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Integrated CO_2 refrigeration and heat pump systems for dairies

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ABSTRACT

Production facilities in the food industry are large consumers of electricity and thermal energy due to energy-intensive processes such as steam production, cleaning, sanitising, refrigeration and drying. Furthermore, there is often a considerable thermal demand for heating the building and for air-conditioning purposes. Dairy plants require both heating and cooling at various temperature levels to process the different dairy products. The thermal demands in these plants have traditionally been covered by separate systems, such as fossil fuel burners or electric boilers for the heating processes and various refrigeration systems for the cooling processes. In recent years, there has been an increased focus on integrated energy systems for dairies as a measure to reduce the overall energy consumption of the plant. This strategy involves integrating all functions to serve the thermal demands into one centralised energy system.

This paper describes the energy system for a dairy plant in central Norway. A CO_2 refrigeration system serves the various cooling loads in the production process. The energy system in the dairy is mapped and the thermal loads have been identified. Based on the current configuration of the cold side of the CO_2 refrigeration units, proposals for improvements are made. Using thermodynamic calculations, the modifications are evaluated in terms of COP improvement and the annual reduction in energy consumption. The calculations show that the energy consumption can be reduced by 12 % to 21.2 % depending on the alteration of the system.

Keywords: CO_2 refrigeration, Energy efficiency, Thermal energy storage, Dairy, Food processing

1. INTRODUCTION

From the projection by the Food and Agriculture Organization of the United Nations, the consumption of milk and dairy products will rise by 19 % by 2050 compared with the 2005-2007 levels (Alexandratos & Bruinsma, 2012). Dairy manufacturing is very energy-intensive (Briam et al., 2015) and the continuous increase of the market demand must cause large energy consumption and greenhouse gas (GHG) emissions (IEA, 2018). To reduce the energy demand in dairy processing as well as the corresponding environmental impacts, it is vitally important to develop more efficient and fossil fuel-free energy systems.

In dairy processing, thermal energy is commonly utilised for sterilisation, evaporation, pasteurisation, drying and cleaning while electricity is usually employed for refrigeration, separation, pumps and control (Ladha-Sabur et al., 2019). At the same time, there is often a considerable thermal demand for heating the building and for air-conditioning purposes. Dairy plants require both heating and cooling at various temperature levels to process the different dairy products. The thermal demands in these plants have traditionally been covered by separate systems, such as fossil fuel burners or electric boilers for the heating processes and various

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refrigeration systems for the cooling processes. The heat required in these processes is all low-temperature heat less than 200 °C (Ramírez et al., 2006) which is a suitable application environment for industrial heat pumps (Arpagaus et al., 2018). Employing the industrial heat pumps in combination with thermal storage tanks to integrate separate systems into one centralised energy system is a promising approach to reduce energy consumption in dairies (Ahrens et al., 2021b; Cox et al., 2022; Kosmadakis, 2019). The energy efficiency can be significantly increased by utilisation of available waste heat streams through the optimization of the energy system (Ahrens et al., 2021a; Clairand et al., 2020; Singh & Dasgupta, 2017).

The present study aims to investigate the potential for reduction of energy consumption in a dairy in Norway that has a high focus on sustainable products and operation of their facilities. The energy system of the dairy has been mapped and the various heat sources and consumers of thermal energy have been investigated. The core of the energy system in the dairy is four CO_2 refrigeration units that connect the cold and warm side of the process through waste heat recovery. The focus of this study is on proposing improvements based on the current configuration of the cold side of the energy system. A project for installing flow and temperature sensors to measure energy flow and thermal demands at various locations in the dairy is ongoing but is not finalised at the current time. For this reason, the measured electricity consumption from the CO_2 refrigeration units over one year is used as a basis to calculate the monthly cooling loads. From the current configuration, improvements of the layout are proposed, and the monthly energy savings are calculated based on the new operational boundary conditions of the CO_2 refrigeration units. The resulting reduction in energy consumption and GHG emissions are then presented.

