ENERGY EFFICIENT SOLUTION IN THE BREWING PROCESS USING A DUAL-SOURCE HEAT PUMP

by

Peter HERMANUCZ [a*], ***Gabor GECZI*** [b], ***and Istvan BAROTFI*** [a]

[a] Institute of Technology, Hungarian University of Agriculture and Life Sciences,
Godollo, Hungary
[b] Institute of Environmental Science,
Hungarian University of Agriculture and Life Sciences, Godollo, Hungary

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In this paper, the investigation of an experimental heat pump equipment is presented. This equipment is capable of serving two different temperature cooling needs and a given temperature heating demand within a given industrial process, adapting to its time and performance requirements. The use of a heat pump is energetically advantageous, if a nearly continuous operation can be realized in the course of an industrial process to serve simultaneous cooling and heating needs. In the field of brewing, it was demonstrated by model experiments that CO_2 emissions can be reduced by 60% with a 10% reduction in energy costs if the energy requirements are satisfied using a dual-source heat pump.

Key words: *dual source heat pump, brewery, waste heat, COP*

Introduction

In the case of industrial processes, the focus these days has been on energy efficiency, environmental protection and the reduction of CO_2 emissions. This is especially true for energy-intensive processes, such as brewing [1]. In this field as well, environmentally conscious energy solutions, which can serve as models for other sectors, are being used increasingly. In the following, the results achieved so far in the field of brewing are presented through some examples, and a system that improves energy efficiency further is also presented.

The *Green Brewery Goss* in Leoben, Austria, is a lighthouse project of the international company Heineken. The site is completely CO_2-neutral. The 90% of the waste heat produced during its operation is reused for preheating water. Solar collectors (1500 m^2) and the production of biogas with a draff fermenter and a wastewater treatment plant contribute to CO_2-neutrality [2].

The Dreher Brewery, Budapest, Hungary, now employs state-of-the-art technical solutions for energy efficiency, for example, the installation of absorption chillers. Annual energy savings of about 500 MWh, equivalent to the consumption of some 380 households, were achieved in 2009. The development was implemented by Dreher in such a way that the waste heat generated during the brewing process was recycled back into the system as a *raw material*. This way, less energy is needed during cooling [3].

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[*] Corresponding author, e-mail: hermanucz.peter@uni-mate.hu

Spitz GmbH, Upper Austria, is a food and beverages company, which produces 54 GWh of heat and 4500 MWh of electricity annually in its in-house biomass combined heat and power generation plant. The company thus meets its own energy needs and the heat requirement of the local heating network in the Attnang-Puchheim region [4].

In the brewing industry, biogas production is popular as described previously. Yingjian *et al.* [5] suggests the following solutions to utilize the produced biogas. One of the best solution is to employ a compression chiller driven by an internal combustion engine (CCDE). In this case, the engine could run on biogas instead of an electric motor to drive the compressor for refrigeration. The CCDE avoids the loss of power transmission and distribution and improves mechanical efficiency. Another benefits of a CCDE is that it partially improves the variable speed performance of the load efficiency and provides hot water by recycling the waste heat generated by the engine.

Another possibility is the direct-fired absorption chiller. This lithium bromide absorption chiller uses more energy-efficient renewable sources, such as solar thermal collectors, geothermal resources and biogas as the heat source of the chiller. The gas-fired absorbtion chiller may apply a primary or secondary steam heated evaporator to enhance the COP of cooling and the primary energy ratio. This expands the scope of use of the chiller.

The use of solar energy in the food industry has also been studied by other researchers. It has been concluded that solar energy can be optimally integrated into the energy supply of certain food industry processes [6, 7].

Borges *et al.* [8] presents another possibility for the energetic utilisation of biological waste generated during the brewing process. They examine the possibilities of energetical recovery of brewers' spent grains by means of chemical processes – such as combustion or pyrolysis – seeking for the economical and environmental advantage. However the utilization of this product is theoretically solved, it is most frequently sold as animal fodder. However, usage in the energy sector also holds promising possibilities. One of the greatest advantages of energetical usage realized by means of pyrolysis lies in the fact that substances with lower quality might also be utilized since the environment is not excessively polluted by using this method.

