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# Heat Pump with Two Heat Sources on Different Temperature Levels

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## OptiRef: Heat Pump with Two Heat Sources at Different Temperature Levels

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### ABSTRACT

Aim of this project is the development of a new heat pump system with economizing that is able to improve the heating performance using two or more heat sources at different temperature levels. These heat sources are incorporated into the system with minimal loss of exergy by adding the heat at different pressure levels. Practical applications for such systems are for example buildings with a heat pump and a solar thermal collector or buildings with waste heat recovery. While solar thermal systems can be used for heating and domestic hot water in summer, they fail to produce sufficient temperatures for direct use in fall, spring and winter. By connecting the solar collectors to the here described heat pump system, they are able to improve the heat pump efficiency significantly. Furthermore, the lower temperatures of the collectors help to improve their effectiveness and raise the usable solar gains. While existing systems using two heat sources are either very costly, inefficient or need large amounts of waste heat, the proposed cycle can use varying amounts of waste heat and increases the heat pump efficiency by up to 30% over a wide range of operating conditions.

The paper presents an introduction to the topic and various heat pump system designs using two heat sources. All of them incorporate the OptiRef. Simulation results and experimental analysis show that depending on the amount of waste heat and the temperature level of the heat pump cycle, efficiency and heating capacity improvements of 20-30% could be shown in a laboratory prototype. Oil management and control of the system are the main challenges when implementing it in the laboratory or field. Altogether, the functionality of the system could be proven, but there is still room for improvement with respect to the control concept and in order to reduce cost.

**Keywords:** heat pump, two heat sources, solar thermal, high efficiency, open economizer

## 1. INTRODUCTION

### 1.1 Background

Integrating solar heat with heat pump systems has been a major topic of several projects in central Europe. This technology is relevant in several aspects since it achieves a high fraction of renewable energy, and allows for a reliable heating system. Task 44 “Solar and Heat Pump Systems” of the International Energy Agency investigates different methods of integrating solar thermal and heat pump systems (Frank et al., 2010; Hadorn, 2010).

The standard solution for the past two decades for the combination of heat pump and solar thermal systems has been to heat up one common hot water tank (Kjellsson et al., 2010). This means, that in summer the hot water tank is heated up by solar radiation and in spring, fall and winter the heat pump is delivering the heat. Downsides are the low solar fraction for heat generation and significant cost increase of the system. Another approach is seasonal storage like borehole heat regeneration during summertime (Bertram et al., 2011 and 2012) or ice-water storage systems (Loose et al., 2012) where researchers have tried to enhance the seasonal efficiency of the heat pump. The downside of these methods is that they store heat at very low temperature (mostly freezing point of water). The average air temperature in Switzerland lies between 5-10°C compared to a temperature of 0°C for ice-storage concepts. This leads to quite low seasonal performance factors (SPF) of the heat pump with ice storage.

Other concepts rely on the direct integration of solar heat as the only heat source (Ito et al., 1999; Yang et al. 2011). Their major problem in implementation is the shortage of solar radiation during winter time when the heat demand is largest. Additionally, most of these heat pump systems encounter problems, when the evaporating temperature exceeds 20°C. For this reason such systems are usually mostly employed for hot water supply of a building and not for heating.

Therefore, there is a need for better concepts of integrating solar power into heat pump cycles. The approach presented here is following the Patent of Granwehr (2013). Granwehr proposes a heat pump system that can incorporate several heat sources at different temperatures. Major application will be the retrofit of existing heating systems, since the proposed cycle shows advantages in efficiency, especially at higher condensing temperatures (40-60°C). The system is not bound to the integration of solar heat, but can be used for other additional heat sources such as waste heat recovery as well.

## 1.2 System description

The heat pump system presented in this paper basically relies on a common refrigeration cycle with an open or closed economizer. Figure 1 presents the concept featuring a schematic with an open economizer along with the according ph-diagram. The heat injection from the second heat source is illustrated on the left hand side labeled  $\dot{Q}_{Optiref}$  where heat is added to the open economizer. This component is the major difference in comparison to a standard open economizer cycle since it allows injecting heat from the second heat source at the intermediate pressure level. From here on we will call this component OptiRef for easy identification.