2. MAIN SECTION

2.1. Existing energy system of the investigated dairy plant in central Norway

The starting point for the investigation in the current study is an existing energy system for a dairy found in central Norway. The energy system satisfies the thermal demands in the plant, spanning from refrigeration to delivery of process steam. A simplified diagram showing the energy flow and current layout of the dairy energy system is presented in Figure 1. The heart of the energy system is four CO_2 refrigeration units, connecting the cold and warm sides of the process. Two units are rated at 40 kW evaporator capacity, while the other two have a maximum evaporator capacity of 80 kW at the current boundary conditions. All four have single compressors with frequency converters for capacity regulation. The CO_2 units are connected to a common glycol circuit, operating at a nominal supply temperature of -12 °C and return temperature of -7 °C. The glycol is accumulated in a 5 m^3 storage tank, where it is supplied to the various consumers in the plant. The calculated storage capacity of the glycol tank is 27.2 kWh at nominal conditions. The cold glycol is utilised for several cooling purposes, where the two main loads originate from cooling of the process water circuit and cooling of the air inside the cold storage rooms. In addition, the glycol is utilised for air conditioning purposes both in the production halls and the offices. The process water is operated in a closed-loop circuit, supplying cold water for the dairy processes. The temperature difference between the supply and return of this circuit is relatively small with 0.6 °C and 2 °C, respectively. Due to the small temperature difference between the supply and return, the cold process water is accumulated in a 9 m^3 storage tank to absorb the short-time peaks in cooling demand. The storage capacity of the cold process water tank is calculated to 14.8 kWh at nominal conditions. The cold process water is supplied to the individual consumers in the dairy by circulation pumps.

On the warm side of the diary energy system, the CO_2 refrigeration units are recovering the heat from the cold side of the process to produce hot process water. City water at approximately 5-10 °C is supplied to the gas coolers of the CO_2 units and heated up to 80 °C by exchanging heat with the refrigerant. The hot process water is used for various purposes in the dairy, including process heating, Cleaning-in-Place (CIP) system for the dairy processes and CIP of equipment in the arrival hall for the milk transport. Furthermore, the heat recovery also covers the demand for domestic hot water (DHW) in the dairy and the feedwater to the electric boiler for steam production. When the CO_2 refrigeration units produce more hot water than required by the consumers in the plant, the excess water is accumulated in a hot water storage system. The total storage

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capacity of the system is 9.6 m3, divided into two parallel packs of 12 x 400-litre tanks that are connected in series. In the existing setup of the dairy energy system, all the cold produced by the CO_2 refrigeration units is at the temperature level of the glycol for simple control and practical considerations. However, several of the thermal demands on the cold side is required at a far higher temperature. This forces the CO_2 units to operate at lower suction temperatures than strictly necessary, negatively affecting their performance. It is therefore of interest to investigate how the total performance of the refrigeration system is affected by altering the layout of the units i.e., connecting some of the CO_2 units directly to the consumers at elevated temperature. Table 1 gives an overview of the thermal demands and their associated temperatures in the dairy energy system.

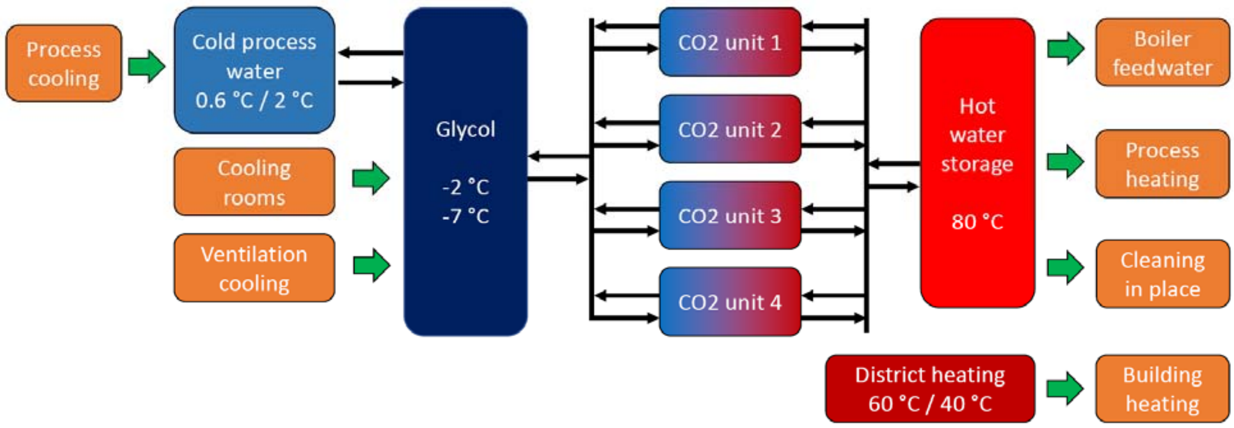

Figure 1: Simplified overview of the current dairy energy system, used as the reference case in this study. The green arrows indicate thermal load to/from the consumers in the dairy, while the black arrows indicate heat transfer between the various circuits in the plant.

