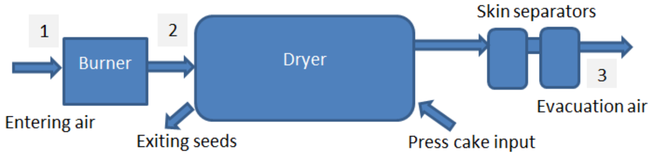

Figure 1, Experimental setup of the drying process of the current system, Case A.

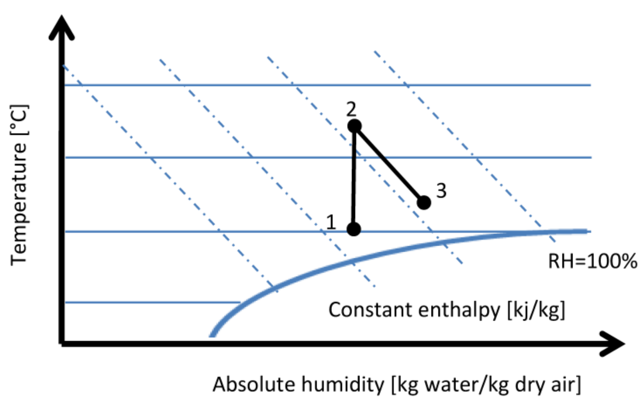

Figure 2, Schematic Mollier diagram of the current drying process (without air recycling).

The experimental setup allows for measuring the humidity difference of the drying air before it enters the dryer compared to the humidity after the dryer. This allows us to calculate how much of the energy in the drying air has been used for the actual drying.

2.3 Improved system with recirculation

The experimental measurements showed that only a small part of the available drying power in the air was used (see Figure 5). It is possible to utilize the drying effect further through recirculation of the exiting air due to the low humidity of the drying air after exiting the dryer. The recirculation solution (where a part of the airflow is recycled at point 3), included in Case B and C this system is shown in Figure 3, with support of the Mollier diagram in Figure 4. This results in a mixing point before the dryer, between the heated outdoor air from the burner, point 2, and recirculated drying air, point 3. The mixed air temperature in 2′ can be calculated through the Markel-Bosnjakovic [10] mixing principle. The fraction of air that can be recycled depends on the humidity and temperature before and after the dryer. This can be calculated through thermodynamics, psychrometrics and conservation of mass and energy.

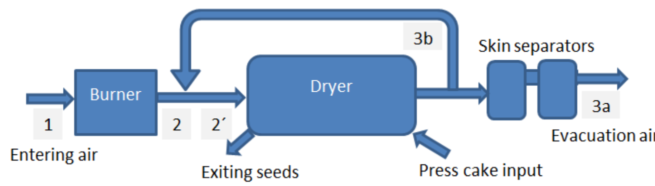

Figure 3, Suggested improvement of the drying system, where the exiting air is partially recycled.

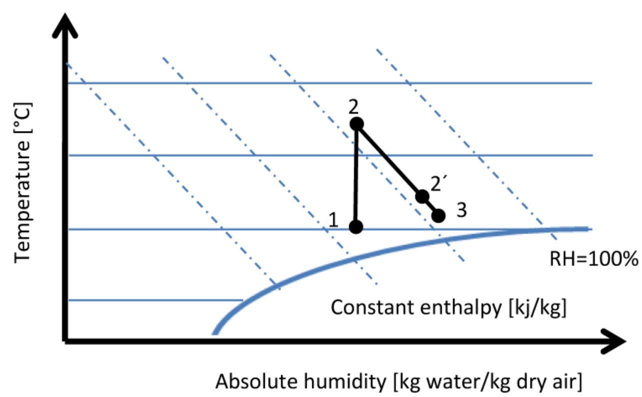

Figure 4, Schematic Mollier diagram of drying process with air recycling.

A theoretical maximal level of the humidity in point 3 would be 100 %RH. However, if the humidity difference between the drying air and press cake is too low, the drying process will be very slow. Therefore, in order to avoid a slow drying process and condensation, we set the upper limit of the relative air humidity to 85 %RH.

3. THEORY AND CALCULATION

In this section, the theory of psychrometrics is explained along with its application to the case at hand. The dry bulb temperature and the relative humidity were known from the measurements.

Furthermore, the wet and bulb temperature were measured in point 3 and the relative humidity and the dry bulb temperature were measured in point 1. In order to calculate the enthalpy and the humidity of the air at these points, Eqs. (3) and (4) were used.

