EU Project No.: 723576

DryFiciency

Waste Heat Recovery in Industrial Drying Processes

H2020-EE-2016-2017-PPP

Valorisation of waste heat in industrial systems (SPIRE PPP)

Specification of performance indicators and validation requirements

D1.2

This is a Deliverable of WP1

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Statement of originality:
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EXECUTIVE SUMMARY

The focus of the DryFiciency project is to develop and implement cost-efficient high temperature heat pump solutions for industrial thermal drying processes. Industrial drying and dehydration processes require intensive use of energy, and estimates show that 12-25% of the national industrial energy consumption in developed countries is attributable to industrial drying. Currently, most of this energy is based on the use of fossil fuels with no utilization of waste heat streams. Hence, there is a great potential for more efficient and environment-friendly technologies within industrial drying processes. The development of such technologies is also of relevance for a number of other industrial sectors, such as the petro-chemical industry and the pulp and paper industry.

Three different industrial drying processes are considered in the project. These are applications and development of thermal drying in the agricultural raw material industry, in the ceramic industry, and in the waste management industry. In the agricultural raw material application, the heat pump technology will be developed for the production and drying of starch from potatoes, wheat and corn. The demonstration of this heat pump system will take place at Agrana Stärke GmbH in Pischelsdorf, Austria. For the application in the ceramic sector, the focus is to integrate novel heat pump technology for green brick drying. This technology will be implemented by Wienerberger AG at Uttendorf, Austria. Finally, the application for the waste management industry focuses on the drying of sludge/biomass. This heat pump drying system will be developed and installed by Scanship, Norway.

To utilize the waste heat streams of the above drying processes, two advanced high-temperature heat pump systems will be developed. The first system is a closed loop cycle based on the low global warming potential (GWP) refrigerant HFO-1336mzz-Z with good thermodynamic properties at the identified drying temperatures. Thus, the closed loop heat pump system is suitable for supply temperatures up to 160°C.[1] The closed loop cycle is for convective air drying and will be integrated into the dryers for starch (Agrana Stärke GmbH) and bricks (Wienerberger AG). The second heat pump system is an open loop cycle, where water (R718) is the refrigerant. Such systems are commonly referred to as MVR (Mechanical Vapour Re-compression). In this system, the heat pump cycle directly utilizes the excess steam from the dryer. The open loop cycle for convective superheated steam drying will be integrated into dryers for sludge (Scanship ASA) using a novel turbo-compression technology.

The goal of the integration of the novel high-temperature heat pump systems in the above industrial drying processes is to recover the waste heat streams so that the specific energy demand is reduced by as much as 60-80%. In this deliverable, key performance indicators are specified in order to validate the performance of the novel heat pump systems to be demonstrated. **These KPIs are the Coefficient of Performance (COP) of the heat pumps, the Specific Energy Consumption (SEC) of the drying system, the Sensible Heat Recovery (SHR) and the Reheating Ratio (RHR). Based on these indicators it is possible to determine the CO_2 emissions avoided in the integrated heat pump dryers, the reduction in primary energy consumption, the increase in energy efficiency as well as the competitive advantage in the form of reduced production costs**.

The data necessary to calculate the different performance indicators will be made available through measured process data at the demonstration sites and benchmarked against the drying system without heat pump. Relevant process parameters are the drying temperature of the dryer which significantly influences the product quality and the production cycle time. Further relevant parameters are the mass flow rates and moisture content (both in and out) of the products to be dried, and the flow rates of the drying agent (air or steam) of the system. A collective establishment of the key performance indicators has been performed for all three end users of the project (Agrana Stärke GmbH, Wienerberger AG and Scanship Holding ASA).

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[1] Fleckl, T., Hartl, M., Helminger, F., Kontomaris, K. and Pfaffl, J., Performance testing of a lab-scale high temperature heat pump with HFO-1336mzz-Z as the working fluid, in European Heat Pump Summit 2015, Nuremberg, Germany, October 20-21

1 DRYING

Drying is one of the oldest preservation technologies and made it possible for humans to store food for extended periods of time. Thus, drying has been a crucial technology in the evolution of human cultures. To this day drying is still one of the dominating industrial preservation processes for innumerable products. Industrialization helped to optimize processes of drying, which are conducted under varying, but controlled conditions. The Handbook of Industrial Drying[2] gives at least 15 different dryer types and identifies more than 20 different industrial drying sectors, which makes it rather difficult to generalize drying technologies. For every product a certain drying process is characterized by the type of dryer, the quality of the final product, the extended product shelf-life, additional needed processing, the ease of handling, sanitation, cost/energy effectiveness, and investment. However, the basic principle of drying is the same as it used to be thousands of years ago, and the most common dryer type is based on convective drying.

In drying technology, the term convective drying is used to indicate when heat and mass transfer are due to the temperature and pressure gradients, respectively, between the drying agent (DA) and the drying product. The molecules that are moved (convected) are the water molecules that diffuse into the drying agent. Convective drying, based on air as the drying agent, is the mostly used drying process, and globally more than 85% of all dryers are of a convective air-drying type.[3] However, also other drying agents like superheated steam are used in convective dryers, because of improved thermal properties and/or drying potential.

The purpose of the drying agent in convective drying is to supply the necessary energy for the evaporation of the moisture from the product, capture the evaporated water and remove it from the drying system. The speed at which the moisture is removed is commonly referred to as the drying rate.

The drying rate is a function of the heat transfer between the drying agent and the water if the moisture of the product is present as surface water. In this case, the drying rate is constant up to the point when a dry layer develops in the product. The (growing) dry layer poses an additional resistance to the heat and mass transfer and results in a reduced drying rate, as illustrated in Figure 1 and Figure 2.

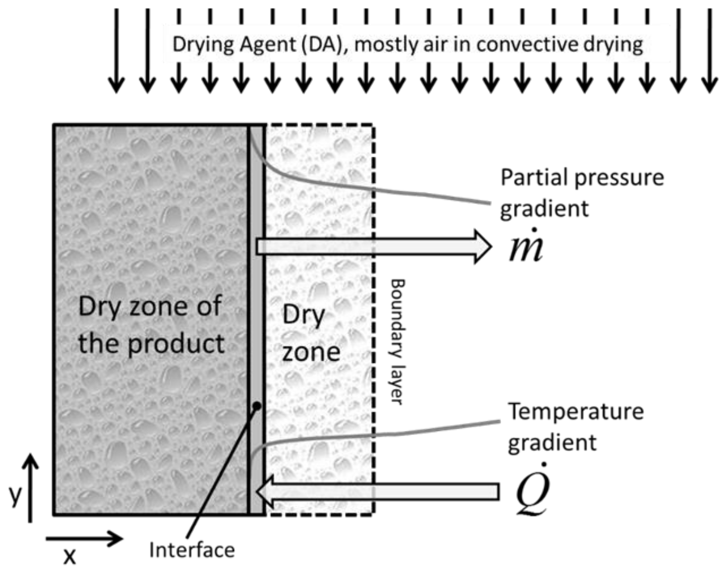

Figure 1: General heat and mass transfer in convective drying of products.

[2] Mujumdar, A.S. (2007). Handbook of Industrial Drying. Boca Raton, Fla., Taylor & Francis

[3] Atuonwu, J.C., et al., Reducing energy consumption in food drying: Opportunities in desiccant adsorption and other dehumidifaction strategies (2011)

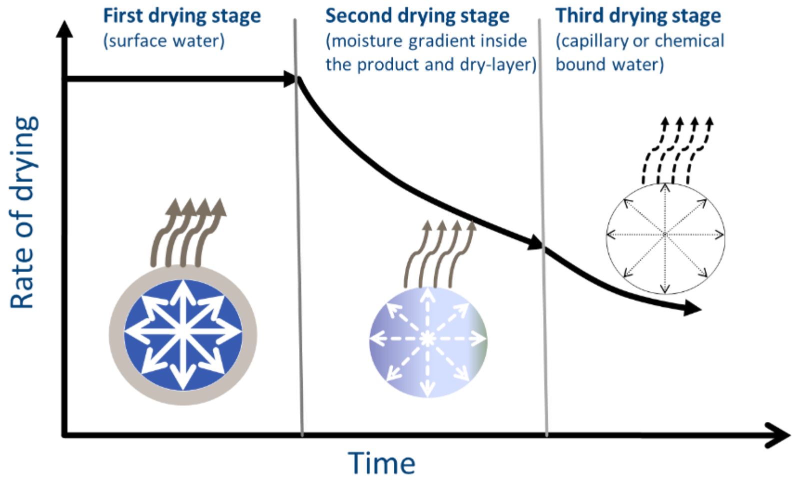

Figure 2: Characteristic different drying stages with constant and falling drying rate periods.

