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High-Temperature Heat Pump for Wellness Applications Using CO_2 as a Refrigerant

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ABSTRACT

Building technology recently has been affected by great innovations to reduce energy demand and to enable self-sufficient operation. To test innovations and prove practicality, EMPA has built a research building called NEST, where different demonstration projects from the building industry can be integrated and scientifically monitored. One of the projects focuses on the sector of wellness applications since it shows significant potential for improving energy efficiency. Usually, spa facilities such as saunas and steam baths are based on direct electric or fossil heating, which is extremely energy intensive and results in high operating costs.
In order to establish a more energy efficient technology to provide heat to wellness areas, the present work proposes the usage of a high temperature heat pump with CO_2 as a refrigerant. Carbon dioxide is currently favored by many heat pump applications due to the demand for low-GWP refrigerants with non-toxic and non-flammable properties. High-temperature heat pumps with CO_2 as a refrigerant are already available on the market and reach supply temperatures of typically up to 100 °C. Wellness applications require temperatures of up to 130 °C on the supply side, which requires new system design. On the consumption side, the heat will be used on a wide variety of temperature levels considering different saunas, showers and space heating. In this context, a stratified storage system ensures the heat output on the desired temperature level. Therefore, the heat exchanger unit as well as the operating range of the heat pump are the major challenges.
In cooperation with Scheco AG, a new CO_2 refrigerant system has been designed and set up as a part of the EMPA NEST research building by means of a pilot installation including different saunas, steam bath and showers. Measurements are performed to allow the optimization of the overall system.

Keywords: High-Temperature Heat Pump, Carbon Dioxide, System Design, Sauna, Energy Efficiency

1. INTRODUCTION

1.1 Situation

The operation of wellness facilities is very energy-intensive and owners are therefore increasingly criticized for being excessively contributing to global warming. This is due to the fact that almost exclusively direct electrical heating or, more rarely, the combustion of fossil fuels is used for heat supply. Spa facilities typically include a finnish sauna (90 °C, low humidity), bio sauna (55 °C, 60 % relative humidity) and a steam bath (45 °C, up to 100 % relative humidity). Typically the finnish sauna requires 30 % of the energy, the bio sauna and the steam bath both 35 %, whereas the total energy consumption for a spa facility for 4-6 people is around 123’000 kWh/a.

In addition to the correct temperature and the appropriate moisture level, the temperature stratification in the sauna room is important and high convection and radiation are undesirable in the finnish sauna. Particularly, there is a large heat demand on the high temperature level due to the production of steam. Within this project, an energy-saving wellness facility was developed and built, which was ecologically improved by the following means:

- Efficient heat generation using a high-temperature CO_2 heat pump instead of direct electrical heating
- Improved thermal insulation to reduce transmission losses
- Heat and moisture recovery from sauna and steam bath to minimize ventilation losses
- Energy-saving "steam" generators by using heat at temperatures <100 °C
- Use of photovoltaics and thermal tube collectors to reduce the remaining energy consumption

The focus of the present work is on the heat generation using CO_2 heat pumps. A new type of CO_2 heat pump is developed by minimizing exergy losses and optimizing the use of energy over a wide temperature range (30-130 °C) by means of cascade utilization. The project aims to achieve a seasonal performance factor of 3. In addition, further efforts will reduce energy consumption by 50 %. Thus, compared to conventional systems, the energy requirement can be reduced by more than 80 %, leading to significantly lower life cycle costs. Figure 1 shows the idea of using heat on different temperature levels.

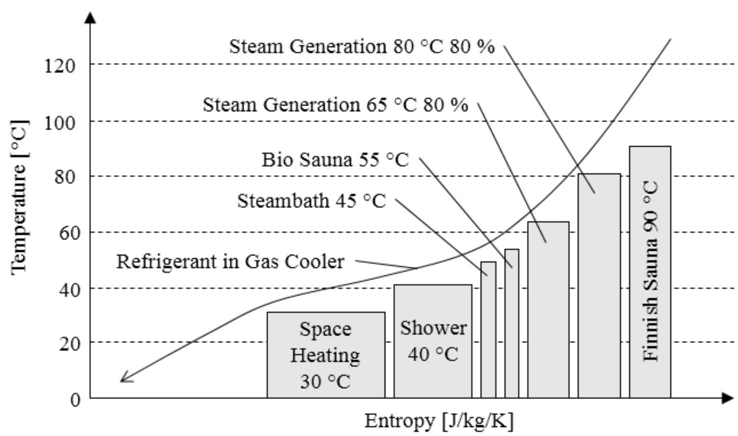

Figure 1: Basic idea of heat utilization on different temperature levels due to gas cooling of CO_2 (pinch method)

In order to advance the research on the building and energy sector to market as quickly as possible, the EMPA research center has set up a building platform called NEST (Next Evolution in Sustainable Building Technologies). This means that new technologies are tested under real conditions and the innovation process is accelerated. The test system of the investigated CO_2 system is demonstrated in this platform together with the wellness equipment.