Table 1: Overview of thermal energy demands and associated temperatures in the dairy

Consumer	Medium	Temperature supply	Temperature return
Cooling room	35 % propylene glycol	-7 °C	-2 °C
Ventilation cooling	35 % propylene glycol	-7 °C	-2 °C
Cold process water	Water	0.6 °C	2 °C
Building heating	Water	60 °C	40 °C
Hot process water	Water	80 °C (inlet 10 °C)	to drain
Process steam	Steam	165 °C (6 bar)	condensate

2.2. Identification of existing thermal demands and reference case

The dairy has for several years focused on energy efficiency measures, where one of the active tasks are monthly registration of the electric energy consumption at various locations in the dairy, use of district heating and city water consumption. The logging also includes the measurement of the heat recovery to hot process water from the CO_2 units from the installed energy (flow) meters. As a basis for the current investigation, the logged electric energy consumption of compressors in the CO_2 refrigeration units are used to give an estimation of the cooling delivered to the glycol cycle. This was done through observing the steady-state operation of one 80-kW CO_2 unit at the dairy and logging the relevant operating parameters, including suction and discharge pressures, various temperatures in the CO_2 cycle and on the heat recovery side. The estimation of cooling COP was calculated by inserting the measured operating pressures and temperatures of the CO_2 unit into the software CoolPack[1]. The assumptions and boundary conditions giving a satisfying agreement with the measured values from the CO_2 unit are presented in Table 2 together with the resulting

[1] https://www.ipu.dk/products/coolpack/

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performance characteristics. As a second reference, the operational parameters were used as input to the Bitzer Software[2], yielding good agreement with the calculated values from CoolPack.

Table 2: Relevant input and resulting operational performance of the CO_2 refrigeration unit. Values of suction volume flow and compressor power consumption are given for both the 40 kW and 80 kW units.

Input parameter	**Value**	**Unit**
Suction/discharge pressure	25/95	bar
Compressor isentropic efficiency	0.67	-
Gas cooler outlet temperature	13	°C
Heat loss factor	0	%
Evaporator superheat	8	K
Additional suction superheat	1	K
Pressure losses	0	bar
Suction gas heat exchanger thermal efficiency	0.40	-
Evaporation capacity	40/80	kW
Resulting data	**Value**	**Unit**
Gas cooler inlet/outlet temperature	134/13	°C
Compressor suction volume flow 40/80 kW unit	10.7/21.4	m^3/h
Compressor power consumption 40/80 kW unit	17.2/34.3	kW
Cooling COP	2.33	-

From the resulting COP calculation, the produced cooling to the glycol cycle is calculated based on the logged compressor electricity consumption per month through the year 2021. The calculation assumes that all four CO_2 units perform identically to the 80-kW unit where the measurements were taken at the plant. The two 40-kW CO_2 units can reject heat to the ambient air by a second gas cooler in case there is no need for hot process water and the water storage is fully charged. The switch between the heat recovery gas cooler and ambient air gas cooler is not recorded at the current time, so it is for all cases assumed that both 40-kW units always operates in heat recovery mode, equal to the 80-kW units. Thus, the second gas cooler for the 40-kW units is omitted for the present analysis. Since the temperature setpoint of the glycol and hot process water are constant through the year, the operation of the CO_2 units is virtually unchanged and independent of the ambient temperature. It can therefore be assumed a constant COP for all four CO_2 units through the year. The three consumers of cold in the plant are the process cooling, cooling room and ventilation cooling. Since there is no data available to determine the share of the total cooling load for each consumer, a load distribution has been assumed based on the month of the year. The monthly distribution is presented in Figure 2 along with the measured use of district heating and the electricity consumption of the four CO_2 units. The process cooling is responsible for 55-65 % of the total load, the cooling room makes up 30-35 % and the ventilation cooling the remaining 5-15 %. The share of the cooling room and ventilation cooling is assumed to increase with higher ambient temperatures and is therefore higher during the summer months. Observing the electricity consumption of the CO_2 units throughout the year, more cooling is required during the periods of higher ambient temperature in addition to being closely linked to the amount of processed milk in the dairy. Furthermore, the use of district heating for heating the building is highest for the months with the lowest temperatures. At the location, the mean ambient temperature in January and July was -16 °C and 14.5 °C, respectively.