For batch dryers this normally results in falling drying rates with the drying process. For continuously operating dryers, the drying rate of the system appears constant even when the product is in the falling drying rate period.

The mass transfer in drying is achieved by evaporating the moisture of the product and let it diffuse into the drying agent. In other words, the water to be dried out needs to undergo a phase change from liquid to vapour in order to be removed from the product. The necessary energy for this phase change is the evaporation energy, which is the minimum required thermal energy in order to enable convective drying. For water, the evaporation energy is 2258 kJ/kg (at 100°C) or 0.627 kWh/kg. This is the ideal theoretical minimum required thermal energy for any convective drying system. Real operational dryers require additional energy to overcome heat and mass transfer losses, product specific moisture binding resistance, thermal heat losses and others (see Figure 3). The sum of all required and supplied thermal energies are commonly referred to as the drying energy.

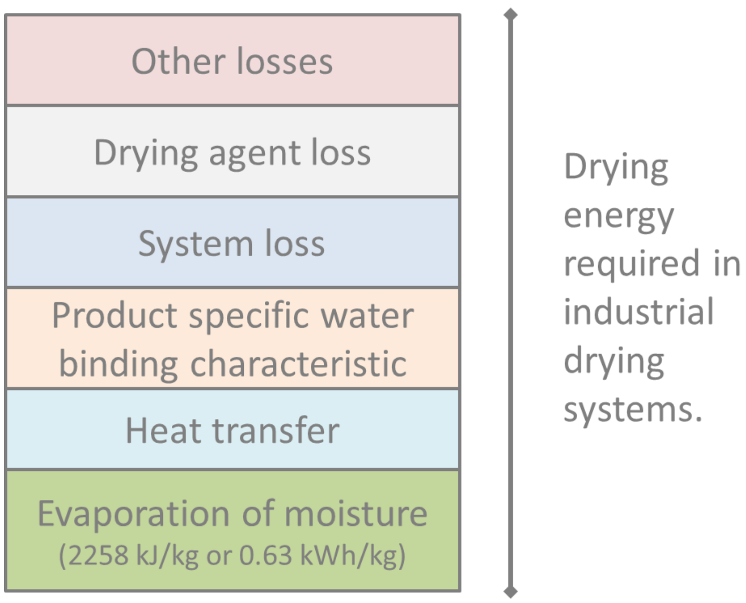

Figure 3: Drying energy in real-life convective drying systems.

The drying energy in convective drying systems is supplied as **sensible heat by the drying agent**. In most drying applications this energy is supplied by simple heating devices which are used to heat up the drying agent so that it contains the necessary sensible heat before drying. The sensible heat of the drying agent is then used to dry the product. The drying agent is consequently cooled down during convective drying and the sensible heat is transferred into water vapour and removed from the system. In other words, the drying agent is hot and dry when it enters the drying system and cold and moist when it leaves the drying system. In order to reuse the drying agent again at the inlet of the dryer, the drying agent needs to be reheated and combined with a supply of heated fresh drying agent as needed. This is illustrated in the simplified system solution in Figure 4. **It is important to notice that the supplied drying energy, *i.e.*, the sensible heat, leaves the system and can be recovered in the form of water vapour in the drying agent.**

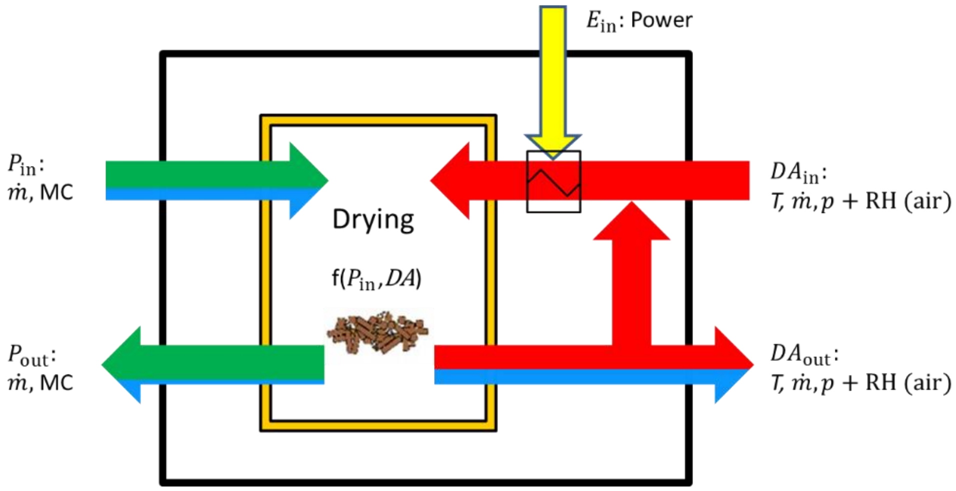

Figure 4: Schematic diagram of conventional drying process.

The specific energy consumption of industrial drying system varies significantly for different products and drying types. For convective dryers, specific energy consumptions of 0.8 kWh/kg removed water are considered as highly energy-effective drying systems. However, most industrial dryers have a specific energy consumption of higher than 1 kWh/kg and an average drying efficiency of 35-40%.[4]

[4] Colak N. and Hepbasil, A., A review of heat pump drying: Part 1 – Systems, models and studies, *Energy Conversion and Management* 50, 2180-2186 (2009)

2 KEY PERFROMANCE INDICATORS FOR HEAT PUMPS

Heat pumps are characterized by their possibility to deliver thermal energy to a heat sink at a higher temperature than the heat source. The temperature increase between the heat source and the sink is enabled by compressing the working media. The required energy input to a heat pump system is normally in the form of electric power to the compressor. The advantage of a heat pump is that it can deliver a higher amount of thermal energy than the supplied (electric) energy to the compressor.

For heat pumps in general the most common key performance indicator is the Coefficient of Performance (COP). The COP of one-stage and two-stage compression heat pumps is defined in the following sub-chapters and will be used as KPI of the integrated heat pump system.

2.1 Heat pump with one-stage compression[5]

HFO-1336mzz-Z (Opteon™ MZ) will be used as the refrigerant for the closed loop systems. It has been developed to deliver heating temperatures in the range of 100°C to 150°C, and is non-flammable and remains chemically stable at high temperatures

The main components of a heat pump are shown in Figure 5 (left), where a schematic diagram of the flow chart of a one-stage compression heat pump system is given using Opteon™ MZ as the working fluid in a closed-loop heat pump. Figure 5 (right) shows the corresponding pressure-enthalpy (p-h) diagram. In the given example, the working fluid is evaporated at 38°C, with a heat source inlet temperature of 70°C. The heat sink is heated up from 85°C to 125°C, where the working fluid is condensed at 127°C.

The heat pump consists of four main components, namely the compressor, the condenser, the expansion valve, and the evaporator. The working fluid of the heat pump evaporates at low pressure at T_{Evap} = 38ºC. It is then superheated to 65 ºC, also occurring in the evaporator. The superheated working fluid is then compressed to the high pressure level. In the condenser, the working fluid is condensed at T_{cond} = 127ºC. In order to close the heat pump loop, the working fluid undergoes an expansion back to the low evaporation pressure. The temperature lift between the evaporator and the condenser for this one-stage compression heat pump is $\Delta T = T_{cond} - T_{evap}$ = 89 K.