The aim of this work, in cooperation with Scheco AG, was to design and build an appropriate CO_2 heat pump, which is capable of providing heat at temperatures of up to 130 °C.

1.2 Carbon Dioxide as a Refrigerant and High-Temperature Heat Pumps

As a result of stricter laws on environmentally harmful refrigerants, natural refrigerants are gaining more and more interest these days. Those refrigerants include ammonia (toxic), which is often used in refrigeration, propane and isobutane (both flammable) and carbon dioxide (also CO_2 or R744), which is preferred in many applications because of its non-flammability and non-toxicity (Lacroix, 2004). Furthermore, there is no ODP and the GWP is negligible. The volumetric heat capacity of CO_2 is 3-10 times higher than synthetic refrigerants. Further advantages are high heat transfer coefficients, availability and therefore low refrigerant costs (Kim, 2004). However, the high working pressures of CO_2 (critical point at 31.1 °C and 73.7 bar) as well as the slightly lower COP in many applications are disadvantageous.

Supercritical CO_2 cycles with separate gas coolers offer an elegant possibility for multi-temperature heat pumps and have recently gained increasing attention for hot water production and air heating (Arpagaus, 2016, Austin, 2011). Neksa (1998) has already experimentally investigated a transcritical CO_2 cycle at an early stage. The plant was able to heat water from 8 °C to 60 °C with a COP of 4.3 (evaporation at 0 °C and high pressure at 90 bar). Simultaneous space heating and water heating for residential buildings was investigated experimentally by Neksa (2002) and Stene (2005). The refrigerant cycle includes a low pressure receiver and an internal suction gas heat exchanger. A triple subdivision of the gas cooler with a total heat output of 6.5 kW allows DHW heating from 6.5 °C to 80 °C (max. high pressure 95 bar) and space heating (33/28 °C) with a COP of 3.6 (Stene, 2005).

Transcritical CO_2 heat pumps for DHW heating are particularly popular in Japan due to their environmental compatibility (Zhang, 2015). According to Hashimoto (2016), maximum achievable output temperatures of commercially available industrial heat pumps for water heating at 90 °C are possible, whereas higher temperatures of 120-175 °C are feasible for hot air and steam generation. One of the heat pumps available on the market heats air from 20 °C to 120 °C (heating capacity is 89 kW) at a COP of 3.1 with water at a temperature of 30 °C as a heat source (Mayekawa, 2017). Another Manufacturer of CO_2 high-temperature heat pumps claims to reach a maximum supply temperature of 110 °C with heat source temperatures between 8 °C and 40 °C (Dürr thermea, 2016). One of their machine types achieves a COP of 4.3 while heating water from 20 °C to 80 °C, whereas the heating capacity is 51 kW. Other technologies are being used on the market to provide hot water with supply temperatures of 65-90 °C by using a two-stage screw compressor with a refrigerant mixture of R134a and R245fa with air as a heat source (Oue, 2013).

In this context, there is no suitable product in the desired power range (<20 kW) available on the market. Therefore, research potential exists.

2. SIMULATION

2.1 Heat Demand of Saunas

In order to determine the energy requirements of the three different saunas, a dynamic simulation model was created in EES (Klein, 2018). On the one hand, the normal operating mode (sauna is already heated up) and on the other hand the heating-up phase was considered. The geometry of the sauna was assumed to be a cube, with the finnish sauna being divided vertically into 5 elements in order to examine the temperature stratification in detail. The air volume in the other saunas is considered to be homogeneous (good air mixing) which resembles the actual situation well. Various materials for floors, walls and ceilings with the corresponding U-value were provided by the sauna builder. Figure 2 shows the modelled sauna room with its specific heat fluxes.

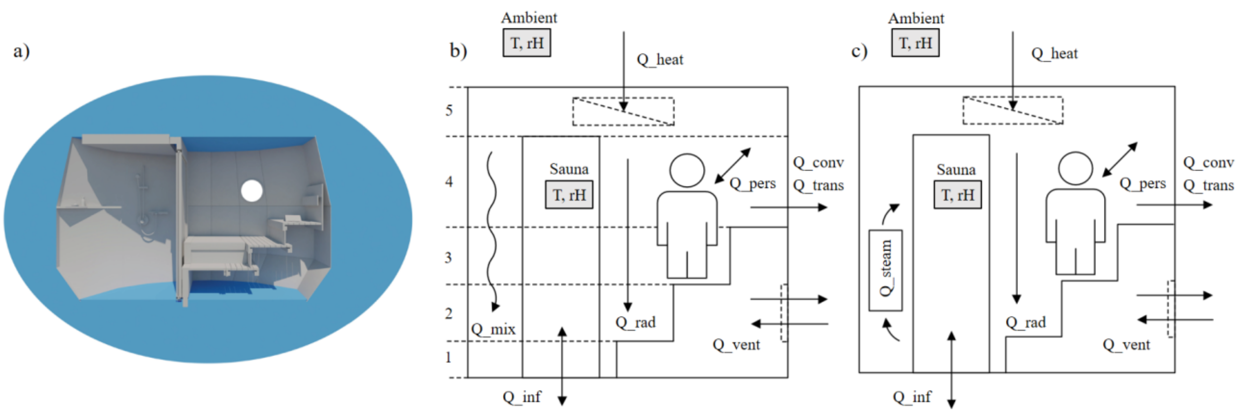

Figure 2: CAD Model of the Saunas (a) and the definition of the basic simulation model (b, c)