[2] https://www.bitzer.de/websoftware/

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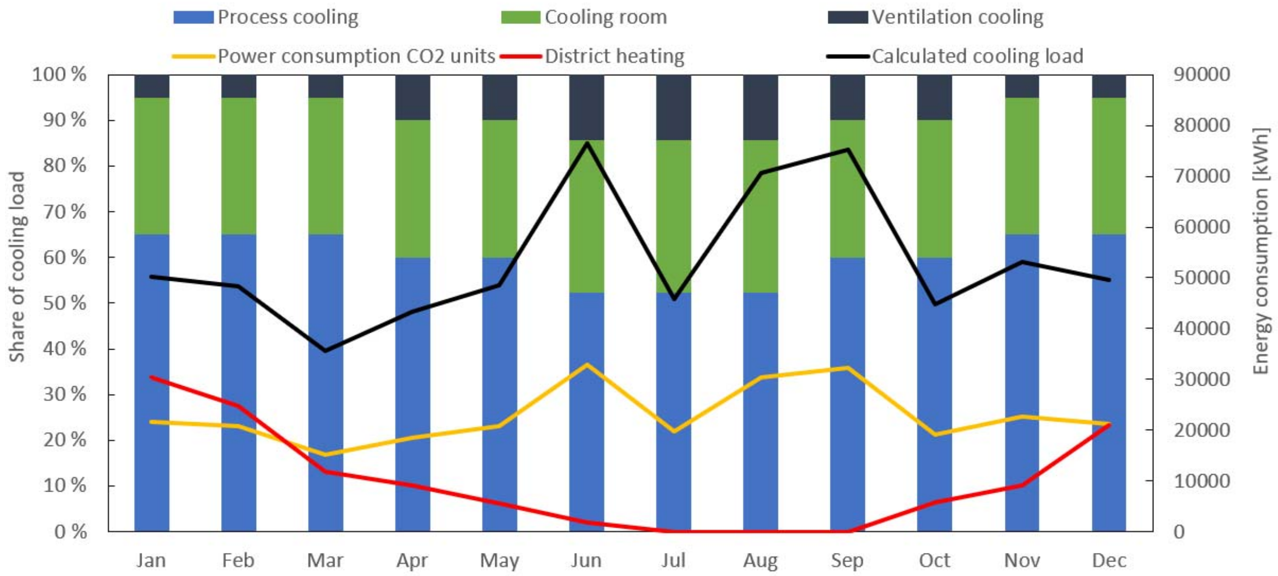

Figure 2: Estimated share of total cooling load for different sources for each month, the reported monthly use of district heating and electricity for CO_2 units and the calculated total cooling load in the dairy.

To propose a new layout of the cold side of the energy system, it is necessary to give an estimation of the required refrigeration capacity for each source throughout the year. This makes sure that the dedicated units have sufficient capacity to satisfy their specified loads. As a simplified approach, it is assumed that most of the cooling load occurs during 12 hours of the day e.g., from 08:00 to 20:00. From the data presented in Figure 2, the monthly thermal demand for each source is calculated. From this, the average cooling load for 12 hours per day over the month is calculated and presented in Figure 3. This is a simplification, and it is expected that short-term peaks will exceed the mean value during certain periods of the day. However, for the analysis here, it is assumed that the TES tanks at each circuit are sufficient to cover these short-term peaks.