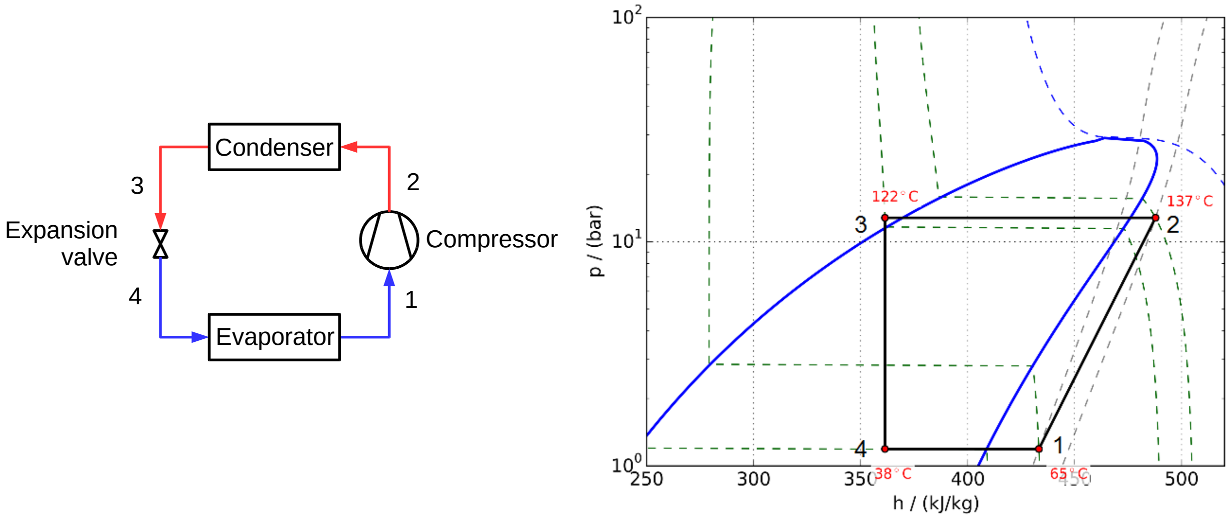

Figure 5: Schematic of the main components and the flow diagram of a one-stage compression heat pump (left) and the corresponding pressure-enthalpy diagram (right).

[5] Applied in the drying systems of Agrana Stärke and Wienerberger.

The heat pump cycle consists of the following processes:

1 – 2: Compression of the superheated working fluid in the Compressor

2 – 3: Isobaric cooling, condensation and subcooling of the working fluid in the Condenser

3 – 4: Isenthalpic expansion in the Expansion valve

4 – 1: Isobaric evaporation and superheating of the working fluid in the Evaporator

The pressure-enthalpy diagram exposes a relatively low evaporation temperature of 38°C compared to the source inlet temperature of 70°C. This is caused by the shape of the vapor dome as seen in Figure 5 (right). In order to ensure a safe and reliable compression cycle, the working fluid has to be superheated sufficiently. This can lead to low evaporation temperatures when the superheating is realized by the evaporator, thus causing high temperature lifts ΔT.

In order to prevent low evaporation temperatures, a suction gas heat exchanger – or internal heat exchanger (IHX) - can be used to superheat the suction gas with the outlet liquid phase after the condenser, as seen in Figure 6 (left). The corresponding pressure-enthalpy (p-h) is shown in Figure 6 (right). In this case, an evaporating temperature of $T_{Evap} = 62$ºC can be achieved for the same source conditions as before. The temperature lift between the evaporator and the condenser for this one-stage compression heat pump with suction gas heat exchanger is $\Delta T = T_{cond} - T_{evap} = 65$ K.

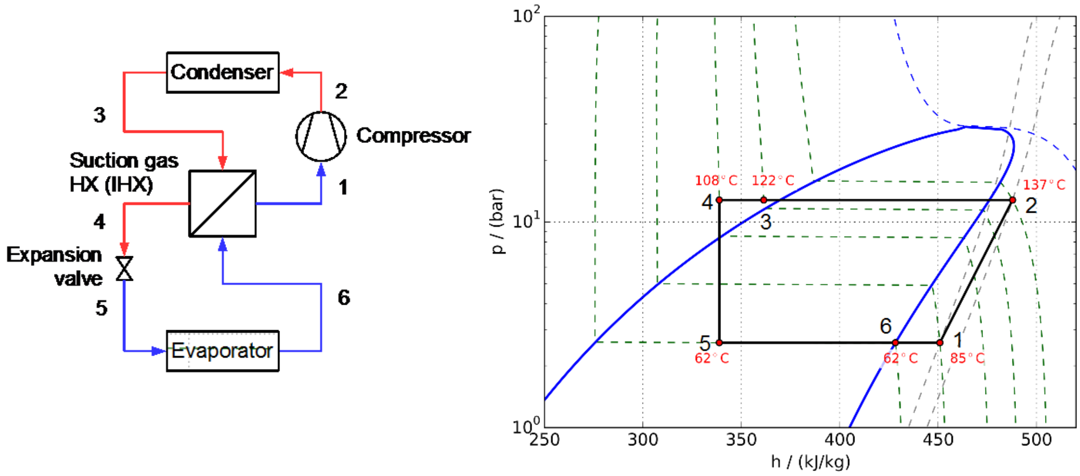

Figure 6: Schematic of the main components and the flow diagram of a one-stage compression heat pump with suction gas heat exchanger IHX (left) and the corresponding pressure-enthalpy diagram (right).

The heat pump cycle consists of the following processes:

1 – 2: Compression of the superheated working fluid in the Compressor

2 – 3: Isobaric cooling, condensation and subcooling of the working fluid in the Condenser

3 – 4: Isobaric subcooling in the suction gas superheater (IHX)

4 – 5: Isenthalpic expansion in the Expansion valve

5 – 6: Isobaric evaporation a of the working fluid in the Evaporator

6 – 1: Isobaric superheating of the working fluid in the suction gas superheater (IHX)

In the assessment of heat pumps, the most used performance parameter is the Coefficient of Performance (COP). The COP of a heating system is

$$COP_{HP} = \frac{\dot{Q}_H}{W}, \quad (1)$$

where $\dot{Q}_H$ [kW] is the heat delivered by the heat pump and W [kW] is the total energy consumption.

The total energy consumption for the one-stage compression heat pump cycle is

$$W = W_{\mathrm{compr}}, \tag{2}$$

where W_{compr} is the work input to the heat pump compressor.

2.2 Two-stage compression with intercooling

R718 (water) will be used as refrigerant for the open cycle integrated heat pump of the superheated steam dryer and the used novel turbo-compressor can only achieve a limited pressure ratio. In order obtain higher pressure (and hereby temperature) lifts between the evaporator and condenser, multi-stage compression is employed. Figure 7 shows a schematic flow chart of the two-stage compression heat pump system, based on the thermal properties of R718 (water). The main components in this system are the two compressors, the evaporator, the condenser, the intermediate pressure vessel and the two expansion valves. In the evaporator, steam is generated at $T_{evap} = 100$ºC and atmospheric pressure ($p_0 = 1$ bar). The steam is compressed in Compressor 1 to an intermediate state of higher pressure ($p_1 = 2.35$ bar). During the compression the working media superheated. The superheated steam is then cooled to 125ºC in the intermediate pressure vessel, before being compressed in Compressor 2 to pressure of $p_2 = 5.4$ bar. Subsequently, in the Condenser the steam is condensed at $T_{cond} = 155$ºC and. Finally, the steam undergoes an adiabatic expansion with a pressure and temperature drop back to 2.3 bar and 125ºC in Expansion Valve 1, and then another adiabatic expansion with a pressure drop back to $p_0 = 1$ bar and a steam temperature at 100ºC in Expansion Valve 2. In this case the temperature lift between the evaporator and the condenser is $\Delta T = T_{cond} - T_{evap} = 55$ºC. The function of the intermediate pressure vessel is to intercool the working fluid after the compressor and to separate the liquid phase (water) before expansion from the steam which is compressed further. It is important to notice that the mass flow of steam in Compressor 2 is higher than in Compressor 1 due to the function of the intermediate pressure vessel.

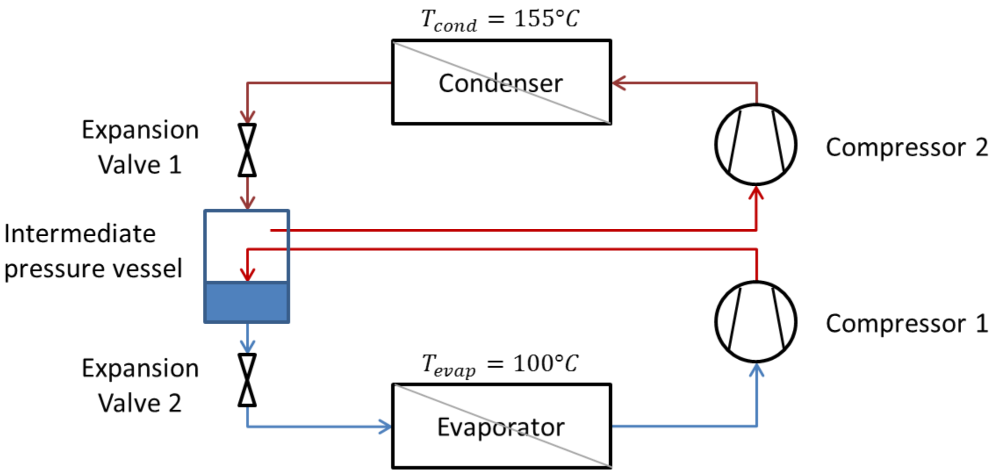

Figure 7: Flow diagram of a two-stage compression heat pump.