The saturation pressure for the dry bulb temperature can be calculated through a polynomial Eq. (1), and the partial water pressure is defined with in Eq. (2).

$$P_{s,d} = -0.00000182t_d^4 + 0.00161t_d^3 + 1.14t_d^2 + 11{,}1 \tag{1}$$

$$P_w = (\theta/100)P_{s,d}, \tag{2}$$

where t_d is the dry air temperature, P_w is the partial water pressure, θ is the relative humidity of the air and $P_{s,d}$ is the saturation pressure of the dry air.

The absolute humidity in the air, x_1, can be calculated through Eq. (3), when the partial water pressure and the absolute pressure, P_{tot}, are known:

$$x_1 = 0.622P_w/(P_{tot} - P_w). \tag{3}$$

The enthalpy, h, can be calculated through Eq. (4) when the temperature and the absolute humidity are known:

$$h = 1.007 * t_d + x * \left(r + \left(c_{p,v} * t_d\right)\right), \tag{4}$$

where r and $c_{p,v}$ are the latent heat and the specific heat of the water.

The incoming air state, state 1, is now defined through Eqs. (1)-(4). With the measured values of the dry and wet bulb temperature in state 3, the enthalpy can be determined through Eq. (4) after the absolute moisture content, x_3 has can be calculated with Eqs. (5) and (6).

$$x_{wet} = 0.622\frac{P_{s,w}}{(P_{tot}-P_{s,w})}, \tag{5}$$

$$x_3 = \frac{(c_{p,d}*(t_w-t_d)+x_{wet}*r)}{(r+c_{p,v}*t_d-c_{p,w}t_w)}, \tag{6}$$

where x_{wet} is the mass ratio of wet air to dry air, $P_{s,w}$ is the saturation pressure of the wet air, t_w is the wet air temperature and $c_{p,d}$ is the specific heat of the dry air.

The air states in point 1 to 3 are now defined. When considering the recycling of air, the mixing of the air stream for air states 2, 2´and 3b needs to be taken in consideration. A mass balance and mixture relationship is used to assure that the moisture content in air state 3 is lower than 85 %, as discussed in section 2.3:

$$x_2 = \frac{\dot{m}_1}{\dot{m}_1+\dot{m}_{3b}}x_1 + \frac{\dot{m}_{3b}}{\dot{m}_1+\dot{m}_{3b}}x_2, \tag{7}$$

$$x_3 = x_{2'} + dx. \tag{8}$$

Since we now know the moisture content of the air in state 3, the necessary temperature in the air state 2 can be found from Eqs. (4), (9) and (10), using the specific amount of recirculation air and its temperature.

$$\dot{Q}_{2'} = \dot{Q}_2 + \dot{Q}_{3b}, \tag{9}$$

$$\dot{Q}_\iota = \dot{m}_\iota * h_i, \tag{10}$$

where $\dot{Q}_i$ is power at point i.

The temperature drop with between state 2´and 3b was measured to be 15 °C. We assume that this temperature drop will remain constant even when air is recirculated, since it represents the drying energy. This will not be exactly true when the heat transfer decreases as the air humidity increases with recirculation, but it remains a fair approximation. The drying power fraction, i.e. the utilized fraction of the available energy in the drying air, can be defined as

$$\beta = \frac{\theta_3-\theta_{2'}}{\theta_{max}-\theta_{2'}}. \tag{11}$$

The experimental result of this equation is presented in Figure 5.

4. RESULTS AND DISCUSSION

The results of the calculations were the recycling of process heat and an increased drying efficiency.

4.1 Experimental measurements

The experimental measurements show that the drying process is done with low efficiency. This means that only a small part of the drying power fraction in the drying air, as defined in Eq. (11), is used. The air only absorbs around 8.5 g H_2O per kg air during the circulation through the dryer. At 100 %RH it could absorb up to 140 g H_2O per kg air. The measurements show that the drying power fraction is never higher than about 10 % for the current setup, see Figure 5.

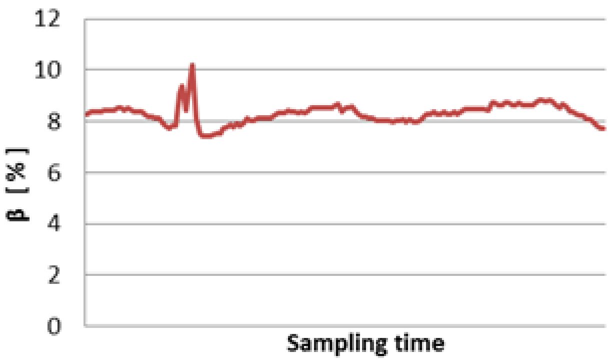

Figure 5, Experimental result of the drying power fraction, i.e. the utilized fraction of the available energy in the drying air, as defined in Eq. (11) over time.