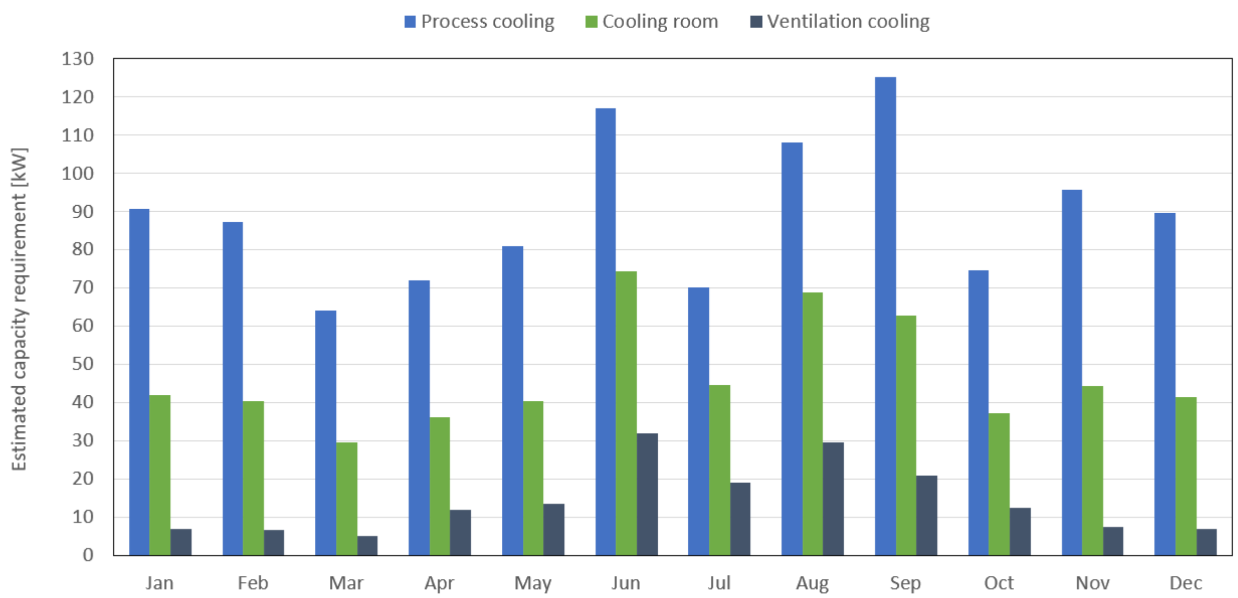

Figure 3: Estimated refrigeration capacity required for each cooling load in the dairy during the year.

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2.3. Proposed system modifications

This section presents the proposed system modifications, named Cases 1-4 in the subsequent text. The focus of this study is on the cold side of the dairy energy system, and on the possible reductions in energy consumption and associated GHG emissions that can be achieved through improving the performance of the installed CO_2 refrigeration units. In the current configuration (reference case, see Figure 1), all the cold is delivered at the setpoint temperature of the glycol of -7 °C. The current configuration is selected based on the simplicity of operation and redundancy, in addition to several modifications to the cold side over the last years. By evaluating the temperature requirements of the consumers, only the cooling rooms require this low temperature. The cooling rooms are kept cold by pumping the cold glycol through air coolers. Consequently, there is a potential for restructuring the cold side of the plant so that the cooling is produced at the highest possible temperature. This will in turn lift the suction pressure of the CO_2 refrigeration units and improve their performance. The proposed modifications for the cold side of the refrigeration units are presented in Figure 4 and described in the following paragraph. Case 1 to 4 represents stepwise changes to a more energy-efficient operation of the CO_2 units e.g., the changes suggested in Case 1 is carried forward onto Case 2 where additional modifications are proposed. The proposed changes need to comply with the estimated refrigeration capacities required for each cooling load, presented in Figure 3.

Case 1 involves separating the CO_2 refrigeration units into two different suction pressures, where unit 1 and 3 (80 and 40 kW) directly cools the cold process water, while units 2 and 4 (80 and 40 kW) remains connected to the glycol cycle like before. For the two CO_2 units connected to the cold process water circuit, the evaporation temperature is lifted from -12 °C (25 bar) to -4 °C (31.4 bar). The process cooling is estimated to account for the largest share of cooling demand in the dairy, so it is expected that this modification can contribute to a significant reduction in energy consumption for the refrigeration system. For Case 2, the proposed change is to move the ventilation cooling from the glycol circuit to the cold process water circuit. This change involves connecting the second 40 kW CO_2 unit to the cold process water circuit so that the only remaining consumers on the glycol circuit are the cooling rooms. In Case 3, the glycol circuit is removed entirely and the cooling of the air inside the cooling rooms is carried out by direct expansion (DX) of refrigerant from CO_2 unit 2 (80 kW). By avoiding the indirect circuit between the cooling room and the refrigeration unit, the evaporation temperature can be lifted from -12 °C to -4 °C, equal to the other three units cooling the cold process water circuit. For the DX refrigeration of CO_2 in the air coolers, an evaporation temperature of -4 °C (31.4 bar) is assumed. Case 4 is the final proposed modification for the cold side of the dairy energy system and involves separating the ventilation cooling into a dedicated chilled water circuit with a supply/return temperature of 7/12 °C. One 40-kW unit (CO_2 unit 4) is dedicated to the new chilled water circuit, operating at an evaporation temperature of 2 °C. In summary, a 5 K temperature difference between the CO_2 evaporation temperature and the setpoint temperature for the cooled media is assumed for all cases. Practical considerations on the presented system modifications in Cases 1-4 are further discussed in Section 4.