Figure 8 and Figure 9 show the P-h and T-S diagrams, respectively, of the two-stage compression of superheated steam with intercooling. The heat pump cycle consists of the following processes:

1 – 2: Isobaric evaporation and superheating of the steam in the Evaporator to 110ºC

2 – 3: Isentropic compression of the superheated steam in Compressor 1

3 – 4: Isobaric cooling of the superheated steam to saturated steam

4 – 5: Isentropic compression of the steam to superheated steam in Compressor 2

5 – 6: Isobaric cooling and condensation of the steam to 155ºC in the Condenser

6 – 7: Adiabatic expansion in Expansion Valve 1

7 – 8: Isobaric cooling of the steam
8 – 1: Adiabatic expansion in Expansion Valve 2

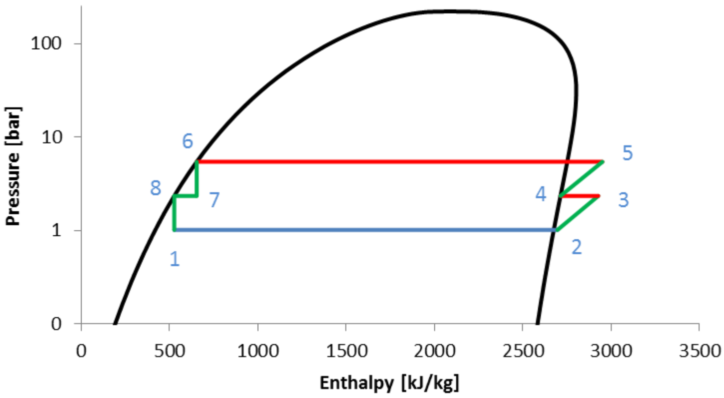

Figure 8: P-h diagram of the two-stage compression heat pump cycle with intercooling.

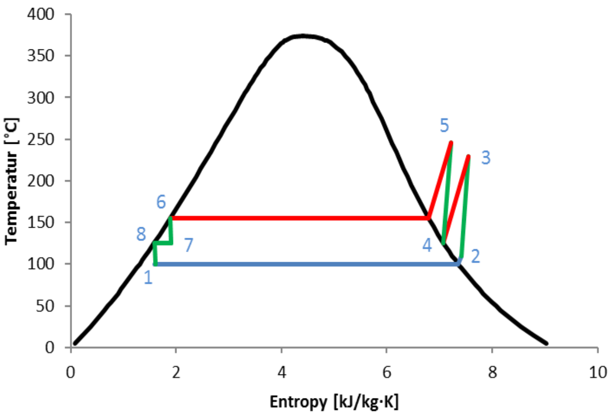

Figure 9: T-S diagram of the two-stage compression heat pump cycle with intercooling.

The COP for the two-stage compression heat pump is

$$COP_{HP} = \frac{\dot{Q}_H}{W}, \tag{3}$$

where $\dot{Q}_H$ [kW] is the heat delivered by the heat pump and W [kW] is the total energy consumption. The total energy consumption for the two-stage compression heat pump cycle is

$$W = W_{c1} + W_{c2}, \tag{4}$$

where W_{c1} and W_{c2} is the work input to Compressor 1 and 2, respectively.

3 KEY PERFORMANCE INDICATORS FOR HEAT PUMP DRYERS

Modern industrial drying processes are either based on open-loop systems using heated ambient air or on closed-loop systems with recirculation of the drying air. As outlined above, heat pumps are characterized by the possibility to utilize a heat source at low temperature (at the evaporator) and supply a heat sink at a higher temperature (condenser). In the case of closed-loop drying, this combined heat and cool load is used in the recovery of the drying energy (basically the latent heat of evaporation of water) and to deliver this energy back into the drying process in the form of de-humidified and re-heated drying air (as illustrated in Figure 10).

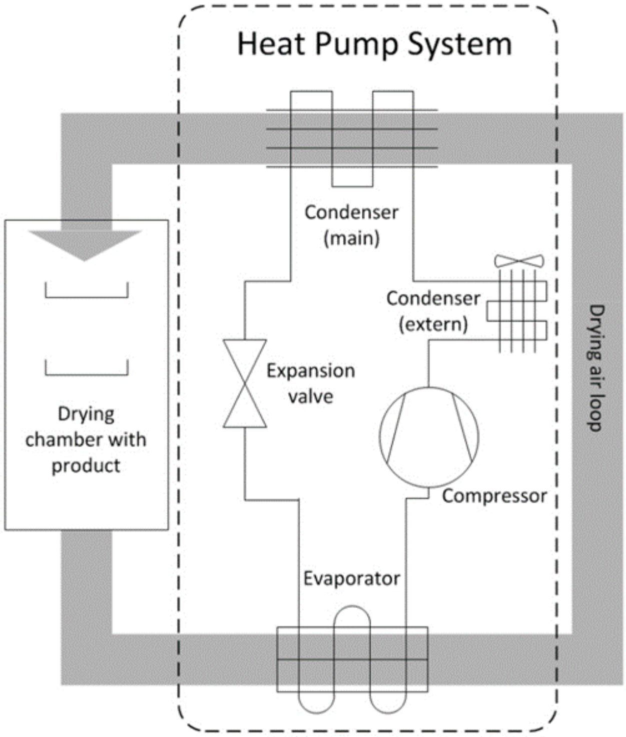

Figure 10: Principal layout for heat pump assisted drying for convective (air) drying in closed loop.

Heat pump drying consists of two loops: one loop for the drying agent (mostly air) and one refrigeration cycle. At the evaporator of the heat pump, the drying air is cooled down below the dew point and the moisture from the air is condensed at the surface of the heat exchanger. Energy is hereby transferred to the refrigerant, which is evaporated. The evaporated refrigerant is then compressed and can now be condensed in the condenser at higher temperature. Hereby, the formally transferred energy is given back to the drying air which is then reheated to its initial desired condition. It is necessary in some cases to install a second external condenser in order to transfer the excess heat out of the system. The external condenser can either be installed in series or in parallel to the main-condenser. Serial condenser installation requires stable drying and operational conditions and the capacity is fixed to the design point. Parallel condensers are more flexible with respect to load profiles and operational but require the installation and control of a 3-way-valve.

The main source for the excess energy is the compressor, which also should be equipped with a rotation speed control in order to ensure optimum working conditions at varying heat and cooling loads. It is also recommended to install a bypass valve for the drying air, so that only the necessary amount of drying air is cooled and re-heated; this makes the operation more efficient.[6]

[6] Petrova, I., M. Bantle and T. Eikevik (2015). "Manufacture of dry-cured ham: A review. Part 2. Drying kinetics, modeling and equipment." European food research and technology: 1-12.

The principle can be also used for open loop drying systems when the ambient air is heated to the initial drying temperature with help of the excess heat from the cold and humid drying air which is exhausted form the drying chamber. However HPD for open drying loops have no possibility to control the humidity of the drying air and most systems are therefore based on closed loop drying.

The key performance indicators are specified in order to validate the performance of the novel heat pump systems to be demonstrated. A total of eight performance indicators measure environmental impact and economic advantages achieved by the heat pumps. Important KPIs are for example the Coefficient of Performance (COP) of the heat pumps and the Specific Energy Consumption (SEC).

For superheated steam drying systems the dryer itself can function as the evaporator of the heat pump as soon as R718 (water) is used as refrigerant. In this case the excess superheated steam is compressed directly after the outlet of the dryer and the circulated drying agent (steam) is reheated in the condenser while the condensate is removed from the system.

3.1 Coefficient of Performance (COP) of heat pump dryers

The COP of a heat pump used in a drying process also includes the total energy consumption of the secondary installations and is determined as follows:

$$COP_{\mathrm{HPD}} = \frac{\dot{Q}_{cond}}{W_{\mathrm{tot}}}, \tag{5}$$

where $\dot{Q}_{\mathrm{cond}}$ [kW] is the heat delivered at the condenser and W_{tot} [kW] is the total energy consumption for the heat pump dryer. The total energy consumption is given by

$$W_{\mathrm{tot}} = W_{\mathrm{compr}} + W_{\mathrm{fan}} + W_{\mathrm{pumps}} \tag{6}$$

where W_{compr} is the work input to the heat pump compressor(s), W_{fan} is the work input to the additional fan/blower unit needed to provide a certain velocity to the drying agent (air/steam) and W_{pumps} is the work input to the additionally needed pumps of e.g. secondary installations.

