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Heat recovery from large scale brewery cooling system

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Abstract

This paper discusses 2 examples of using waste heat from a brewery cooling process with heat pumps. The first example is the transfer of condensation heat to the heat usable for bottling, mashing or in the return flow of a district heating system to increase the water temperature. The second is the use of superheating heat to increase the return water temperature of a district heating network or mashing, lautering or bottling. Both possible solutions for the use of heat pumps offer real possibilities of introducing part or all of the waste heat of the cooling system to the level of useful temperatures. The 1st concept (usage of heat of condensation and discharge gas heat) is much more interesting because it gives real savings for the plant and possibilities of selling heat to an external recipient. The temperature level is also sufficient to cover all own technological purposes at temperatures up to 70°C.

Keywords: Energy efficiency; Industrial heat pumps; District heating; Industrial refrigeration

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1. Introduction

Nowadays, heat and electricity are key components of production costs in breweries [1], hence the desire to manage them in the best possible way. Additionally, waste heat utilization for breweries is, after waste water utilization/reduction or bio-waste usage [2,3], one of the most potent ways of decreasing the plants environmental impact [4]. The waste heat in a brewery is being rejected at a temperature range of 30−130°C [5]. While the usage of high temperature heat is not a problem, the heat with temperatures below 50°C is often wasted. One of the possibilities of using low temperature waste heat is to integrate it with its own heating/preheating system or local district heating. If the temperature level of the waste heat is low, it can be transformed using heat pumps [6].

In many plants, the results of the energy audit indicate specific improvements that should be applied to obtain the appropriate environmental effect [5,7]. However, auditors rarely delve into improving the efficiency of cooling processes.

Breweries have a constant need for moderate to low temperature cooling [8]. The plant analysed in this article has a very effective high-capacity compressor refrigeration system, with R717 or anhydrous ammonia (NH_3) as the refrigerant. The installation is constantly modernized and its operation is optimized in order to obtain the maximum efficiency. From the perspective of the optimization of the cooling system itself, there are few to no possibilities to improve the efficiency, but there are many possibilities of waste heat management. This opens up completely different possibilities for the plant, which, from the point of view of the brewery, could expand its activities with the commercial sale of heat for consumers or for its own needs. In the first case, the effect would be to increase the plant's income, and in the other case, it would reduce the plant's operating costs by increasing the energy efficiency. In both cases, the positive ecological effect is the reduction of primary energy, which directly translates into the reduction in carbon dioxide (CO_2) as well as other pollutant emissions.

Nomenclature

COP – coefficient of performance
e – emission
EC – emission coefficient
GWP – global warming potential
h – specific enthalpy, kJ/kg
ODP – ozone depletion potential
P – power, kW
Q – heating power, kW
T – temperature, ºC

Greek symbols

η – isentropic efficiency

Subscripts and Superscripts

a – annual
b – including irreversibility
c – condenser
g – heating
hp – heat pump
ref – refrigeration

Abbreviations and Acronyms

CHP – combined heat and power plant
DH – district heating

In the case of selling the heat to local district heating, there are two scenarios for existing municipal systems: connect the hot water to the supply pipeline at high temperatures or connect it to the return pipeline as a kind of preheating for the combined heat and power plant (CHP) [9,10].

In the case of using the heat for own purposes, it can be used either for hot water and heating of premises, or for production processes (as a direct heat source or a source for preheating) [5]. This article may be a guideline for both plant owners and energy auditors whose possibilities to improve efficiency are worth analysing. For plants with similar cooling systems, the presented solutions may turn out to be interesting

2. Materials and methods

2.1. Plant and cooling system description

The cooling system of the described brewery is an installation that uses R717 (ammonia) as a refrigerant. It is a single-stage installation equipped with a compressor rack system consisting of screw compressors and with three central filter dryers. The heat of condensation is dissipated by spray evaporative condensers. Refrigeration system receivers can be divided into 3 main groups:

- working on direct evaporation (tanks, fermentation tanks and air coolers),
- plate and shell-and-tube exchangers for cooling liquids (below 0°C monopropylene glycol solution necessary in the production process and for the air-conditioning cycle),
- accumulative water coolers.