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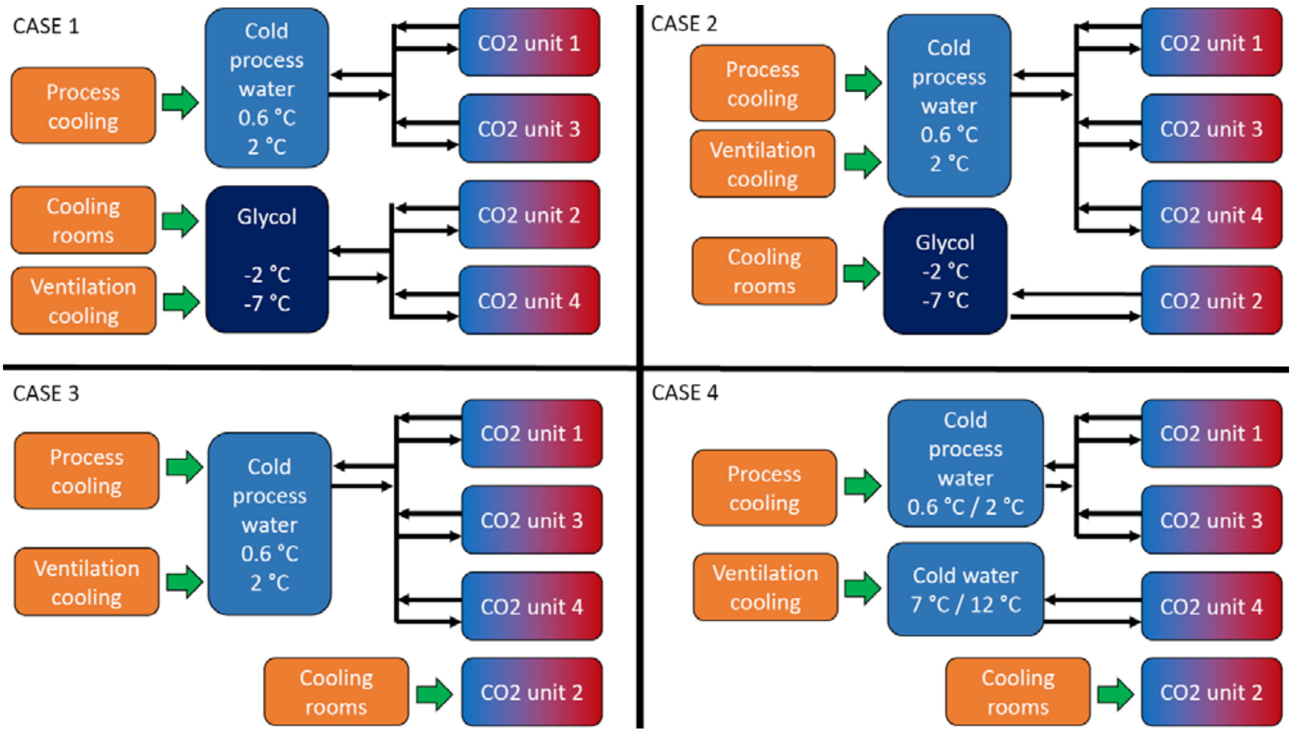

Figure 4: Proposed cold side modification for energy savings in the dairy.

2.4. Calculation procedure

To find the potential reduction in energy consumption of the refrigeration system with the proposed modification in Case 1-4, a calculation procedure has to be established. First the cooling COP for the CO_2 units under the new boundary conditions i.e., elevated suction temperatures are calculated by using the CoolPack software. The boundary conditions and input parameters to the software are kept the same as for the initial COP calculation (see Table 2), except for the new evaporation temperatures. By using the maximum compressor suction flow rate found in Section 2.2, the additional evaporation capacity due to the elevated suction pressure is also calculated for each boundary condition. The monthly cooling demand for process cooling, cooling rooms and ventilation cooling is then calculated based on the total cooling demand and the assumed distribution of cooling load between the consumers according to Figure 2. The monthly electricity consumption for satisfying the various cooling demands at the operating conditions specified for each Case presented in Section 2.3 is then calculated along with the associated CO_2 emissions. For calculation of emission reductions, the CO_2 equivalents for electricity are calculated based on the average carbon intensity in the Norwegian electricity mix 25 g/kWh and European electricity mix 230 g/kWh (EEA, 2021).