3.2 Specific energy consumption (SEC) of heat pump dryers

For the drying process, we define the thermal efficiency η_{thermal} [kg water/kJ] as

$$\eta_{\mathrm{thermal}} = \frac{\Delta x}{\Delta h} \tag{7}$$

where Δx [kg] is the amount of removed water from the product and Δh [kJ] is the amount of energy consumed. Here, for the heat pump dryer system, Δh equals heat delivered at the condenser, *i.e.,* $\Delta h = Q_{\mathrm{cond}}$.

A commonly used performance parameter for the combined heat pump dryer system is the Specific Moisture Extraction Rate (SMER). The SMER number [kg/kWh] is defined as the product of the heat pump COP and the thermal efficiency η_{thermal}, that is,

$$SMER = COP_{\mathrm{HPD}} \cdot \eta_{\mathrm{thermal}} \tag{8}$$

Using the relation $\Delta h = Q_{\text{cond}}$, we find that

$$SMER = \frac{\Delta x}{W_{\text{tot}}} \tag{9}$$

and, hence, the SMER is defined as the mass [kg] of removed water from the product divided by the required energy [kWh] for this. In other words, the SMER gives a key specification of the drying efficiency of the heat pump dryer system. Alternatively, the Specific Energy Consumption (SEC), which is the reciprocal of the SMER, is used to compare the energy efficiency of various heat pump dryers. In this work we use the SEC [kWh/kg] as the key performance parameter, which is defined as

$$SEC = \frac{1}{SMER} = \frac{W_{\text{tot}}}{\Delta x} \tag{10}$$

Hence, the SEC gives the required energy input per kg of evaporated water in the dryer. For some heat pump dryer systems, external heat is used to replace part of the work input W_{tot}. The SEC can then be expressed as

$$SEC = \frac{W_{\text{tot}} + Q_{\text{ext}}}{\Delta x} \tag{11}$$

where $W_{\text{tot}} = \sum W_i$ is the sum of work input to the compressor, fan, pumps, etc. and Q_{ext} is the external heat input.

3.3 Sensible Heat Recovery (SHR) of heat pump dryers

In order to evaluate how much sensible heat is recovery from each waste heat carrier addressed in the project, the following ratio is calculated:

$$SHR = \frac{\dot{m}_{\text{WH}} c_{\text{p,WH}} (T_{\text{WH,in}} - T_{\text{WH,out}})}{\dot{m}_{\text{WH}} c_{\text{p,WH}} (T_{\text{WH}} - T_0)} \tag{12}$$

where $\dot{m}_{\text{WH}}$ is the mass flow of the waste heat steam, $T_{\text{WH,in}}$ is its temperature before recovery, $T_{\text{WH,out}}$ after recovery and T_0 is the lowest useful temperature for the stream. The lowest useful temperature of the waste stream is defined by space heating, when the waste stream is used to supply warm air at 20°C (room temperature). To account for the temperature difference in the heat exchanger, T_0 is set to 30°C. As the heating capacity $c_{\text{p,WH}}$ is approximately independent from temperature, the SHR can be expressed by a ratio of temperatures as following

$$SHR = \frac{T_{\text{WH,in}} - T_{\text{WH,out}}}{T_{\text{WH,in}} - T_0} \tag{13}$$

$T_{\text{WH,out}}$ is the temperature of the waste heat stream after recovery. SHR will be used to document Impact 1 of the project: "*Recovery of at least 40% of the sensible heat contained in each waste heat carrier*".

3.4 Reheating Ratio (RHR) of heat pump dryers

Reheating Ratio (RHR) describes how much energy is recovered from the heat source as a function of the supplied energy for re-heating the drying agent by the heat pump. RHR is defined as:

$$RHR = \frac{\dot{Q}_{\mathrm{DA,reheat}} - (W_{\mathrm{tot}} + \dot{Q}_{\mathrm{ext}})}{\dot{Q}_{\mathrm{DA,reheat}}} \qquad (14)$$

where $\dot{Q}_{\mathrm{DA,reheat}}$ is the heat delivered to the drying agent, W_{tot} is the total work input to the system, and $\dot{Q}_{\mathrm{ext}}$ is eventual external heat input needed to run the process. The delivered sensible heat is defined as:

$$\dot{Q}_{\mathrm{DA,reheat}} = \dot{m}_{\mathrm{DA}} c_{p,\mathrm{DA}} \Delta T_{\mathrm{reheat,DA}} \qquad (15)$$

where $\dot{m}_{\mathrm{DA}}$ is the mass flow rate of the drying agent into the dryer, $c_{p,\mathrm{DA}}$ is the specific heat of the drying agent, and $\Delta T_{\mathrm{reheat,DA}}$ is the temperature difference of the drying agent before and after the reheat. RHR will be also used to document Impact 1 of the project: "*Recovery of at least 40% of the sensible heat contained in each waste heat carrier*".

3.5 Reduction of CO_2 emissions by heat pump dryer

The key performance indicator used to determine the CO_2 emissions avoided in the heat pump dryer systems will be based on the SEC of the dryer itself (today's system) and the SEC of the heat pump dryer:

$$CO_{2,\mathrm{avoided}} = (SEC_{\mathrm{dryer,initial}} EF_{\mathrm{initial}} - SEC_{\mathrm{HPD}} EF_{\mathrm{HPD}}) \dot{m}_{\mathrm{water,removed}} \qquad (16)$$

where EF is the CO_2 Emission Factor of the energy source of the dryer before (initial) and after the installation of the heat pump (HPD). EF_{initial} [CO_2/kWh] will be primarily based on general emissions related to fossil fuels and EF_{HPD} [CO_2/kWh] will be related to the emission related to electricity production at the respective production sites. Thus, the reduction of CO_2 emissions are documented for each production site (Impact 2 of the project).

3.6 Primary energy savings (PES) by heat pump dryer

The key performance indicator used to determine the primary energy savings in the heat pump dryer systems will be based on the SEC of the dryer itself (today's system) and the SEC of the heat pump dryer:

$$PES = (SEC_{\mathrm{dryer,initial}} PEF_{\mathrm{initial}} - SEC_{\mathrm{HPD}} PEF_{\mathrm{HPD}}) \dot{m}_{\mathrm{water,removed}} \qquad (17)$$

where PEF is the primary energy factor of the energy source of the dryer before (initial) and after the installation of the heat pump (HPD). PEF_{initial} [kWh/kWh] will be based on primary energy consumption of fossil fuels used to operate the initial drying system and PEF_{HPD} [kWh/kWh] will be related to the electricity production at the respective production sites. Thereby, the primary energy savings are documented for each production site (Impact 2 of the project).

3.7 Improvement of energy efficiency (EE) by heat pump dryer

The main improvement of the heat pump drying system will be an increase in energy efficiency, while simultaneously switching from fossil fuel based direct energy supply to electric energy supplied by a national power grid.

Improvement of energy efficiency will be calculated from the SEC of the dryer itself (today's system) and the SEC of the heat pump dryer. Two different ratios are defined. For the energy efficiency of the system EE_{sys}, the initial SEC of the dryer is used as the reference:

$$EE_{\mathrm{sys}} = \frac{SEC_{\mathrm{dryer,initial}} - SEC_{\mathrm{HPD}}}{SEC_{\mathrm{dryer,initial}}} \cdot 100\% \tag{18}$$

To calculate the increase of energy efficiency by the heat pump EE_{HP}, the energy consumption of the heat pump is compared to the consumption of fossil fuel in the initial system to provide the heating capacity of the heat pump $\dot{Q}_{\mathrm{cond}}$. Thereby, also the efficiency of the fossil heat supply system is required ($\eta_{\mathrm{heat,initial}}$):

$$EE_{\mathrm{HP}} = \frac{\frac{\dot{Q}_{\mathrm{cond}}}{\eta_{\mathrm{heat,initial}}} - W_{tot}}{\frac{\dot{Q}_{\mathrm{cond}}}{\eta_{\mathrm{heat,initial}}}} \cdot 100\% \tag{19}$$

The improvement of energy efficiency is evaluated for each demonstration site (Impact 3 of the project).

3.8 Advancement in competitiveness

In close relation to the improvement of energy efficiency, the heat pump dryers allow for cost reduction for the products to be dried. The advancement in competitiveness will be evaluated in the form of reduced Production Costs (PC) that can be determined by the SEC:

$$PC_{\mathrm{reduced}} = \left(SEC_{\mathrm{dryer,initial}} EP_{\mathrm{fossil}} - SEC_{\mathrm{HPD}} EP_{\mathrm{electricity}}\right) \dot{m}_{\mathrm{water,removed}} \tag{20}$$

where *EP* is the energy price of the production site for fossil fuels and electricity and the mass flow of removed waters ($\dot{m}_{\mathrm{water,removed}}$) is the drying capacity of the single drying units.