The installed cooling capacity of the compressor unit is 18 600 kW with an evaporative condenser at a condensing temperature of T_c = 25.5°C and an evaporator with an evaporation temperature of $T_0 = -7$°C. The developed power of the compressors' electric motors is 4400 kW under these conditions, while the spray-evaporative condensers are prepared to dissipate 24 950 kW with the use of 370 kW of fan power. The information received shows that there is a power reserve of 1800 kW on the side of the condensing unit, to cope with in the event of a failure of one of the compressors. In addition, one of the discharge lines has been equipped with a shell-and-tube exchanger for heat recovery from superheated vapour with a nominal power of 1 MW. During the on-site inspection, it was found that it is possible to retrofit all discharge lines with heat recovery exchangers for superheated refrigerant.

Discharge high-temperature heat is mainly used to produce hot water at a temperature not exceeding 60°C. In the case of the described installation, it is not possible to obtain higher temperatures, therefore the heat is not used for the production of domestic hot water, but only for the heating of the office building in winter and transition periods, in which fan-convectors are installed. In summer, heat is dissipated into the outside air.

Obtaining higher water temperatures from the heat recovery system in the analysed system is unjustified due to the increase of condensing pressure, which in turn causes a decrease in the coefficient of performance (COP). In addition, it is difficult to obtain constant operating parameters of the heat recovery installation due to the variability of the heat flux, which is shown in Fig. 1. The discharge heat can not be used in winter, as the refrigeration system is not used frequently. Also, the temperature of the obtained medium makes the use of such a heat source questionable.

The specificity of the breweries is the fact that the demand for cooling power varies depending on the production volume,

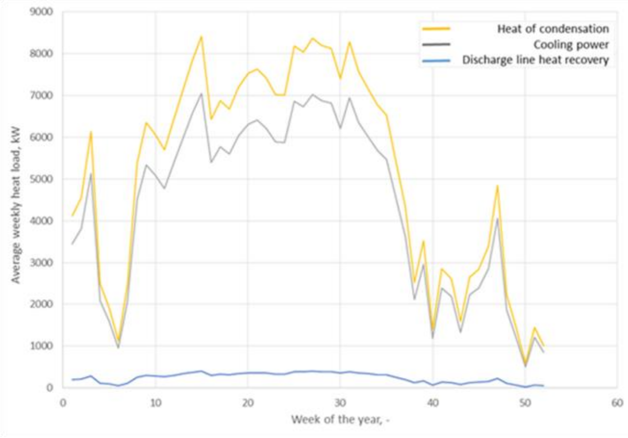

Fig. 1. Weekly heat exchangers load in the compressor refrigeration system. 1st week is in January, while 52nd week is in December.

which in turn depends on the beer sales, which increase with the outside air temperature. According to the data provided by the plant staff and shown in Fig. 1, the demand for cooling power decreases significantly in the winter months and the main reason for that is the decrease in sales. In the case of small and microbreweries, the heat usage depends on the brewing period − most of the high temperature is used during the first days of the process (mashing, malting, lautering), while the cooling effect is used at the end of the process. This creates problems with the efficient utilization of waste heat from the cooling system. On the other hand, for big plants, as in our case, the beverage production is constant and parallel, meaning that all processes occur at almost the same time, but for different batches of product.

Based on Fig. 1, the average cooling capacity of the plant varies significantly from 498 kW in week 50 (December) of the year to over 7050 in week 15 (April) and 7023 kW in week 27 (July). In view of the above, the waste heat load on the condensers varies from 600 kW to 8500 kW. After eliminating the extreme values, it can be assumed that an average of 6000 kW of waste heat can be recovered. The average annual cooling capacity developed by the cooling plant is $Q_0 = 4344$ kW. The average annual load of the condensers $Q_c = 5179$ kW.