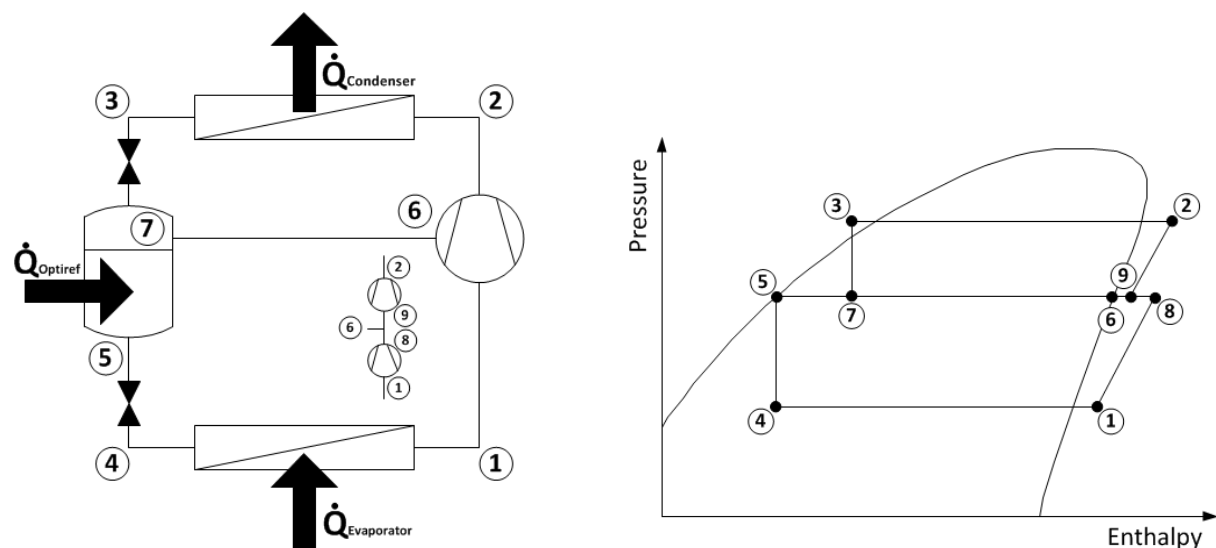
The OptiRef is basically a vessel at intermediate pressure with an internal heat exchanger for the heat addition from the second heat source. It features three refrigerant ports:

- One inlet near the top where the two-phase refrigerant from the condenser and following expansion valve enters,
- One outlet at the bottom for the liquid refrigerant that passes towards the downstream expansion valve and the evaporator, and
- One outlet high up in the vessel that allows gaseous refrigerant to exit towards the injection line at intermediate pressure.

Besides allowing heat from the second heat source being added to the refrigeration circuit the OptiRef also acts as a refrigerant receiver adjusting for different refrigerant charge at varying operating conditions.

Major advantage of this system is the fact, that the refrigeration system is fully operational at varying heat input from the second heat source. The more heat we inject at the intermediate pressure level, the higher that amount of gaseous refrigerant out of the OptiRef. This then reduces the electrical power consumption, since some refrigerant only needs to be compressed from the intermediate pressure level on. Altogether, the electrical power requirement is reduced and the heat output at the condenser increases.

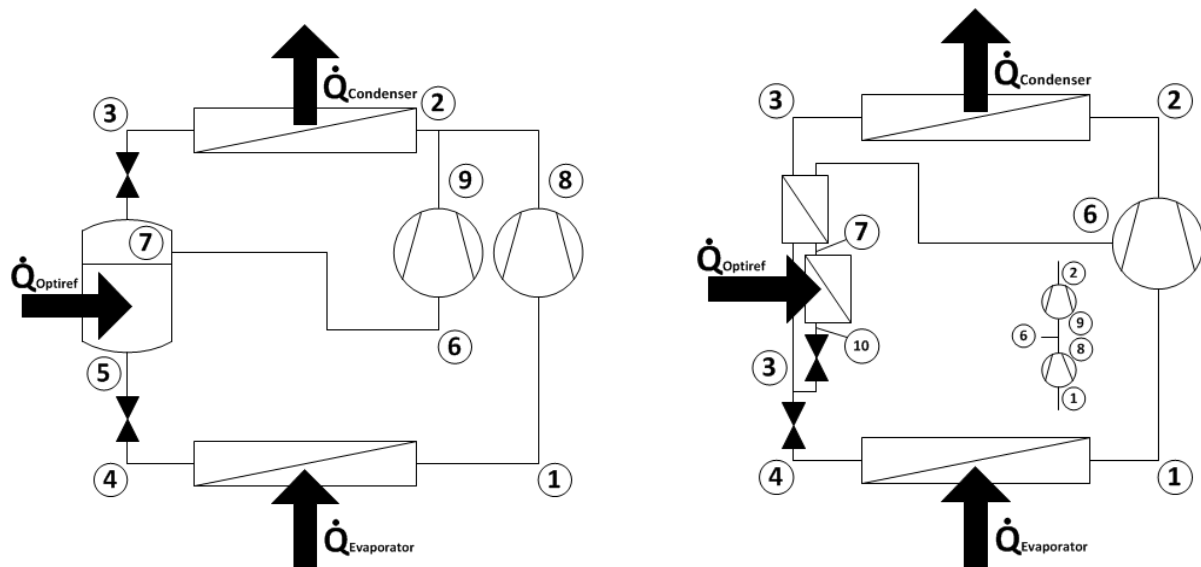
Control of the system is relatively simple with a standard superheat control of the evaporator. In addition, the expansion valve between condenser and OptiRef can be controlled either by subcooling or liquid level in the OptiRef.