For Impact 3, relative cost reduction is required, which is calculated in analogy to the improvement of energy efficiency either referring to the whole system RCR_{sys} or to the heat pump RCR_{HP}.

$$RCR_{\mathrm{sys}} = \frac{\left(SEC_{\mathrm{dryer,initial}} EP_{\mathrm{fossil}} - SEC_{\mathrm{HPD}} EP_{\mathrm{electricity}}\right)}{SEC_{\mathrm{dryer,initial}} EP_{\mathrm{fossil}}} \cdot 100\% \tag{21}$$

$$RCR_{\mathrm{HP}} = \frac{\frac{\dot{Q}_{\mathrm{cond}}}{\eta_{\mathrm{heat,initial}}} EP_{\mathrm{fossil}} - W_{tot} EP_{\mathrm{electricity}}}{\frac{\dot{Q}_{\mathrm{cond}}}{\eta_{\mathrm{heat,initial}}} EP_{\mathrm{fossil}}} \cdot 100\% \tag{22}$$

4 CASE SPECIFIC DETERMINATION OF KPI'S

In the following sub-sections the determination of the KPIs as described in Chapters 2 and 3 is discussed for each individual case. These KPIs are strongly related to the expected impacts of DryFiciency:

- **Impact 1**: **Recovery** of at least **40% of the sensible heat** contained in each waste heat carrier addressed by the project.
- **Impact 2**: Measureable **substantial primary energy savings** clearly quantified and substantiated, and subsequent **reduction of CO_2 emissions**.
- **Impact 3**: The improvement of the energy efficiency and the reduction of energy cost will lead to **a demonstrated advancement in competitiveness** by the end of the project. This will expand the available portfolio of energy resources and technologies, which can be integrated within sites, across sectors and along value chains.

4.1 Heat pump assisted drying of starch

The starch drying process in DryFiciency is a continuous process with an integrated closed loop heat pump system, as seen in Figure 11. The drying agent (air) is preheated by a water-to-air heat exchanger from a heat recovery cycle with water as heat transfer fluid. After this initial preheating, the water also serves as the source for the evaporator of the heat pump system. The inlet temperature is then ≈70°C. At the condenser side, high temperature heat is released to the drying agent via an intermediate water circuit and a water-to-air heat exchanger. The heat supply temperature of the heat pump is up to 160°C and is measured at the outlet of the condenser in the intermediate water circuit. As a result of these consecutive preheating steps of the drying agent, less energy is needed in the steam generator to reach a desired temperature level of ≈160°C at the inlet of the flow stream dryer.

In order to determine the KPIs, the following parameters need to be measured:

- Drying agent:
  - Volumetric flow
  - Relative humidity (RH) and temperature before the initial preheating
  - Temperature at the dryer inlet
  - RH and temperature at the dryer outlet
- Steam:
  - Mass flow
- Heat pump
  - Electric power (compressor work)
  - Secondary side mass flows (water mass flow on evaporator and condenser side)
  - Secondary side inlet and outlet temperatures (water temperatures at inlet and outlet of evaporator and condenser)
  - The water flow on the secondary side of the evaporator is the waste heat stream used at this site
- Product
  - Mass flow

The following parameters are either calculated from measured values or are provided from different sources:

- Product
  - Moisture extraction: The amount of water removed from the product will be determined by the mass balance of the drying agent. The inlet and outlet conditions of the drying agent are measured (volumetric flow, temperature and relative humidity). Thus, moisture extraction can be calculated online.
- Production site:
  - Energy price for fossil fuel and electricity [€/kWh] will be provided by the production site

- CO_2 emissions factor [CO_2/kWh] and primary energy factor [kWh/kWh] for fossil fuel and electricity will be determined for the country, in which the production site is located (retrieved from current studies and reports).

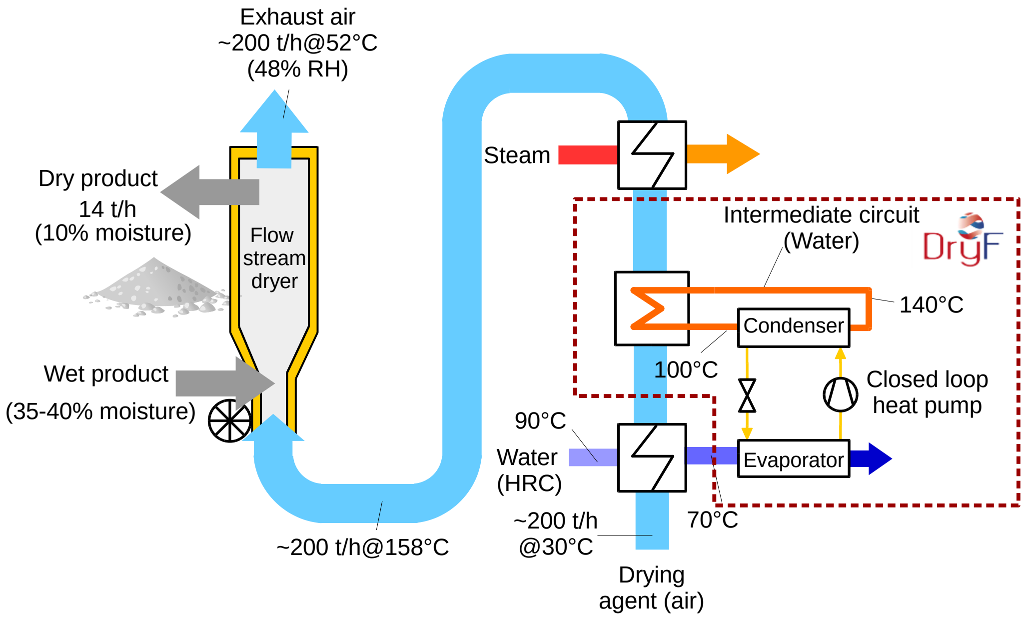

Figure 11: Schematic diagram of the heat-pump assisted starch dryer. The drying agent (air) is preheated in three consecutive steps: Initially, by an existing water heat recovery cycle. Secondly, by the intermediate circuit which is heated up by the heat pump condenser. At last, the steam heat exchanger brings the drying agent to its final temperature of ≈160°C.

The required measurement points are illustrated (green color) in Figure 12.

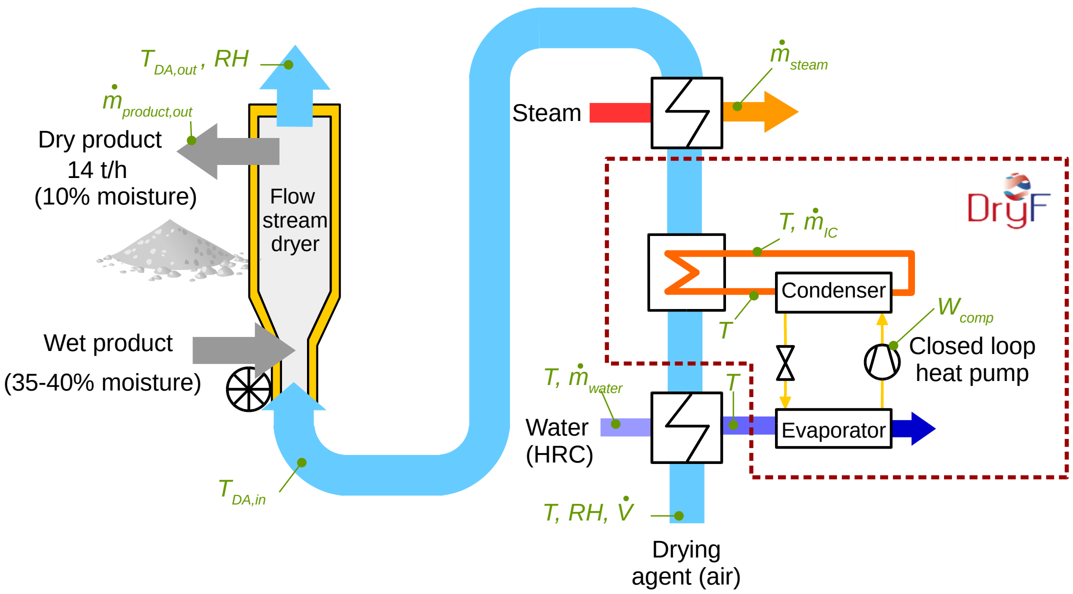

Figure 12: Schematic diagram of the required measurements (in green) in order to determine the KPIs of the closed-loop heat pump system for the flow stream dryer at Agrana.