Due to the abundance of low temperature processes in the food industry, low grade heat sources are abound. These include equipment, such as boilers, power plants, refrigerators and air compressors which release heat into the environment, *e.g.*, in the form of hot gases [9]. There are also manufacturing processes, for instance, typical drying and pasteurisation processes that operate below 100 °C, superheated steam drying, sterilisation (>100 °C), cooking (~100 °C), baking (200-250 °C), *etc.* Hence, the exhaust streams of these processes (sometimes after heat exchange for feed preheating) and the associated final cooling streams are typical low grade heat sources. Figure 1 shows some of the application possibilities of an industrial heat pump.

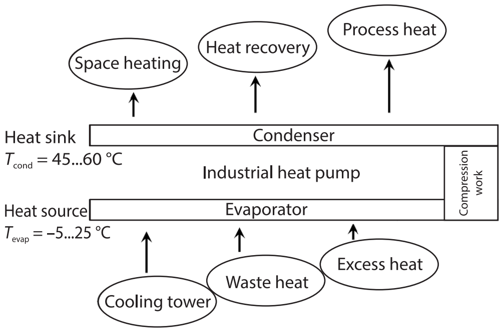

Figure 1. Application possibilities of industrial heat pumps

Sensible and latent heat energies of these low grade heat sources are recovered by electrically driven heat pumps. Commercially available heat pumps operate below 100 °C, most of them at temperatures not exceeding 80 °C [10]. The best COP of 3-5 are achieved using typical refrigerants such as R134a [11].

A good example for heat pump utilisation in the brewing industry is the Schwechat brewery in Lower Austria. Here, the biogenic fermentation heat of the brewing process is recovered via a heat pump for local heat supply. Construction started in 2018 and, after completion in 2021, around 900 apartments will be heated by this source [12]. In this system, the heat pump has only one heat source, and the heat sink is outside the boundaries of the brewery.

A Project Report from 1983 [13] stated that the application of heat pumps in the brewing industry is restricted to heat recovery from liquid streams in the range of 10-30 °C and to the use in liquid process streams in the range of 55-80 °C**.** Even a 300000 hl per year brewery has a heat demand for boiler make-up of 108 kW at 55 °C condensing temperature from a heat pump. The same sized brewery also has a waste heat source from bottle and can pasteurisation at 15 °C, and form bottle washing at 20 °C evaporation temperature, respectively. These are good conditions for a heat pump with two parallel heat sources and a heat sink at 50-60 °C. In our present work, the model of a similar system is described and its effectiveness is examined.

Material and methods

The medium sized brewery in this case study is located in southeastern Scotland and has an annual beer production of around 250000 hl, nearly the same size as that in a previous investigation [13]. The brewery in question has a batch cycle duration of six hours. The process starts with mixing malted barley with hot water at 80 °C, stored in a hot liquor tank. The hot water is produced by a steam boiler fed with natural gas. After mixing the ground grains with water, the mixture is left idle for one hour in the mash tun at an average temperature of 65 °C. The residual wort is pumped into the copper vat where hops are added before the mixture is boiled. To supply the required heat, another steam boiler is used. Before the fermentation process starts, the wort is cooled down in a heat exchanger from 97-20 °C (fermentation temperature) by heating up feed water from 8 °C to approximately 68 °C. The brewing process studied by Eiholzer *et al*. [7], supplemented by our proposed DSHP system, is shown in fig. 2.