The most important energy flows are taken into account in the energy balance within the system boundary. The transmission losses due to heat conduction of the sauna shell are calculated according to the following correlation.

$$\dot{Q}_{trans} = \left(\sum \frac{s}{\lambda}\right)^{-1} A\,(T_{amb} - T) \tag{1}$$

The heat transfer of the air convection inside the sauna and on the outer wall is calculated by means of empirical correlations implemented in EES (Klein & Nellis, 2009). In the steam bath, drop condensation has to be taken into account due to the high humidity and low air temperature in the sauna, increasing the heat transfer values.

$$\dot{Q}_{conv} = \alpha_{air}\, A\, (T - T_{wall}) \tag{2}$$

In addition, heat radiation from the ceiling to the floor and seats must be included, which represents a significant proportion of the energy exchange. The view factors describe the position and orientation of the surfaces.

$$\dot{Q}_{rad} = \sigma\, F\, A\, (T^4 - {T_{wall}}^4) \tag{3}$$

When the door is opened, a large part of the humidity in the sauna is lost and the temperature decreases due to air exchange with the surroundings. A constant value of 0.1 m/s is assumed for the air velocity.

$$\dot{Q}_{inf} = \dot{m}_{inf}\, (h_{amb} - h) \tag{4}$$
$$\dot{m}_{inf} = u_{air}\, A_{door}\, \rho_{air} \tag{5}$$

Further heat losses are generated by the exhaust air and fresh air supply (with heat recovery efficiency of 50 %) which are also calculated according to (4). The air ventilation rate is set to three air changes per hour. In addition, the sauna guests absorb heat which is calculated according to (2) and sweating adds additional moisture to the room which is being evaporated by requiring heat. In the case of the finnish sauna, the heat exchange between the different temperature levels due to mixing is also taken into account according to (4) and (5) with a constant velocity of 0.1 m/s. Input parameters are the target temperature and humidity in the sauna as well as the constant settings of the system. For each time step, the current state of the air is calculated based on the enthalpy difference, which results in the humidity and the required heat flux.

Table 1: Constant input parameters for the simulation of the finnish sauna

Geometry	Length 2.2 m, width 2.1 m, height 2.1 m
Heat Exchanger	UA-value 0.25 kW/K
Wall and Ceiling	Wood paneling, wood construction, insulation
Floor	Ceramic, adhesive, insulation XPS
U-Values	Floor 1 W/m^2/K, ceiling 0.3 W/m^2/K, inner and outer walls 0.3 W/m^2/K, door 2.1 W/m^2/K
Mass	Floor 210 kg, walls and ceiling 405 kg, window and seats 240 kg
Emissivity	Ceramic floor 0.91, wooden walls 0.8, ceiling 0.98
Humans Properties	Skin const. 40 °C, sweating 300 ml/h, heat transfer 0.01 kW/m^2/K, surface 1.8 m^2
Number of People	Minimum 0, Maximum 4

In the finnish sauna, the temperature in the fourth element with most of the people should be at 90°C. The heat is brought into the uppermost element under the ceiling by means of a simulated heat exchanger, whereby a UA-value is given and the supply temperature is set to 115 °C (calculation of the heat flow via the logarithmic temperature difference). Figure 3 shows the temperature profile and heat demand of the finnish sauna in operation when all disturbances are active. A time profile was assumed for the door opening (duration of opening is 15 s, 30 s, 2 min and 4 min).