From the data provided by the brewery technical staff, the average condensing pressure over the year is 10.2 bar. The evaporation pressure is virtually constant at 3.28 bar and is the same throughout the year for process reasons. In the months when the outside temperature makes it possible to lower the condensing pressure, it is around 9.2 bar. The automatic control system makes it possible to achieve this low condensing pressure and it is the optimum operating point for the system, which is impossible to achieve in the summer months. According to the data received from the brewery, the average coefficient of performance of the installation on the side of the condensing unit is COP = 4.45. However, to maintain such a low temperature of condensation, it is necessary to continuously optimize the process of heat extraction, which is associated with the cost of water evaporated in the spray evaporative condensers and the operation of fans used to disperse heat by forced convection.

The brewery staff did not provide us with the amount of heat utilized by the system, so we gathered data for different breweries from the literature [11−13] and calculated the heat shares consumed for different technological processes. It is presented in Table 1 together with the corresponding temperature levels.

Table 1 Brewery heat sources temperatures and heat shares in plants total heat consumption [11−13].

Heat source	Temperature, °C	Heat share, %
Mashing	45−70	25
Lautering	70−80	13
Bottling	70−80	10
Boiling	100	52
Cooling	2−20	20

2.2. Waste heat utilization concepts

The plant does not have large needs in terms of heat supply for office spaces, but its needs are not sufficiently met in winter, while in summer there is no real possibility of using waste heat. However, in the vicinity of the plant, there is a district heating network supplying heat to single-family housing estates as well as a number of plants, which realize their heating needs throughout the year using local boiler houses. Therefore, an attempt has been made to propose such solutions, which would contribute to waste heat management. One of them assumes the use of both superheated vapour and condensation heat, while the other presents a solution that uses only the superheated vapour heat. Both concepts are feasible and can benefit both the plant and other heat consumers. The biggest benefit is in the reduction of emissions. In the first case, there is virtually no change to the existing refrigeration plant and in the second case, the change to the existing plant consists of adding superheated vapour heat recovery exchangers.

2.2.1. 1st case: Usage of heat of condensation and discharge gas

The concept is based on the implementation of a cooling and heating system realized by means of two-stage water/water heat pumps, using R717 as a refrigerant. A potential heat receiver taken into account is at the return of the district heating network - the increase of water return temperature reduces heat supplied by the heating plant. Supply and return temperatures vary from 70°C to 125°C and from 44°C to 60°C, for outdoor temperatures 14.4 and −18°C, respectively. We have chosen the return of the district heating as the heat receiver because it has a lower temperature than the supply, which will increase the heat pumps COP.

Another potential receiver, according to Table 1, is the bottling stage or first two parts of mashing (requiring 45−62°C). Bottling requires a temperature of 70°C, but it is performed directly by the heated water, so no additional heat exchange occurs in this step.

The criteria that were set for the design task are as follows:

- obtain heat at a temperature level of +70°C, which is suitable as a heat source for the district heating return during the whole year or for mashing or bottling (technological heat);
- the condenser has to remain evaporative-type, as it ensures sufficient heat dissipation even if there is no heat reception on the district heating or technological side;
- use the entire condenser heat load (heat of condensation and superheated discharge vapour);
- do not make significant structural changes to the plant’s refrigeration system that could cause a decrease in its COP;
- minimize heat transfer losses of the produced heat at useful temperatures;
- produce heat with the lowest operating and investment costs;
- possibly increase the coefficient of performance of the plant’s cooling system;
- reduce water and/or electricity consumption for the condenser drive;

- use techniques that avoid legal restrictions on the use of synthetic refrigerants;
- do not increase flow resistance in the district heating system.

The heat pumps lower heat source is the water used for cooling of spray-evaporative condensers, as there is no need to interfere with the cooling installation of the plant on the refrigerant side. The danger of a fatal influence of the heat recovery installation on the operation of the primary cycle is avoided. A water-to-water heat pump installation makes it possible to utilize all waste heat. The water used for condenser cooling would be cooled by about 3 to 4 K, which changes the dew temperature and has a beneficial effect on the cooling cycle of the refrigeration plant – COP is increased due to a lower condensation temperature. By installing heat pumps, it is possible to reduce the amount of water evaporated for dissipating the heat load of the plant's refrigeration system or to reduce the energy consumption of the condenser fans. The schematic and simplified drawing is shown in Fig. 2, which includes the utilization of the cooling effect in air conditioning (AC) systems etc.