4.2 Heat pump assisted drying of bricks

The brick drying process in DryFiciency is a continuous tunnel dryer that is illustrated in Figure 13. Air is used as drying agent that flows counter-currently to the bricks. Bricks enter the dryer with 28% moisture and are dried to 2%. Drying air in the tunnel is heated by internal heat exchanger surfaces, which are supplied with water with 90°C by a heat recovery cycle. The heat pump also uses the heat recovery cycle as the heat source. The evaporator is inserted before the heat exchangers. The heat pump provides hot air via an intermediate circuit, heat supply temperatures up to 160°C can be reached there. The hot air is fed into the outlet zone of the tunnel dryer, where the highest temperatures are required. The heat pump acts as a booster for the heat recovery cycle.

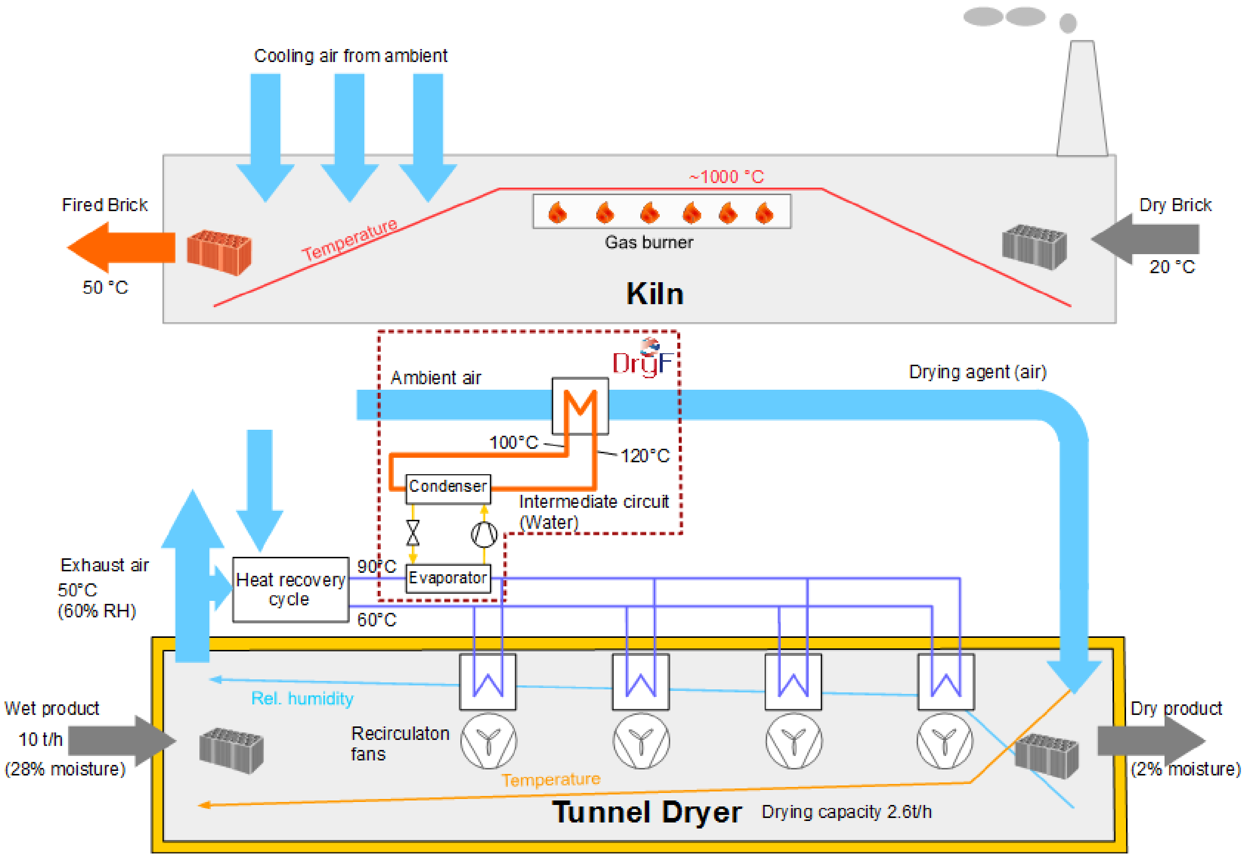

Figure 13: Schematic diagram of the heat pump integration at the Wienerberger tunnel dryer.

An appropriate determination of the KPIs as shown in Chapter 2 and 3 needs the following measurements:

- Tunnel dryer/kiln:
  - Gas flow meter
  - Volumetric flow, RH and temperature
    - Drying agent (air)
    - Exhaust air (waste heat stream 1)
- Heat recovery cycle
  - water mass flows and temperatures before and after the heat exchanger surfaces
- DryFiciency heat pump:
  - Electric power (compressor work)
  - Secondary side mass flows (water mass flow on evaporator and condenser side)
  - Secondary side inlet and outlet temperatures (water temperatures at inlet and outlet of evaporator and condenser)
- Product
  - Mass flow of bricks

The measurement points are illustrated (green color) in Figure 14.

The following parameters are either calculated from measured values or are provided from different sources:

- Product
  - Moisture extraction: The amount of water removed from the product will be determined by the mass balance of the drying agent. The inlet and outlet conditions of the drying agent are measured (volumetric flow, temperature and relative humidity). Thus, moisture extraction can be calculated online.
- Production site:
  - Energy price for fossil fuel and electricity [€/kWh] will be provided by the production site
  - CO_2 emissions factor [CO_2/kWh] and primary energy factor [kWh/kWh] for fossil fuel and electricity will be determined for the country, in which the production site is located (retrieved from current studies and reports).

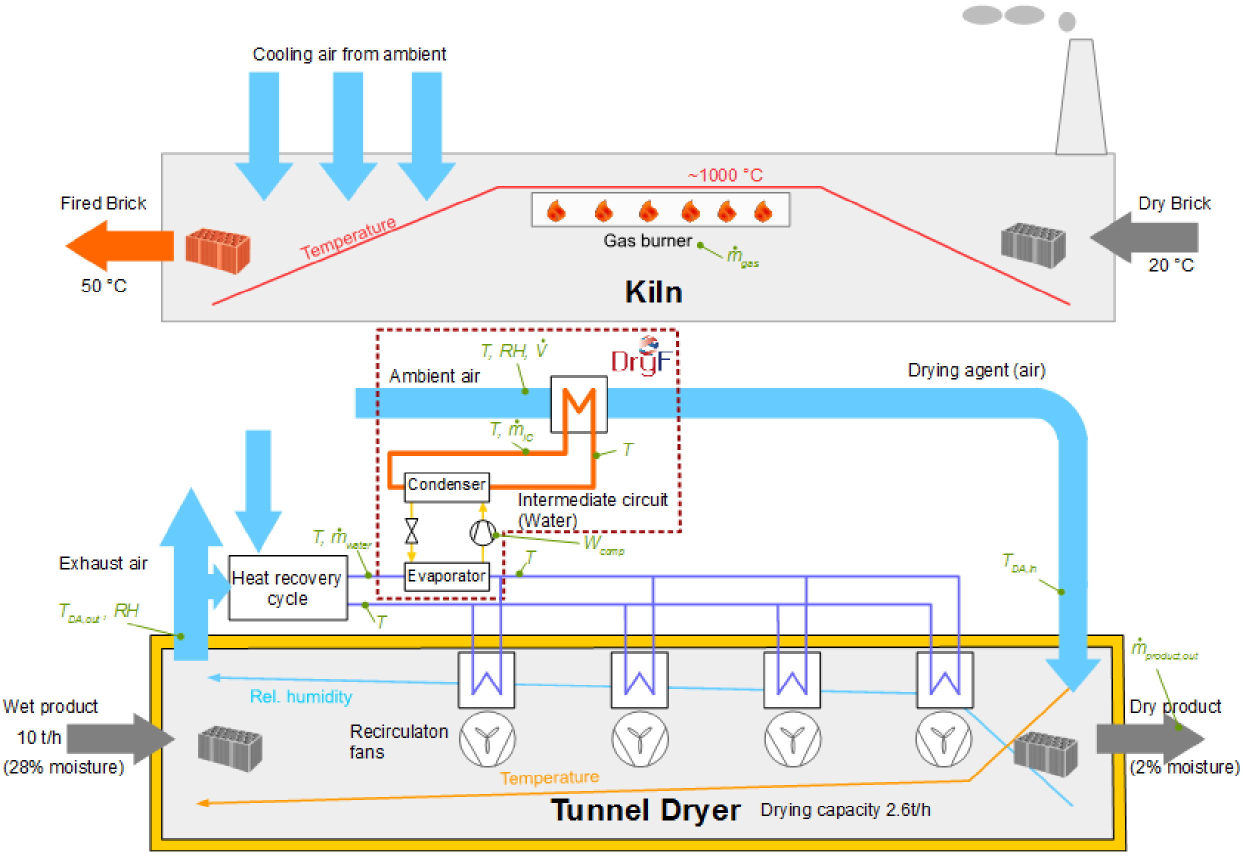

Figure 14: Schematic diagram of the required measurements (in green) in order to determine the KPIs of the closed loop heat pump system for the brick drying at Wienerberger.