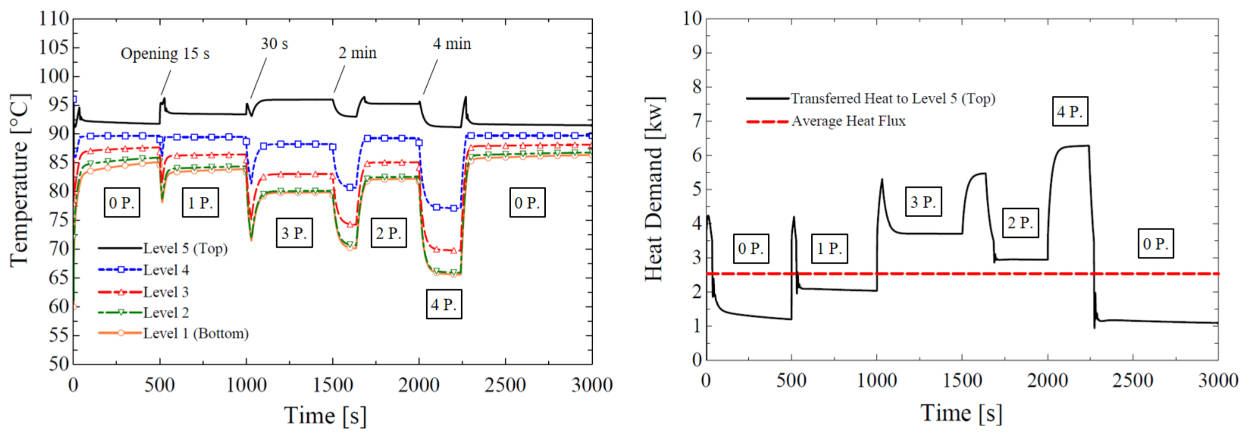

Figure 3: Temperature profile and heat demand in the finnish sauna

Since the temperature in element 4 is controlled, the top layer of air under the ceiling will become warmer after a door closing since more heating is required (the door is not included in the top layer and no infiltration takes place). All temperature decreases are caused by the door openings. After the last door opening, only a small temperature stratification can be detected due to air mixing and radiation exchange in between the zones. The mean value of the heat demand is 2.58 kW.

During the heat-up process of the finnish sauna from ambient temperature to 90 °C no disturbances such as door openings are assumed and the ventilation is switched off. When using the high temperature level heat from the storage at 115 °C, the warm-up takes 58 min and requires 3.2 kWh. Alternatively, the sauna can be preheated to 58 °C with the medium temperature level of the storage (supply 90 °C, return 60 °C) to reduce high temperature energy usage. In this context, the preheating takes 16 min and requires 0.8 kWh from the medium temperature level. The high temperature heat consumption is reduced to 2.4 kWh but the heating-up time is extended by 6 min.

In the steam bath, the operating state is generated exclusively by the steam generator. In contrast to the finnish sauna, no additional heat exchanger is used to support air heating. For the simulation, the "steam generator" is assumed to blow hot air of 80 °C with a relative humidity of 80 % into the sauna room. In Reality, a tank filled with deionized water is heated up to approximately 80 °C. Then, pressurized air taken from the steam bath flows through the water tank and absorbs water to increase humidity. The "steam" is then supplied to the steam bath at 60-70 °C and 80 % relative humidity. Furthermore, relative humidity has a higher priority compared to temperature in the control algorithm of the steam bath. This means that if the humidity is too low, the temperature can rise above the target value. In order to avoid too high temperatures, the supply of fresh air is increased in real operation. In this model, it is also necessary to take the condensation of water from the air into account (on the sauna visitors, on the wall and when mixing with cooler air).

Most of the constant input parameter for the steam bath can be taken from Table 1. New values are defined for materials and masses (various), the U-values (ceiling 0.5 $W/m^2/K$, inner and outer walls 0.5 $W/m^2/K$) as well as the removal of the air heat exchanger. The desired condition is 45 °C and 97.5 % relative humidity, whereas the steam injection is controlled by a hysteresis (±1.5 %). Although, in real operation the moist air is supersaturated, in order to minimize the computing effort, a maximum humidity of 100 % is specified in the calculation (heating to condensation of the surplus humidity). Figure 4 shows the air condition and heating requirement of the steam bath in steady state when all disturbances are active. It can be seen that the influence on the air temperature only becomes noticeable through longer door openings (>1 min) but the humidity changes very quickly. Therefore, a high steam volume flow of 18 m^3/h has been selected to compensate losses quickly. If the air change rate is chosen to be lower, the steam volume flow rate can be decreased as well. The wavy behavior comes from switching the steam generator on and off. The required steam output in continuous operation is 3.8 kW.

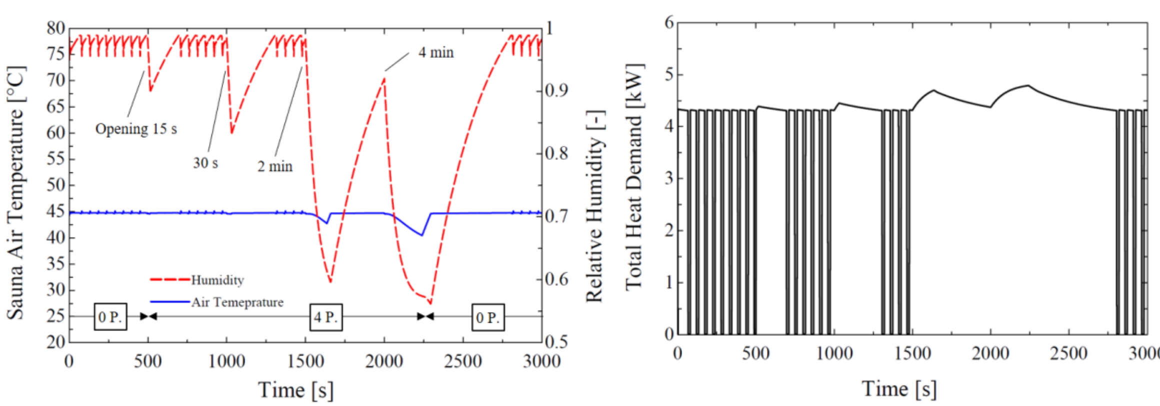

Figure 4: Temperature and humidity profile in the steam bath as well as heat requirement