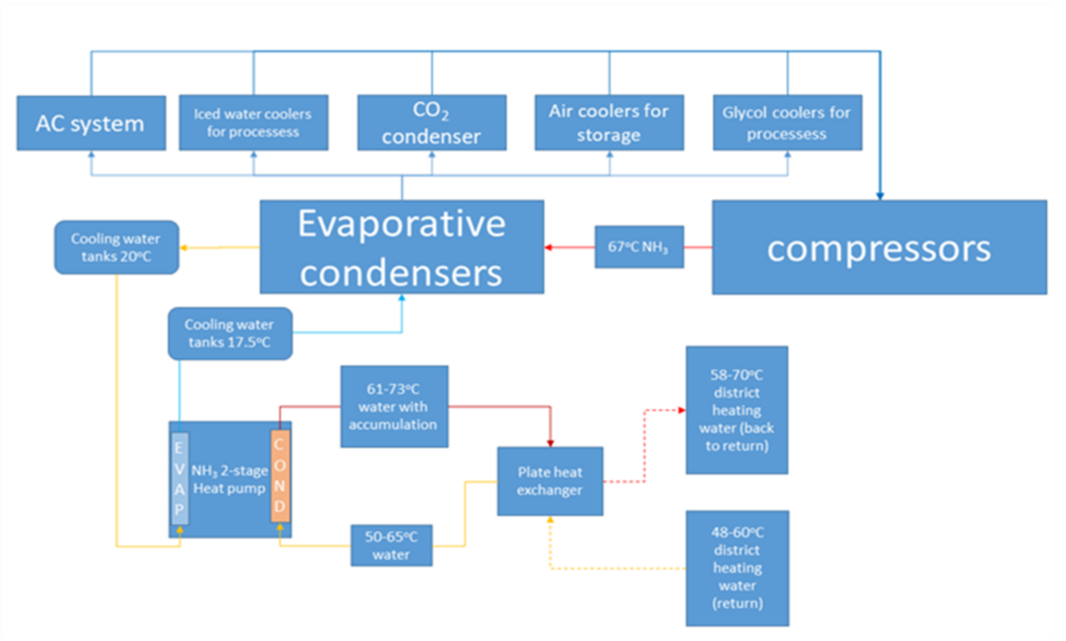

Fig. 2. Simplified schematic diagram of the system proposed in the first case.

2.2.2. 2nd case: Usage of discharge gas heat

The criteria that were set for the design task are as below:

- obtain heat at a temperature level up to +87°C, which is suitable as a heat source for the district heating network;
- utilize the heat of superheated vapours without changing the operating conditions of the refrigeration plant;
- do not make significant changes in the design of the plant's refrigeration system that could cause a decrease in its COP;
- minimize heat transfer losses of the produced heat at useful temperatures;
- produce heat with as little operating and investment cost as possible;
- use techniques that avoid legal restrictions on the use of synthetic refrigerants;
- do not increase flow resistance in the district heating system.

The water leaving the superheated vapour heat recovery system is too cold (up to 53.7°C) to be used for direct injection into the district heating network or for processes other than the first stage of mashing. Thus, it was decided to use single-stage high-temperature heat pumps that would be able increase the temperature of water leaving the heat recovery system.

The concept assumes the use of high-temperature single-stage water/water heat pumps for which the bottom heat source would be water used to remove the heat from the superheated vapours of the brewery's refrigeration system, and the refrigerant making the heat pump cycle would be R717. Such an approach, compared to the idea presented in the previous subsection, requires interference with the brewery's refrigeration plant and the installation of an exchanger to recover the heat of superheated vapours from the discharge line. It was decided to apply a high-temperature single-stage heat pump, which is able to heat up water returning from district heating network receivers from 60°C to 87°C.