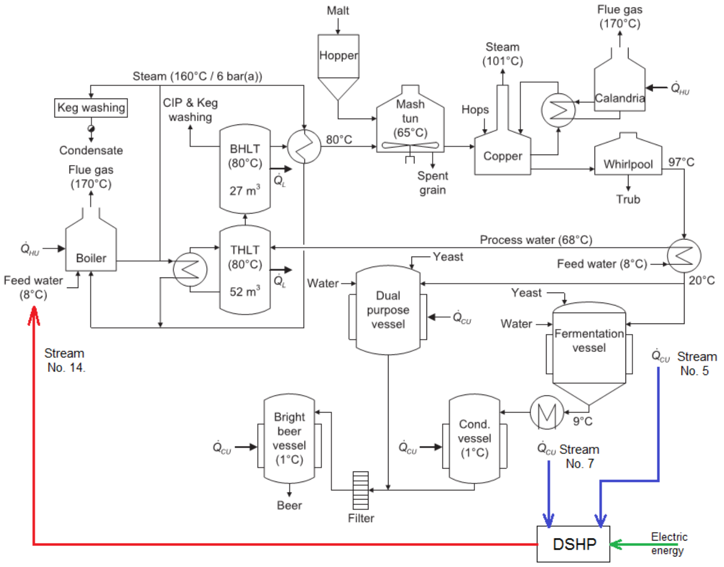

Figure 2. The investigated brewing process [7] edited by the author

One of the existing heat recovery measures is the pre-heating of feedwater during the transfer phase of wort from the whirlpool to the fermentation tanks. According to the composite curves calculated with the time average model, not all the waste heat can be utilised in this way. Eiholzer *et al*. [7] consider a fairly large minimal temperature difference of 12 °C during the heat recovery process. The wort cooling starts at 97 °C, so it is possible to transfer some heat from the fermentation cooling process to the feed water preheating, with a condensation temperature around 55 °C [13]. For this purpose, another heat exchanger, the condenser of the double-source heat pump (DSHP) must be inserted into the process. In this paper, the exact construction of this implementation is not investigated, only the model of a heat pump system was established and measurements were conducted.

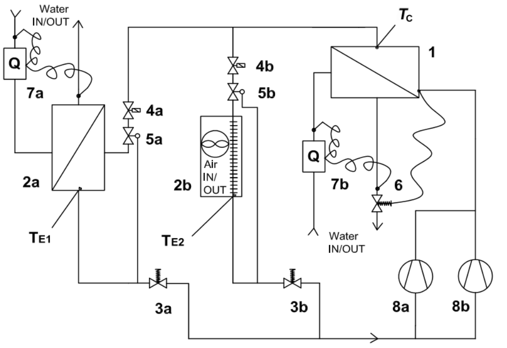

Figure 3. Lay-out of the experimental heat pump model

Establishment of the heat pump system model

In this model, a heat pump with two heat sources which could be operated in parallel, or standalone mode is presented. In industry, operation of two heat sources in parallel mode is not a common solution, however, some investigations have presented nearly the same system [14]. The lay-out of the system is shown in fig. 3, the description of the components is presented in tab. 1.

The experimental system has two compressors – 8a and 8b, and two evaporators – 2a and 2b, working on a common condenser – 1. The system uses refrigerant R404a. The evaporation temperatures can be set independently by changing the temperature and flow rate of the water passing through the heat exchanger – Case 2a, or by adjusting the temperature of the air passing through the heat exchanger – Case 2b. The latter is necessary because the target evaporation temperature of –4°C could no longer be achieved using water as a heat source. Evaporator – 2b is housed in a chamber with electric heating with a PID controller. This means that, from the point of view of the experiment, it plays the same role as if a water heat source operating at the same evaporation temperature had actually been used. The change in power

Table 1. Description of the system components

System component	Description
Compressor – 8a	Bitzer 2KC-05, 2Y-40S
Compressor – 8b	Bitzer 2GC-2, 2Y-40S
Condenser – 1	Alfa-laval ACH-18-10H-F
Evaporator – 2a	Swep B8THx20/1P-SC-M 4x3/4
Evaporator – 2b	Inter-thermo 1-24B7E
Solenoid valve – 4a, 4b	Castel 1068/M10
Thermostatic expansion valve – 5a, 5b	Danfoss TES2
Pressure regulator – 3a, 3b	IPR-6-5/8 ODF
Heat flow meter – 7a, 7b	Techem compact Ve. heat flow meter
Thermostatic pressure regulator – 6	Castel 3210/04

was achieved by switching the compressors on and off. Compressor – 8a operated in time interval 0-390 seconds, while compressors – 8a and 8b, operated in time interval 390-720 seconds. Keeping the condensation temperature almost constant was realized by the six thermostatic pressure regulators, controlling the flow rate of the water flowing through the condenser.