During the warm-up phase, the steam generator delivers its full capacity. A condensation rate of 0.5 kg/h is assumed to take into account for the condensation on the walls. The steam bath reaches the desired state after 16 min and requires 0.9 kWh (steam volume flow of 12 m^3/h). The desired room temperature of 45 °C is reached earlier than the required humidity and thus achieves a maximum value of 54 °C.

The third simulation model describes the bio sauna and is also examined without temperature stratification. In this case, the humidity is lower than in the steam bath. Therefore, a different method is used for “steam generation”. Identically to the finnish sauna, the bio sauna has a heat exchanger for air heating. In the bio sauna, cold sauna air close to the ground is being transported by a fan (6 m^3/h) and humidified by spraying deionized water before entering the heat exchanger (cooling process). The heat exchanger then heats up the moist air. This mechanism enables a low return temperature due to the air cooling. The UA-value is assumed to be 0.18 kW/K (supply 90 °C and return 60 °C). In contrast to the steam bath, the temperature has a higher priority in the control strategy. The results used for the design of the heat pump are summarized in Table 2.

Table 2: Summary of the simulation results

	Finnish Sauna	**Bio Sauna**	**Steambath**
	90 °C **Heat Exchanger**	55 °C, 60 % **Heat Exchanger**	45 °C, 100 % **Steam**
Operation	2.6 kW	2.9 kW	3.8 kW
Warm-up	3.2 kWh (58 min)	1.0 kWh (62 min)	0.9 kWh (16 min)

2.2 Refrigerant Cycle

To estimate the heating capacity and efficiency of the heat pump, simplified thermodynamic calculations are performed in EES. The investigations apply to a simple transcritical CO_2 cycle with the following basic assumptions:

- Source temperature 15 °C
- Superheating 10 K
- Minimum temperature difference at pinch point 5 K
- Isentropic efficiency of the compressor of 0.6

Based on these assumptions the following simulation has been carried out. As shown in Figure 5, the gas cooler outlet temperature of the refrigerant has a direct influence on the COP. This means that a low return flow temperature is a key requirement for efficient operation of the system. In order to achieve the desired temperatures of 130 °C on the supply side, the high pressure should theoretically be regulated to a pressure of 140 bar. Therefore, the refrigerant must be cooled to at least 36 °C so that the COP is higher than 3. Table 3 shows the heat distribution over the different temperature levels. For this purpose, a suction gas volume flow of 1.5 m^3/h for the compressor was assumed (equivalent to a capacity reserve of approx. 40 %).

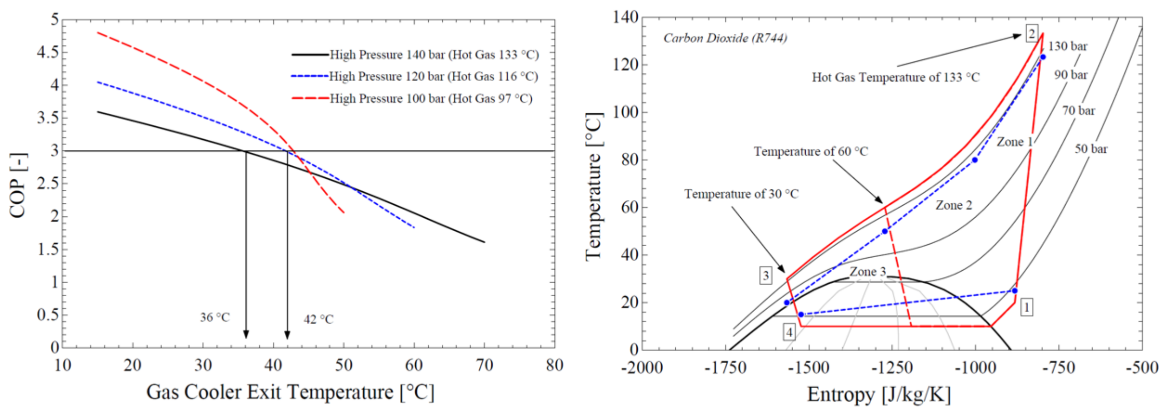

Figure 5: Investigation of heat pump efficiency

A large part of the heat is required at temperatures >60 °C. This part decreases considerably with lower gas cooler pressure. According to table 3, at 100 bar, the heat output at the high and medium temperature level is only 3.6 kW in comparison to 8.4 kW at a pressure of 140 bar. With lower source temperature, the hot gas temperature increases but the efficiency gets lower. Therefore, the design point of the heat pump was defined at 119 bar. A higher degree of superheating ensures that the desired temperature of 130 °C can still be achieved.