Heat exchangers would be installed in the mixing branch. In such a solution, it is possible to increase the temperature of water returning from the district heating and to introduce heat to the consumers. For this purpose, it is necessary to install pumps to cover the hydraulic losses in the transmission on the supply pipe between the location of the mixing branch to the consumers and in the return pipe to the mixing branch with the heat exchangers of the additional heat source. It is also necessary to install a three-way mixing valve to change the flow through the heat recovery system and to control the temperature in the heating network supply, which varies with the outside air temperature.

The system is designed to operate all year. What is variable is the system efficiency and the amount of heat supplied to the district heating network depending on the return temperature from the heating system consumers, load on the cooling system and load of the heating installation. The schematic and simplified drawing is shown in Fig. 3.

The application of such a solution, as opposed to the previous one, does not change the COP of the brewery's refrigeration system. It will also not significantly affect the savings on cooling water for spray-evaporative condensers. It only serves the purpose of utilizing the heat of superheated vapours in a useful way. In order to utilize it and feed it into the district heating network, it is necessary to use a heat pump capable of heating the return water from the district heating network by a minimum of 25 K.

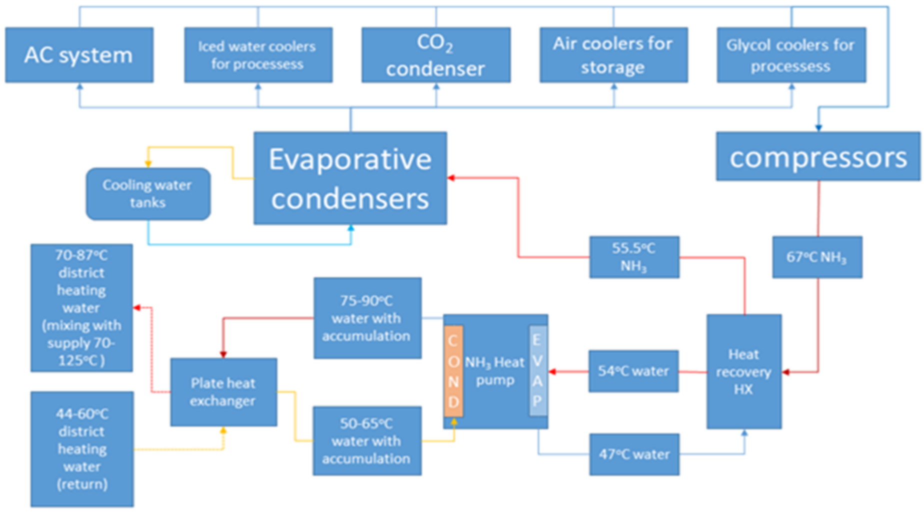

Fig. 3. Simplified schematic diagram of the system proposed in the second case.

2.3. Calculations

The cycle consists of four components: compressor, condenser, throttling valve and evaporator. Pressure drops are not considered in the ideal cycle. All refrigerants thermo-physical data were calculated using the CoolProp library [14].

For both the brewery refrigeration system and heat pumps, we first determined the pressure of evaporation and condensation for given saturation temperatures. The suction enthalpy h_1 was calculated at the evaporation pressure with a 3−10 K superheat. The ideal enthalpy at discharge h_2 was calculated using the condensation pressure and under the assumption of isentropic compression $s_2 = s_1$. The real enthalpy h_{2b} was calculated assuming isentropic efficiency η:

$$h_{2b} = h_1 + \frac{h_2 - h_1}{\eta}. \tag{1}$$

Throttling was isenthalpic from the subcooled point h_3 (2−5 K) at the condensation pressure. The calculations do not take into account the power consumed by the pumps, as this is highly dependent on the length and height of the pipeline.

The coefficient of performance for the refrigeration system (COP_{ref}) is the ratio of its specific cooling power $h_1 - h_3$ to specific compression power $h_{2b} - h_1$ or real cooling power Q_0 to real compressor power consumption P in the case of the real COP. The coefficient of performance for the heat pump system (COP_{hp}) is the ratio of its specific condensation power $h_{2b} - h_3$ to specific compression power $h_{2b} - h_1$.