The experiments show that the contact area, the press cake time in the dryer and the drying temperature are too low to obtain an effective drying. The contact area between the press cake and the drying air could be increased by changing the particle size or improving the air flow. The experimental measurements show that a high air flow forces the press cake particles towards the walls of the dryer, which decreases the contact area and the press cake time in the drying air. Note that the dryer walls have lower temperature than the drying air due to the transmission losses to the surroundings. The measurements show that with a drying air temperature of 90 °C and a surrounding air temperature of 0 °C the dryer wall temperatures were 52 °C close to the air outlet. The press cake time in the drying air could be improved by increase the rotation speed of the dryer and increase the length of the dryer, but this would require a investment cost. The drying temperature cannot be increased much beyond 90 °C due to product quality aspects but the temperature could be more uniform over the dryer with the help of better insulation.

It was found that 165 kW is sufficient heat for the dryer to dry 90 kg/h press cake with an incoming air temperature of 90 °C and a surrounding air temperature of 20 °C. This results in a total heat load of 1204 kW for the juice plant, including the dryer system.

4.2 Improved drying system with recirculation

Recirculating air (Cases B and C), as shown in Figure 3, means that the incoming air in point 2 must have a higher temperature than 90 °C. When mixing with the cooler and wetter recirculated air, the mixed drying air must contain enough heat for the drying. The required incoming temperature at point 2 depends on the fraction of recirculated air. Using the equations in section 3, the temperature T_2 can be calculated. This is shown in Figure 6. For a reasonable recirculation fraction of 80 %, the incoming air temperature T_2 must be 152 °C in order to achieve the necessary amount of heat (165 kW) in the drying air at point 2'. A higher recirculation fraction would lead to worse moisture transfer from the press cake to the drying air. The total heat load of the dryer could thereby be decreased from 165 kW to 54 kW. This would, however, require a different heater capable of delivering smaller air flows at higher temperatures.

The recirculated air has a lower temperature mostly due to the thermal losses during the drying and the recirculation.

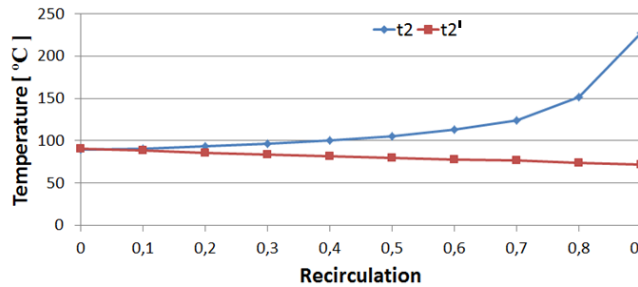

Figure 6, Temperature depending on recirculation fraction. The blue curve is the air temperature at point 2, the red curve is the air temperature at point 2'.

In Figure 7, the heat in the air at the different points is shown.

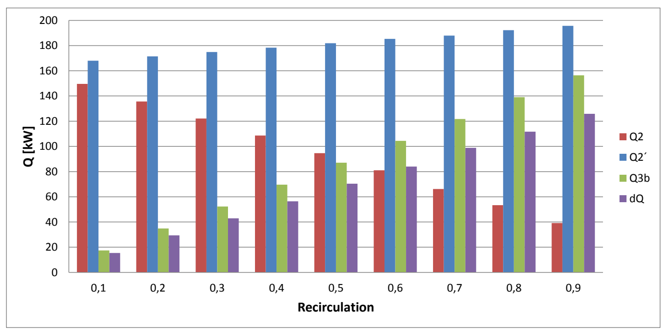

Figure 7, Energy streams dependence of recirculation grade, dQ represents the decreased amount of heat needed for the drying process.

The heat recovery decreases the heat supply to the dryer by between 9 and 64 %, depending of the fraction of recirculation.

4.3 Process analysis and process integration

It has been found that the energy in the flue gases is sufficient to deliver heat to the dryer as an alternative to the oil burner. The flue gases supply and target temperature were 220 °C and 105 °C, respectively. The target temperature was set to 105 °C so no condensation of hydrochloride and water vapor would occur to avoid acid.
The process integration is described below.