Table 3: Theoretical performance assessment for a compressor with a volume flow rate of 1.5 m^3/h

Zone	**Refrigerant Temp. Range**	**Heat Capacity at different Discharge Pressure**		
		140 bar	**120 bar**	**100 bar**
1	>90 °C (finnish sauna)	29.5 %	18.2 %	5.0 %
2	90-60 °C (bio sauna and steam bath)	35.2 %	33.9 %	27.7 %
3	60-30 °C (showers and space heating)	35.3 %	47.9 %	67.3 %
Total	>30 °C	13.0 kW	12.1 kW	11.1 kW

3. SYSTEM DESIGN

The storage and distribution system is key element in plant engineering. One way of providing the heat at the right temperature level is to use multiple storages with standardized components. The high temperature storage tank for the finnish sauna should reach 120 °C with the first priority. Thermal oil had to be avoided in the entire circuit due to safety reasons. This means, subsystems with temperatures >100 °C must use a higher line pressure of up to 4 bar to avoid boiling (not an unusual value since buildings with multiple floors typically run 2.5-3 bar to avoid cavitation). In the domestic hot water storage tank, the low return flow temperature of fresh water (approximately 15 °C) should be used to increase the efficiency of the heat pump by means of achieving a low gas cooler exit temperature. The hot water storage for the bio sauna and the steam bath can in principle be implemented with a common storage tank (max. 100 °C, min. 60 °C). However, the separation into two individual storages allows the use of standard components and an independent operation of the two saunas. The control strategy is designed in such a way that the charging pump is switched on as soon as heat is required in the corresponding storage tank and the temperature falls below a minimum value.

Another approach is, to use only one large storage tank with high temperature stratification. Due to the cost savings and easier space utilization, this option was implemented in the end. Disadvantages are the thermal insulation (constant thickness on top and bottom) and the part load operation is more difficult. Additionally, the complete circuit is at increased pressure. For the integration of the CO_2 heat pump, a heat network with different temperature levels is available as a backbone for thermal management within the NEST building platform (high temperature network with 45 / 65 °C, medium with 28 / 38 °C and low temperature with 5 / 28 °C). When using the energy network, the temperature specifications must be considered and a pump must be provided to the consumer side. The evaporation heat is thus taken from the low-temperature level, which is primarily used for cooling and as a heat pump source. At the same time, excess energy is transferred to the medium temperature network. This is primarily used for space heating in the building. The volume flow of the water pump must be adjustable so that the return flow temperature does not exceed the maximum value. The complete schematic, which has been elaborated by Scheco AG, is shown in Figure 6.

The transcritical CO_2 cycle contains three gas coolers on the high-pressure side at different temperature levels. As soon as heat is required at the corresponding temperature level, the heat exchanger is switched on by increasing the volume flow of the pump. When the pump is switched off, the heat is delivered to the next heat exchanger. The third gas cooler is used for space heating in the wellness area and for emergency cooling as soon as the pressure in the CO_2 circuit becomes too high due to high room temperature at standstill. The heat pump is controlled by an electric expansion valve which regulates the high pressure. On the suction side, a large receiver is installed as a liquid separator. This protects the compressor from liquid slugging and is used as an expansion reservoir to prevent the high pressure of CO_2 during standstill. An additional internal heat exchanger ensures proper superheating. On the market, there is only a small variety of CO_2 compressors in the desired capacity range. Thus, a small variable speed compressor with a suction flow rate of 3.3 m^3/h was chosen. The excess heat is transferred to the building system. Maximum allowed discharge pressure is 129 bar, whereas the operating pressure is chosen to be as low as possible to ensure maximum efficiency. Furthermore, a special design was used for the low pressure receiver (35 dm^3).