As stated in the system description, the brewery's refrigeration system works on R717, condensing temperature of $T_c = +25.5°C$ and evaporation temperature of $T_0 = -7°C$, superheat 10 K and subcooling 2 K. Knowing that the measured annual cooling COP was 4.45 we assumed the compressor isentropic efficiency (η) to be 0.63.

We performed simulations of the heat pump cycle working on two different pure low global warming potential (GWP) and zero ozone depletion potential (ODP) refrigerants: R290 (propane) and R717 (ammonia). We chose these refrigerants because there are many commercial heat pumps on the market for the two chosen refrigerants.

For the 1st case heat pump cycle, the following values were assumed: evaporation temperature 10−20°C, condensation temperature 75°C. The assumed cooling power of the evaporator is Q_0 = 5.179 MW (the power from the refrigeration systems condensers). The compressor isentropic efficiency is η = 0.78 for the second stage and 0.83 for the first stage.

For the 2nd case heat pump cycle, the following values were assumed: evaporation temperature 43°C, condensation temperature 90°C. The assumed cooling power of the evaporator is Q_0 = 1.2 MW (the heat of superheated vapour from the refrigeration system compressors) and the compressor isentropic efficiency $\eta = 0.78$.

The carbon dioxide emission caused by the electricity consumed by the system is equal to the used electricity P and carbon dioxide emission coefficient (EC_{CO2}):

$$e_{CO_2} = P_a \cdot EC_{CO2}. \tag{2}$$

The carbon dioxide emission factor can be taken from the annual data for a given country. In the exemplary brewery, the end user electricity is taken from the Polish electrical grid, so its EC_{CO2} = 685 kg/MWh [15]. In order to compare the carbon dioxide emissions of the electrical heat pump with the professional heat and power plant we used the value of EC_{CO2} = 94.83 kg/GJ [16], which corresponds to the direct burning of coal.

The exact compression power of the exemplary brewery's refrigeration plant is not known, but its mean yearly COP is equal to 4.45. The average annual compressor power P_a of the refrigeration system can be calculated as in Eq. (3):

$$P_a = \frac{Q_{0,ref}}{\mathrm{COP}_{ref}}. \tag{3}$$

The heat losses from the preinsulated pipes connecting the brewery with local district heating (DH) depend on the type of insulation, pipe diameter and most importantly the distance and temperature. The average heat losses for our pipeline are between 18–80 W/m [17]. For the calculations we used the medium value of 49 W/m, knowing that the distance to the DH network was 1 km. It means that the transfer losses constitute less than 1% of the heating power in the first case and less than 4% in the second case.

3. Results and discussion

For R717 in the first case, the heat pump obtained Q_c = 6.93 MW of thermal power in the condenser with the use of P = 1.75 MW of electric power to supply the compressor. It translates to the heating COP of 3.95.

For R290 in the first case, the heat pump obtained Q_c = 7.14 MW of thermal power in the condenser with the use of P = 1.97 MW of electric power to supply the compressor, which translates to the heating COP of 3.64.

The heat pumps COP is higher for the R717 heat pump than for the R290, so this refrigerant was chosen. The advantage of this solution is that the current exemplary plant is also equipped with the refrigeration system working on R717, so both devices can be maintained by the same staff.

Heat pumps produce hourly 24.9 GJ of heat, which translates to 218 124 GJ/year supplied to the district heating at the average meteorological year in Central and Eastern Europe (Poland). If this amount of heat would come directly from burning coal then the emissions would equal 20 684.7 tons of CO_2. Instead, it comes from the electrical grid, so it consumes 15 330 MWh a year which translates to 10 501 tons of CO_2.

By investing in heat pumps to absorb the heat load of the condensers, the plant can additionally benefit from the lowering of condensation temperature of the existing refrigeration installation by about 1.5−2 K, which would increase its annual average efficiency by a further 8%. Currently, the average annual efficiency of the cooling system is COP = 4.86. Therefore, the average annual electricity demand for the compressor (P_a) would decrease from 969 kW to 829 kW. The decrease of CO_2 emissions e_{CO2} caused by the decrease of the plants refrigeration system power consumption is 840 tons a year.