Measuring system and initial parameters

The heat output of evaporator – 2a and condenser – 1 was measured with heat flow meters – 7a and 7b. The electric power absorbed by the compressors was measured with an electricity meter (not shown in the figure). The power of evaporator – 2b was calculated:

$$\dot{Q}_{\mathrm{e2b}} = \dot{Q}_{\mathrm{cond}} - P_{\mathrm{comp}} - \dot{Q}_{\mathrm{e2a}} \tag{1}$$

The power of the evaporators was adjusted in accordance with the proportion of the power of the original processes – 5 and 7, and the power of the condenser was a given. Evaporation and condensation temperatures were determined based on the refrigerant pressures in the heat exchangers using the COOLPACK software. A list of the measuring instruments used is given in tab. 2.

Table 2. Measuring instruments used

Measured parameeter	Measuring device	Accuracy
Pressure in cindeser – 1	Dixell XC400, Honeywell PX3 transducer	+/–1%
Pressure in evaporator – 2a	Refco analogue pressure gauge	Class 1.6
Pressure in evaporator – 2b	Refco analogue pressure gauge	Class 1
Heat flow of evaporator – 2a	Techem Compact Ve. heat flow meter	Measure from $DT = 0.2$ K
Heat flow of condenser – 7b	Techem Compact Ve. heat flow meter	Measure from $DT = 0.2$ K
Electric power of compressor – 8a, 8b	Ganz GH41f electric power consumption meter	+/–1%

To determine the exact parameters of the model, the data of Eiholzer *et al.* [7] were taken into consideration. These are shown in tab. 3 for the brewery under study:

Table 3. Heat sources and heat sinks of the brewery under study

Stream No.	Operation	T_{in} [°C]	T_{out} [°C]	T_{evap} [°C]	T_{cond} [°C]	$\dot{Q}$ [kW]	t_{start} [hour]	t_{stop} [hour]
5	Fermentation	20	9	+4	–	60	0	6
7	Treatment	9	1	–4	–	81	3.25	6
14	Preheating	8	80	–	50	104	0	6

In the heat pump system model, we use streams – 5 and 7 as parallel heat sources, while stream – 14 is the heat sink for this model. Not all of the heat demand must be covered by the heat pump system, because of the already implemented heat recovery system. A condensation temperature of $T_c = 55$ °C was selected, as of [13]. For evaporation temperatures, $T_{\mathrm{e2a}} = +4$ °C and $T_{\mathrm{e2b}} = -4$ °C were selected, taking into account the temperatures of streams – 5 and 7, with $\Delta T = 5$ °C. The similarity between the experimental and the actual process is ensured by the fact that the power ratio of the heat pump evaporators is the same as the power ratio of streams – 5 and 7. The evaporation and condensation temperatures are the same as well.

In the experiment, the amount of heat and the electric energy consumption for the whole process was not investigated, so there was no need to apply a real-time model. In our case, every two seconds represent one minute in the real process, meaning that the experiment takes 720 seconds instead of six hours.

Results and discussion

Fluctuations in the parameters were observed during the experimental measurements, therefore, the averages of the measured values are shown in tab. 4.

Table 4. Results of the experimental measurements

Time [second]	T_{e2a} [°C]	T_{e2b} [°C]	T_{cond} [°C]	$\dot{Q}_{e2a}$ [kW]	$\dot{Q}_{e2b}$ [kW]	P_{comp} [kW]	P_{cond} [kW]	*COP*
0... 390	4.2	–	52	1.69	–	0.92	244	2.66
390... 720	4.3	4.2	54	1.77	2.48	3.07	7.19	2.35