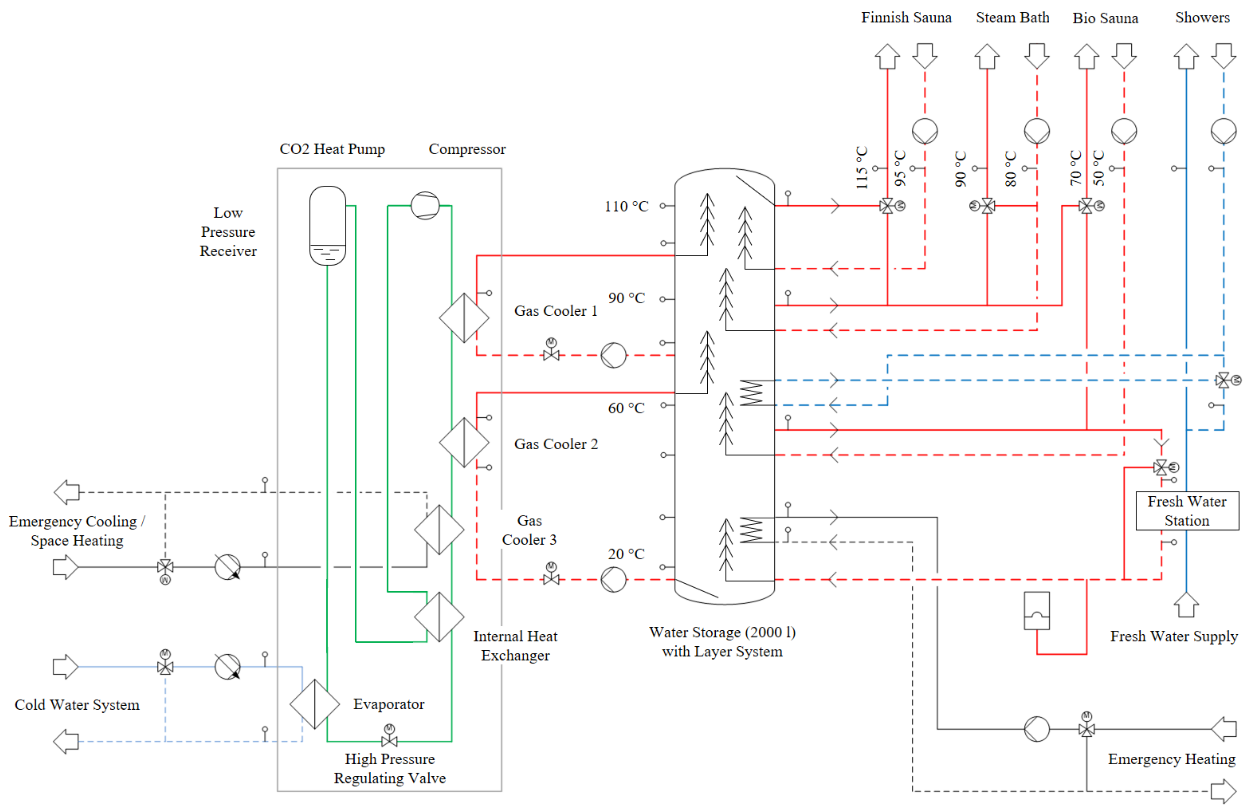

Figure 6: Hydraulic schematic of the overall system

A proper temperature stratification in the storage tank is required for efficient operation. In order to guarantee this, vertical lances are provided in the storage vessel at various water inlets, where the water is injected at the appropriate height due to the temperature dependent density. On the consumer side, the heat required for the various saunas is taken from the storage tank at the appropriate temperature level. Internal heat exchangers are designed for heating domestic hot water during circulation and for emergency heating.

4. INITIAL OPERATION

After assembling the CO_2 heat pump, the first measurements were carried out by Scheco AG on the test bench at the Zurich University of Applied Sciences (ZHAW) to verify the heat pump before the installation in the wellness area. The measurement results in Table 5 show that a supply temperature of 119 °C has been reached. Approximately 26% of the heat output could be given to the high temperature level in gas cooler 1 (test 1). Also, the target value of the COP of 3 could partially be achieved (test 1 and 2). Operating pressure of test 1 corresponds to a discharge pressure of 117 bar and test 2 to 96 bar.

Table 4: Results of measurement performed on the test stand at Zurich University of Applied Sciences ZHAW

Variable	Unit	Design	Test 1	Test 2
Heat Capacity of Gas Cooler 1	[kW]	6.0	4.7	0.7
Water Inlet	[°C]	82	86	89
Water Outlet	[°C]	120	119	93
Heat Capacity of Gas Cooler 2	[kW]	13.0	13.3	13.7
Water Inlet	[°C]	35	35	24.5
Water Outlet	[°C]	74	74	61
Capacity of Evaporator	[kW]	13.7	14.2	11.5
Water Inlet	[°C]	17	16	19
Water Outlet	[°C]	12	11	15
Electric Input Power	[kW]	5.7	6.3	4.0
COP	[-]	3.3	2.9	3.6

The heat pump was then put into operation in the wellness area. After optimization of the water flow rate, a maximum flow temperature of 110 °C could be achieved (storage temperatures in Figure 7). Therefore, the finnish sauna is basically operative, with the temperature difference between top and bottom of the sauna room being approximately 30 °C (high radiation of the ceiling). The high floor temperature of up to 60 °C leads to very high return temperatures (bad for the efficiency of the heat pump and the well-being of the sauna visitors). The steam bath requires a large part of the energy, whereas the high condensation rate in the sauna room is a problem. Although the top temperature is 60 °C (dry air), the lower room is at 25 °C where the seats cannot be heated enough. There were no difficulties with the bio sauna, where the temperature stratification is approx. 10 °C. As can be seen in Figure 7, the target value of 130 °C has been achieved on the supply side.