Further environmental and economic savings can be obtained by reducing the need for evaporating water in the cooling process of spray-evaporative condensers. In relation to the data on water consumption by the plant, with the consumption of 71 388 m^3 of water, the savings may be about 2500 m^3 a year.

For the second case: R717, the heat production is Q_g = 1.45 MW in the condenser at a temperature 90°C with the use of P = 0.65 MW of electric power to supply the compressor. It translates to the heating COP of 2.23.

Compared to the previous solution, the plant cannot count on electricity savings due to the lowering of the condensing temperature. Savings due to the reduction of water evaporation in spray evaporative condensers will be about 10 times lower than in the 1st case.

The cooling power constitutes on average 20% of the total heat usage in the brewery (Table 1). From the data of the brewery, we know that the average annual cooling system condenser power is 20% higher than the evaporator power (24% of the total). If this heat is used as a lower heat source for the heat pumps described in the 1st case, then the heat gained from the heat pumps could cover the demand for 31% of the total heat used in the brewery. According to Table 1, the temperatures obtained from this case are usable only for 2/3 of mashing and all bottling, which takes 26.5% of the total heat consumed by the brewery. This means that the heat pumps could work with safety allowance. This is the most rational usage of waste heat as the waste heat source. It is the most coherent with the needs for heat, compared to selling it to district heating.

If we proceed with the same calculations for the second case, then the heat pumps heat source (heat of superheat vapour) consists only of 5.5% of the total heat used in the brewery. This means that only 6.6% of the brewery's total heat consumption could be served by the heat pumps.

4. Conclusions

This paper discusses 2 examples of using waste heat from a brewery cooling process with heat pumps. The first example was the transfer of condensation heat to the heat usable for bottling, mashing or in the return flow of a district heating system to increase the water temperature. The second was the use of superheating heat to increase the return water temperature of a district heating network or mashing, lautering or bottling.

Both possible solutions for the use of heat pumps offer real possibilities of introducing part or all of the waste heat of the cooling system to the level of useful temperatures. The 1st concept (usage of heat of condensation and discharge gas heat) is much more interesting because it gives real savings for the plant and possibilities of selling heat to an external recipient. The temperature level is also sufficient for own technological purposes. The 2nd concept provides very small amounts of heat but on temperature levels that can be directly used for district heating. Both cases will be less effective if the distance between the brewery and the heat recipient is large. In our case, the distance of 1 km generated heat losses of less than 1–4%.

Looking more broadly when selling the heat to the municipal district heating, it should be analysed whether the heat generated by the heat pumps will be supplied to the district heating network, the source of which is a combined heat and power plant (CHP, cogeneration) or a heating plant. In the case of a heating plant, when heat is introduced to DH from another source the benefits are manifested directly in the reduction of fuel consumption and in an almost double decrease of annual carbon dioxide generation. In both cases, if the heat pumps and cooling devices were coupled with renewable energy sources, the decrease of environmental impact would be even higher. Unfortunately in the example considered here, that was impossible.

When the heat source for the heating network is a CHP plant, connecting additional heat sources to the network is even harmful. This is due to the fact that heat in district heating powered by cogeneration plants is a waste energy generated in the conversion of chemical energy contained in fuel into electricity. All CHP plants have a problem with heat management because it is not used for central heating in summer. The coherence of the solutions presented in cases 1st and 2nd does not coincide with the increase in demand for heat, which will result in the fact that any amount of heat introduced into the district heating network means that it must be dispersed in the environment by cooling towers. This situation could happen if a city has one company responsible for municipal district heating network and another for heat generation (for example city of Wroclaw, Poland).

When analysing the feasibility of a heat recovery system, it is important to consider the issues of cooperation of the designed system with other systems and the global impact on the environment. Often, the environmental outcome is dependent on the cooperation of the heat sources and one may come to the wrong conclusion by considering only the local effect at the plant level. The final conclusion should therefore be as follows: the processed waste heat should only be fed into district heating networks whose sources are heat plants or local boiler houses.

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