Three different process cases were analyzed; Case A – The reference case, where the dryer is heated by a separate oil burner, as it is today. Case B – The dryer is heated by a separate oil burner with a recirculation fraction of 80 %, as discussed in section 4.2. Case C – The dryer is heated with heat from the flue gases from the steam boiler and the drying air recirculation fraction remains at 80 %.

The integrated grand composite curve below illustrates the heats loads of the system. Figure 8 shows the heat load of the system for Case A and B. Comparison can be done between the two cases to see the effect of the recirculation in the drying system.

The total required heat load for the juice factory, including the dryer, is 1204 kW for Case A and 1093 kW for Case B.

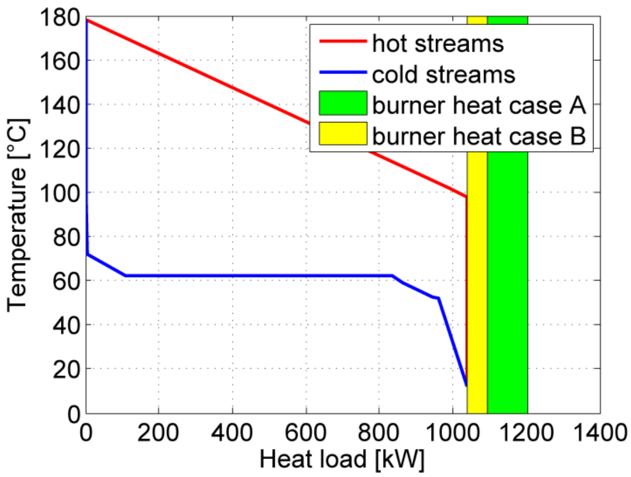

Figure 8, Composite curve for Case A and B.

In Case C, the total heat load can be deceased with 14 %, which corresponds to a total heat supply of 1039 kW, This is schematically shown in the integrated grand composite curve for Case C in Figure 9.

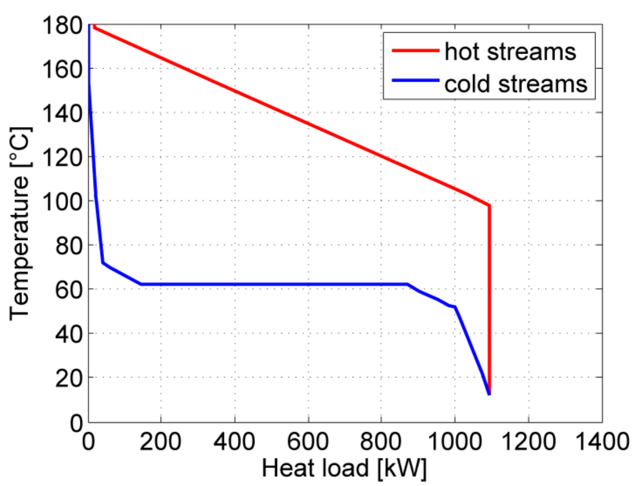

Figure 7, Composite curve for Case C.

5. CONCLUSIONS

It has been found that the current dryer system uses less than 10 % of the total drying power fraction, i.e. the exiting drying air could absorb a lot more water. This depends on the construction of the dryer and the particle size of the dried material. Some changes could be done to make better use of the drying power fraction and thereby decrease the amount of energy needed. One alternative is to let the material stay a longer time in the drying drum. This could be done by increasing the length of the dryer. Another alternative would be to increase the drying temperature, but this might result in a lower quality of the skins and seeds.

The drying process heat demand is 165 kW at a production of 90 kg press cake per hour to achieve a good quality. For the current situation, this results in a total heat load of the juice plant of 1204 kW.

A decrease of the heat supply, with a low investment cost, could be done if Case B is implemented, where 80 % of the drying air is recirculated. This would lead to a decreased heat load of the drier by 64 %, resulting in a total heat load of the juice plant of 1093 kW. This would require the heated air, in point 2 in Figure 3, to be 152 ºC.

If Case C is implemented, where 80 % of the drying air is recirculated and the drier heat is provided by a heat exchanger from the exhaust gases of the steam boiler of the plant, the total heat load for the juice plant is decreased to 1039 kW.

The investment costs have not been calculated but they include some piping for the recirculation and possibly a heat exchanger between the flue gases and the drier.

6. ACKNOWLEDGEMENTS

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