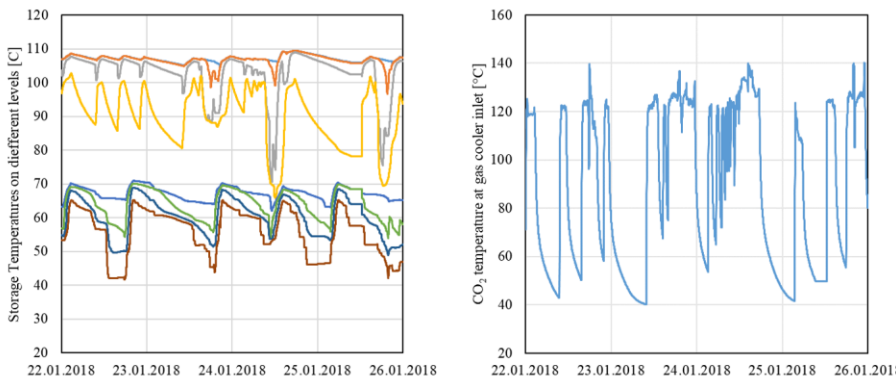

Figure 7: Water storage and CO_2 Temperatures during operation of the spa facilities

System optimization is currently being worked on to reduce the return flow temperature and to improve the air condition in the sauna rooms. Photographs of the saunas and the heat pump are shown in Figure 8.

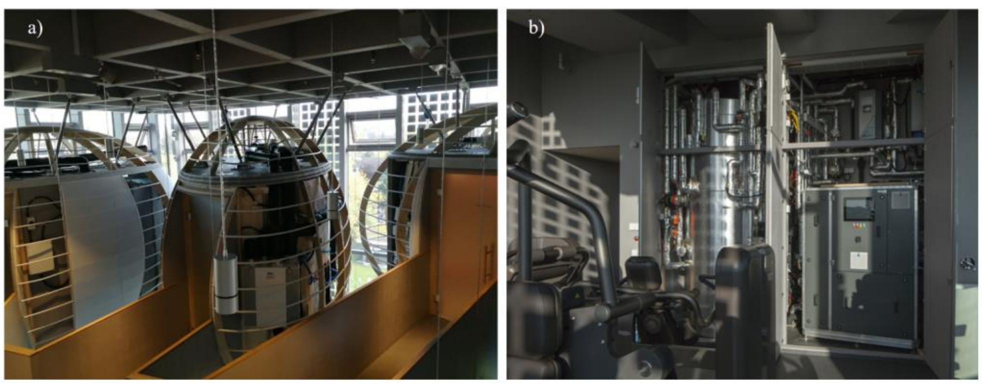

Figure 8: Photograph of the hanging saunas and the CO_2 heat pump

5. CONCLUSIONS

Research has shown that there is currently no heat pump available on the market that can be used in wellness facilities to provide high temperature heat. Within the present work, a suitable heat pump with natural refrigerant has been developed, installed and tested. The system is convincing in terms of high temperatures on both the supplier (120-140 °C) and consumer side (<119 °C). Depending on the discharge pressure, the heating capacity on the high temperature level is 0.7-4.7 kW and on the medium temperature level 13.3-13.7 kW. A maximum COP of 3.6 has been reached. There is potential for improvement to reduce return flow temperature for a higher efficiency of the heat pump. The following conclusions are drawn from this study:

- Transcritical CO_2 heat pumps are a feasible solution for high temperature heat
- Low return temperatures are essential for efficiency
- The application in the wellness area successfully proves that implementation already works today
- Efficiency depends on both the heat pump but also system integration

NOMENCLATURE

A	Surface in general	(m^2)
A_{door}	Surface of sauna door	(m^2)
F	View Factor of emissivity	(-)
h	Enthalpy in general	(J/kg)
h_{amb}	Enthalpy of ambient air	(J/kg)
$\dot{m}_{inf}$	Infiltration mass flow	(kg/s)
$\dot{Q}_{trans}$	Transmission heat flux	(W)
$\dot{Q}_{conv}$	Convection heat flux	(W)
$\dot{Q}_{rad}$	Radiation heat flux	(W)
$\dot{Q}_{inf}$	Infiltration heat flux	(W)
s	Thickness	(m)
T	Temperature in general	(°C)
T_{amb}	Ambient Temperature	(°C)
T_{wall}	Wall Temperature	(°C)
u_{air}	Air velocity	(m/s)
α_{air}	Heat transfer of air flow	($W/m^2/K$)
λ	Heat Conductivity	(W/m/K)
ρ_{air}	Density of air	(kg/m^3)
σ	Stefan-Boltzmann constant	($W/m^2/K^4$)

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