4.3 MVR drying of sludge

The open loop MVR heat pump for the sludge drier will be installed on a batch dryer of Scanship Holding ASA in Drammen/Norway. The installed MVR system will be large enough to connect at least 2 dryers in parallel to the open loop heat pump.

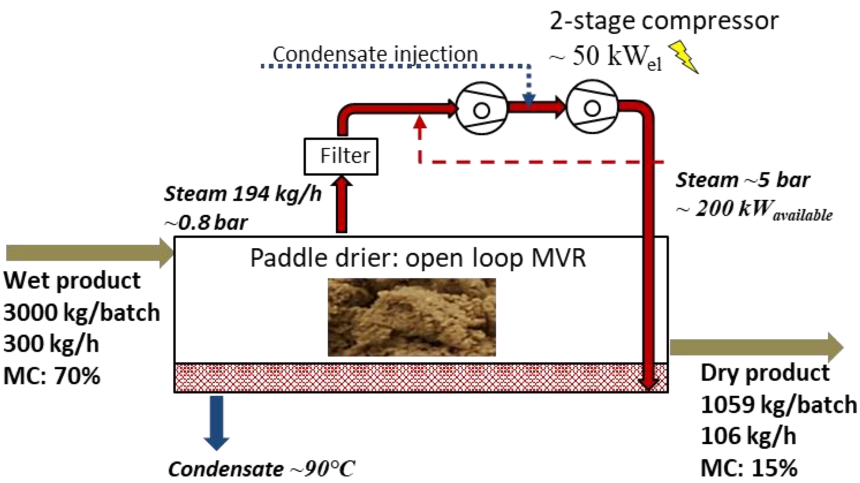

Figure 15: Schematic diagram of MVR-dryer for sludge drying. The excess steam is compressed to high pressure and temperature by a two-stage compressor. The latent heat is used to supply the necessary process steam to the system.

The dryer of the demonstration unit will have a thermal capacity of approximately 200 kW and will be operated at a pressure of 0.8 bar to 1 bar. Operation at 1 bar (=atmospheric pressure) will be more beneficial with respect to achievable temperature lift of the open loop heat pump. For the full scale it will be possible to connect 2 driers to one open loop heat pump. The dryer of the demonstration unit will have a reduced sized compare to the full-scale system and will also allow for flexible product testing under varying drying conditions, products and capacities. An additional steam generator will be installed in order to compensate for capacity variations as well as for the start-up procedure. In order to determine the KPIs the following measurements need to be implemented:

- Dryer:
  - Product mass flow ($\dot{m}_{\mathrm{product,in}}$)
  - Moisture content of product in and out of the drying chamber ($MC_{\mathrm{product,in}}$; $MC_{\mathrm{product,out}}$)
  - Oxygen content (O_2) of the excess moisture
- MVR system:
  - Work (=electric power needed) of Compressor 1 and 2 (W_{Comp1}; W_{Comp2})
  - Temperature and pressure (T, p) before and after each compressor
  - Mass flow of water used in the intercooler ($\dot{m}_{\mathrm{water,\ intercool.}}$)
  - Mass flow and temperature of condensate out of the condenser (T_{cond}; $\dot{m}_{\mathrm{cond}}$)
- Additional steam generator
  - Mass flow of additional steam provided ($\dot{m}_{\mathrm{steam,\ additional}}$), if applied

The measurement points are illustrated (green color) in Figure 16.

The following parameters are either calculated from measured values or are provided from different sources:

- Production site:
  - Energy price for fossil fuel and electricity [€/kWh] will be provided by the production site

- CO_2 emissions factor [CO_2/kWh] and primary energy factor [kWh/kWh] for fossil fuel ieieand electricity will be determined for the country, in which the production site is located (retrieved from current studies and reports).

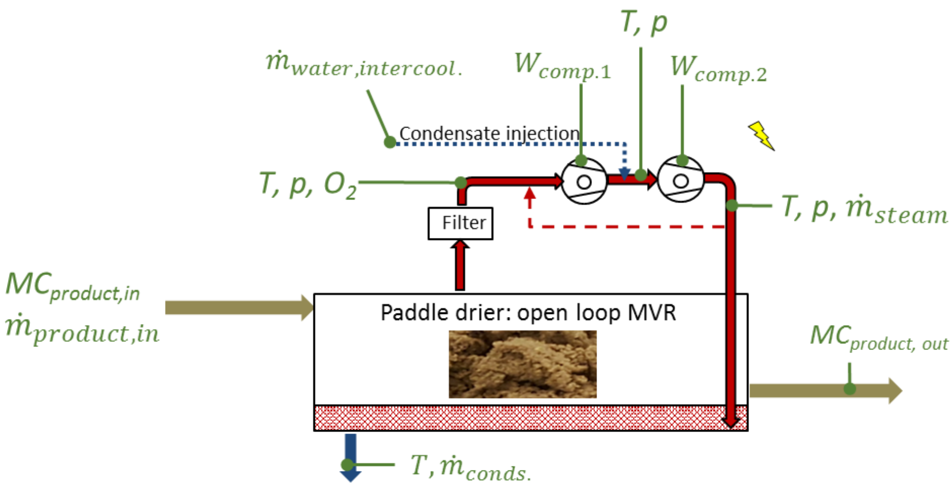

Figure 16: Schematic diagram of the required measurements (in green) in order to determine the KPIs of the open loop heat pumps system (MVR) for the sludge drier.

The measurements of the dryer will document the drying conditions and can also be used to determine the SEC of today's systems. The measurements of the MVR system are needed to document the COP of the heat pump and the COP of the heat pump dryer as well as the SEC of the complete system. The SHR will be determined based on the measurements of the condenser and the MVR system (Impact 1). Based on the determined SECs the avoided CO_2 emissions (Impact 2) as well as the reduced production costs (Impact 3) can calculated with the emissions factors and energy prices of the production site.

5 CONCLUSIONS

Three different heat pump drying systems will be built and evaluated in the DryFiciency project. The key performance indicators need to enable a comparable evaluation and to validate clearly the impacts of the system solutions. The impacts are:

- **Impact 1**: **Recovery** of at least **40% of the sensible heat** contained in each waste heat carrier addressed by the project.
- **Impact 2**: Measureable **substantial primary energy savings** clearly quantified and substantiated, and subsequent **reduction of CO_2 emissions**
- **Impact 3**: The improvement of the energy efficiency and the reduction of energy cost will lead to **a demonstrated advancement in competitiveness** by the end of the project. This will expand the available portfolio of energy resources and technologies, which can be integrated within sites, across sectors and along value chains

The following key performance indicators will be determined and used to evaluate the systems:

1. Coefficient of Performance (COP) of the heat pump dryers (Impact 2) Eq. (5)
2. Specific Energy Consumption (SEC) of the system (Impact 2) Eq. (11)
3. Sensible Heat Recovery (SHR) of the system (Impact 1) Eq. (13)
4. Reheating Ratio (RHR) of the heat pump dryers (Impact 1) Eq. (14)
5. Reduction of CO_2 emissions (Impact 2) Eq. (16)
6. Primary energy savings (PES) (Impact 2) Eq. (17)
7. Improvement of energy efficiency (EE) (Impact 3) Eq. (18), (19)
8. Relative cost reduction (Impact 3) Eq. (21), (22)

The COP and the SEC will be used to demonstrate the principal potential of the technological solutions. SEC will be also required to calculate the avoided CO2 emissions and primary energy savings (Impact 2) and the competitive advantage in the form of reduced production costs and increase in energy efficiency (Impact 3). The SHR and the RHR will demonstrate Impact 1.