3. RESULTS

From the new operational conditions for the CO_2 heat pumps proposed in Cases 1-4, the resulting COP and maximum compressor refrigeration capacity at each condition is presented in Table 3. It can be observed that the COP increases by approximately 25 % and 50 % by increasing the evaporation temperature from -12 °C to -4 °C and 2 °C, respectively. Due to the higher density of the refrigerant vapour at the evaporator outlet, the compressor refrigeration capacity is increased from 80 kW to approximately 100 kW and 120 kW for the two new evaporation temperatures. The increased capacity of the CO_2 units can make the refrigeration system more robust in handling the peak load periods in the plant, as well as allowing for increasing the processing capacity of milk and other products in the dairy without investing in a new refrigeration system.

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Table 3: Results from CoolPack calculation for new boundary conditions for the CO_2 refrigeration units.

Cooled medium: Evaporation temperature / evaporation pressure	COP cooling	CO_2 unit 1 & 2	CO_2 unit 3 & 4
		Suction volume flow rate / compressor maximum refrigeration capacity	Suction volume flow rate / compressor maximum refrigeration capacity
Glycol: -12 °C / 25.0 bar	2.33	21.4 m^3/h 80 kW	10.7 m^3/h 40 kW
Cold process water: -4 °C / 31.3 bar	2.91	16.8 m^3/h 100 kW	8.4 m^3/h 50 kW
Air in cooling rooms: -4 °C / 31.3 bar	2.91	16.8 m^3/h 100 kW	8.4 m^3/h 50 kW
Cooling water ventilation: 2 °C / 37.7 bar	3.51	14.1 m^3/h 120 kW	7.1 m^3/h 60 kW

The results from the evaluation and comparison of Case 1-4 to the reference case is presented in Figure 5 for each month through the year and summarised in Table 4. It is observed that Case 1 provides a reduction in energy consumption in the range of about 10-13 %. Case 2 marginally improves the performance compared to Case 1 during the cold months but become more significant during the summer month due to higher demand for ventilation cooling. It can be seen that switching to DX refrigeration for the cooling rooms (Case 3) can improve the total performance by an additional 6 % compared to Case 2, while only a marginal increase in performance of 1.2 % is found by implementing Case 4.

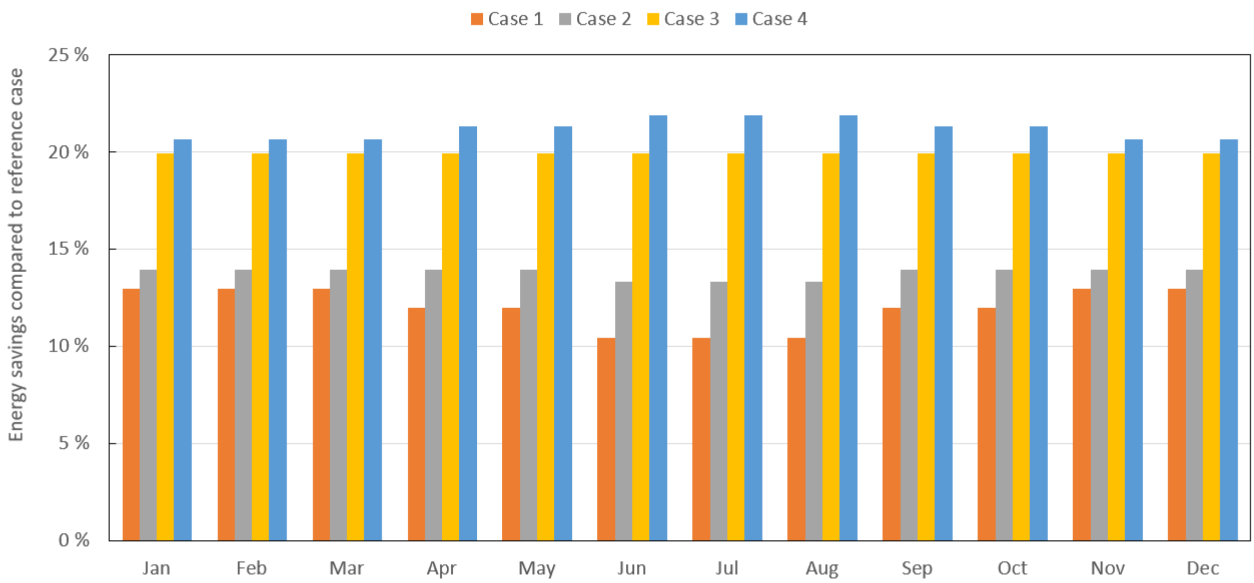

Figure 5: Monthly energy reduction for Case 1-4 compared to the reference case.