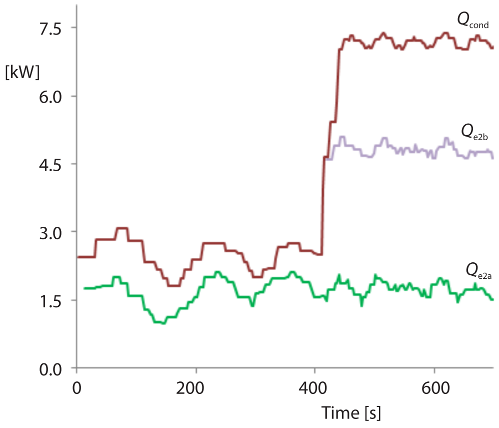

Figure 4. Power of the evaporators and the condenser as a function of time

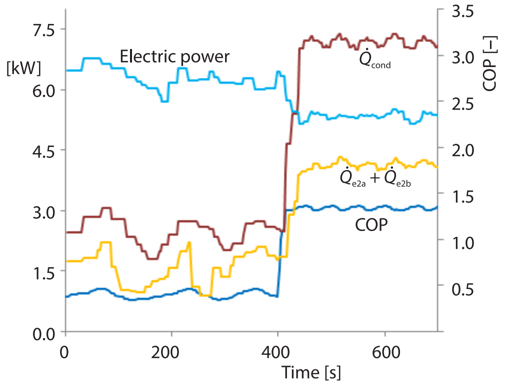

Figure 5. Energy characteristics of the experimental heat pump as a function of time

Figure 4 shows the power absorbed in evaporators – 2a and 2b and the power delivered in the condenser as a function of time. System-specific oscillations, as well as an increase in power at 390 seconds can be observed. The latter was achieved by connecting compressor – 8b. This was necessary in order to be able to adjust the power ratio of the evaporators according to the power ratio of processes – 5 and 7 which served as starting points.

Figure 5 shows the total absorbed and delivered power, as well as the absorbed electrical power and the COP as a function of time. Similarly to the aforementioned, oscillations can be observed here, as well as a lower COP value associated with the operation of the two heat sources and two compressors. The reason for this is that the evaporation temperature of evaporator – 2b operating in time interval 390-720 seconds is lower than that of evaporator – 2a operating in time interval 0-720 seconds. Thus, the average evaporation temperature of the entire cooling cycle was lowered. This increased the temperature difference between the heat source and the heat release side, causing a decrease in the COP [15]. Despite the reduction in the COP, there are clear environmental benefits to using the DSHP system.

Previous research has shown that a heat pump system suitable for the simultaneous utilization of two heat sources can be created. For example, Corberan *et al*. [16] demonstrated the utilization of air and water heat sources using the same equipment. In the course of their work, the two heat sources are not utilized simultaneously, and the aim of the work was to fulfil the energy demand of a building.

Busato *et al.* [17] described a system using three heat sources (ground, solar and recovery heat from an air handling unit). However, their heat pump employed only one heat exchanger as evaporator, therefore, only one evaporation temperature existed in the system at a time. By connecting two heat sources to a heat pump they could create an arrangement similar to the one we designed and investigated in this paper. However, a big difference is that our system utilizes two water (waste heat) streams as heat sources, and at the same time fulfil three energy demands: two types of cooling and one type of heating demand.

A solution similar to the one we examined is used at the Puntigam brewery, Graz, Austria. There, the waste heat from the fermentation process is decoupled with heat exchangers from the cooling system. Parts of the waste heat can be used directly at 75 °C, the low temperature heat is warmed with a heat pump to temperatures between 50 and 70 °C [18]. In this system, the heat sink is part of the energy system of the brewery, but the heat pump employs only one heat source instead of two.

Conclusions

In the course of our research, it was proved that a heat pump system with two evaporators operating in parallel at different temperatures and times can be implemented in conjunction with a brewing process. The cycle is suitable for providing heat energy at a temperature level that can be used during brewing and thus replace heat energy produced from natural gas.

Compared to the technology used as the starting point, the process studied by us has significant advantages both in terms of environmental protection and energy consumption, while being able to serve simultaneous cooling and heating needs at a slightly lower energy cost. The environmental benefit of the system is that energy produced from natural gas is replaced by the condenser of the heat pump. This significantly reduces energy consumption because of the system's average COP of 2.5.