Table 4: Summary of results from the study

	Annual electric energy reduction	Annual CO_2 reduction Norway mix	Annual CO_2 reduction European mix
Case 1	12.0 %	0.83 tons	7.62 tons
Case 2	13.8 %	0.96 tons	8.84 tons
Case 3	20.0 %	1.39 tons	12.82 tons
Case 4	21.2 %	1.48 tons	13.65 tons

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4. DISCUSSION

The reason for operating all four CO_2 units on the same circuit in the current configuration is the simplicity of operation and redundancy in case of failure of one of the units. At the current state, the exact distribution of cooling load between the different sources is not known. It is for this reason safer to provide all the cooling from the same glycol circuit regardless of the actual temperature requirement of the cold demand. From the indicated energy-saving potential presented in Section 3, the attractiveness of making modifications to the system in terms of energy savings and related GHG emissions are clearly demonstrated. On the other hand, there is a cost involved in altering the layout of the energy system. The required physical modifications for implementing Case 1 are routing of the main supply pipe and return pipe from the cold process water tank to the location of the CO_2 refrigeration units. From this common manifold, a supply and return line to each of the CO_2 units (unit 1 and 3) must be installed and connected to the existing evaporators on the units. For Case 2, the ventilation cooling is shifted from the glycol circuit to the cold process water circuit. Ventilation systems normally operate at 7/12 °C water circuits supplying the cooling batteries in the air handling units (AHU) under nominal conditions, so operating at a temperature of 0.6/2 °C so be sufficient to obtain the required cooling during summertime. One challenge is the low differential temperature of the cold process water circuit, which increases the needed volume flow of chilled water to the AHUs. The increased volume flow to the cooling batteries might result in a higher pressure drop over the heat exchangers in the circuit. On the other hand, changing from glycol at -7 °C to water at 0.6 °C reduces the viscosity of the circulated liquid significantly both due to the fluid change and the increase of temperature. This translates to less pumping power required by the circulation pumps and lower pressure drop in heat exchangers, in addition to the increased specific heat capacity of water compared to the 35 % propylene glycol solution.

The largest practical changes to the cold side of the energy system are expected to be required for Case 3, shifting from glycol circulation to DX of refrigerants in the cold rooms. A common liquid refrigerant line and suction line have to be routed from CO_2 unit 2 to the location of the cooling rooms in the dairy. New air coolers suited for CO_2 refrigerant with an adequate pressure class have to be installed in addition to electronic expansion valves with associated controllers upstream of each evaporator. The refrigerant charge for CO_2 unit 2 will increase substantially, and a larger liquid receiver needs to be installed in the unit to keep the charge during a maintenance operation on the suction side. On the other hand, additional energy savings due to the removal of the glycol circuit is expected when shifting from a glycol-circulated system to a DX system, due to no need for circulation pumps. For Case 4 the ventilation cooling is shifted to a dedicated water circuit, which requires installation of a water tank for buffering and routing supply and return lines from CO_2 unit 4 and connecting to the existing evaporator.

5. CONCLUSIONS

This study has mapped the energy system in a dairy plant in Norway using CO_2 refrigeration units with heat recovery for providing the required cooling by a common glycol circuit. To improve the performance of the CO_2 units, four stages of modifications of the colds side of the energy system was proposed and the annual energy savings and reductions in GHG emissions were calculated. It was shown that the annual combined electricity consumption of CO_2 units can be reduced by up to 21 % by switching from a common glycol circuit to direct cooling of process water, elevated temperature for ventilation cooling and DX refrigeration for the cold rooms.

Further work in the project is to install flow and temperature sensors on both the cold and warm sides of the energy system for detailed mapping of the energy consumption of the various thermal consumers in the plant. The results from the energy measurement campaign will be used for a more detailed evaluation of the presented modifications of the energy systems and map the potential for installing a cold thermal energy storage system to increase the flexibility and robustness of the refrigeration system.

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