Figure 6 shows that using the DSHP system instead of the original boiler+chiller system (BCS), heat production by natural gas fired boiler, cooling by electric liquid chiller, CO_2 production from the consumption of electrical energy is reduced by 60%.

This significant decrease can be explained by the average COP value of 2.5 of the DSHP system, as well as by the simultaneous serving of cooling and heating needs, using a single piece of equipment. There is no need to produce the amount of heat, using natural gas, that was originally responsible for 83% of the emission. In the course of our calculations, the CO_2 intensity characteristic of Hungary (250 g CO_2/kWh electrical energy) was taken into account [19], as well as a boiler efficiency of 85% and a COP value of 3.5.

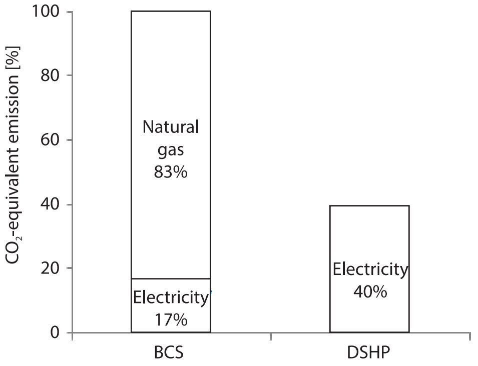

Figure 6 Comparison of CO_2 emissions of the conventional system (BCS) and the DSHP system recommended by us

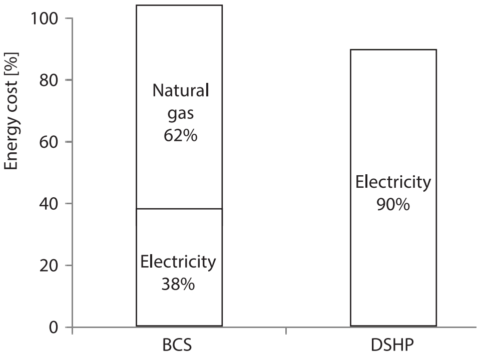

Figure 7. Comparison of the energy costs of a conventional (BCS) system and the DSHP system recommended by us

A comparison of the BCS and DSHP systems in terms of energy cost is shown in fig. 7. It can be seen that the energy cost of the DSHP system is 10% lower. In the course of our calculations, typical energy costs for industrial consumers in Hungary [20] and the average COP values resulting from our measurements were taken into account.

In summary, it can be stated that a heat pump system can be created that can meet some of the simultaneous cooling and heating needs of breweries. At the same time, it can achieve a reduction in CO_2 emissions. The energy cost of such a system is not higher than meeting the same heating needs on a natural gas basis and cooling needs using electrical energy. By optimizing the heat pump cycle and selecting the appropriate refrigerant, the COP values can be further improved [21-23], resulting in a further reduction in energy consumption. Our long-term goal is to investigate the application possibilities of natural refrigerants, as they have additional environmental and energy benefits.

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Nomenclature

COP	– coefficient of performance, [–]
$\dot{Q}$	– heat flow of Stream No. 5, 7, 14, [kW]
$\dot{Q}_{\mathrm{e2a}}$	– heat flow of evaporator (2a), [kW]
$\dot{Q}_{\mathrm{e2b}}$	– heat flow of evaporator (2b), [kW]
$\dot{Q}_{\mathrm{cond}}$	– heat flow of condenser (1), [kW]
P_{comp}	– electric power absorbed by compressor (8a, 8b), [kW]
T_{e2a}	– temperature of evaporation in evaporator (2a), [°C]
T_{e2b}	– temperature of evaporation in evaporator (2b), [°C]
T_{cond}	– temperature of condensation in condenser (1), [°C]

Acronyms

BCS	– boiler+chiller system
CCDE	– compression chiller driven by an internal combustion engine
CU	– cold utility
HU	– hot utility
L	– heat losses

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