**Figure 1:** Schematic and ph-diagram of basic refrigeration cycle with OptiRef (according to Granwehr, 2013)

In order to simplify the system and to decrease cost, a compressor with intermediate refrigerant injection such as the vapor injected scroll from Copeland can be used or for more flexibility two separate compressors (fig. 1).

Keeping the same idea in mind of adding heat from a second heat source at intermediate pressure, there are several more possible refrigeration circuits possible. Two of the most promising versions are shown in figure 2.



**Figure 2:** Schematic of two additional promising solutions (according to Granwehr, 2013). Left side: Open economizer with two parallel acting compressors. Right side: Closed economizer version with heat input at intermediate pressure.

## 2. SYSTEM EVALUATION

In order to evaluate and compare the different possible heat pump circuits a system model has been developed for a total of eight different circuits. The simulation model for the heat pump cycle is based on the models for compressors, heat exchangers, and expansions valves described in Bertsch et al. (2006, 2008). The following assumptions for the simulations were made:

- Refrigerant: R134a
- Standard operating conditions:  $T_{\text{supply}} = 45^{\circ}\text{C}$ ,  $T_{\text{source,in}} = 0^{\circ}\text{C}$  ; B0W45 (in this paper operating conditions will be referenced as BxWy – brine x°C, Water supply y°C)
- Compressors are simulated with a fixed isentropic efficiency of 70% and a fixed volume flow rate for the low pressure compressor.
- Heat exchangers are modeled using efficiency models with fixed heat transfer (UA) value and fixed mass flow of the fluid at the heat sink and heat source. The UA Values and mass flow rates were set at standard operation conditions and then kept constant for all following calculations.
- Superheat of  $10^{\circ}\text{C}$  at the compressor inlet
- Subcooling of  $5^{\circ}\text{C}$  at the condenser outlet
- The intermediate pressure was set to an optimum for the system with no heat injection into the OptiRef by adjusting the size of the high pressure stage compressor. The size of both compressors was then kept constant for all simulations.

The OptiRef was simulated as an open vessel with liquid refrigerant accumulating at the bottom and gaseous refrigerant accumulating at the top. This leads to a complete phase separation of the refrigerant with only liquid passing towards the evaporator and only gas passing towards the injection at intermediate pressure. The heat input from the second heat source was used to evaporate more refrigerant in the OptiRef and was calculated with an energy balance. Therefore, the injection mass flow rate at intermediate pressure increases with added heat to the OptiRef, while reducing the refrigerant mass flow through the evaporator accordingly. There was no heat exchanger model implemented inside the OptiRef for the heat addition from the second heat source. But the necessary temperature of the additional heat source can be derived from the intermediate pressure and the according temperature of the two-phase refrigerant. In practice the temperature of the additional heat source should be approximately  $5^{\circ}\text{C}$  higher than this intermediate temperature.

The simulation model was implemented using EES (Klein, 2013). While the assumptions given above lead to a fairly simplified model of the heat pump systems, a comparison between the different system variations seems valid.

For easy comparison of the results the amount of heat injected from the second heat source was related to the heating capacity of the heat pump using the factor  $\beta$ . Furthermore, the size of the high stage (high pressure) compressor was related to the low stage (low pressure) compressor, resulting in the factor  $\gamma$ .

$$\beta = \frac{\dot{Q}_{OptiRef}}{\dot{Q}_{Condenser}} \quad (1)$$

$$\gamma = \frac{\dot{V}_{HP}}{\dot{V}_{LP}} \quad (2)$$

Using this normalized heat injection  $\beta$  the effects of heat injection from the second heat source to the OptiRef were compared to the baseline system with no heat injection at standard operating conditions (source 0°C, sink 45 °C). Table 1 shows the results of this comparison for four different cycles.

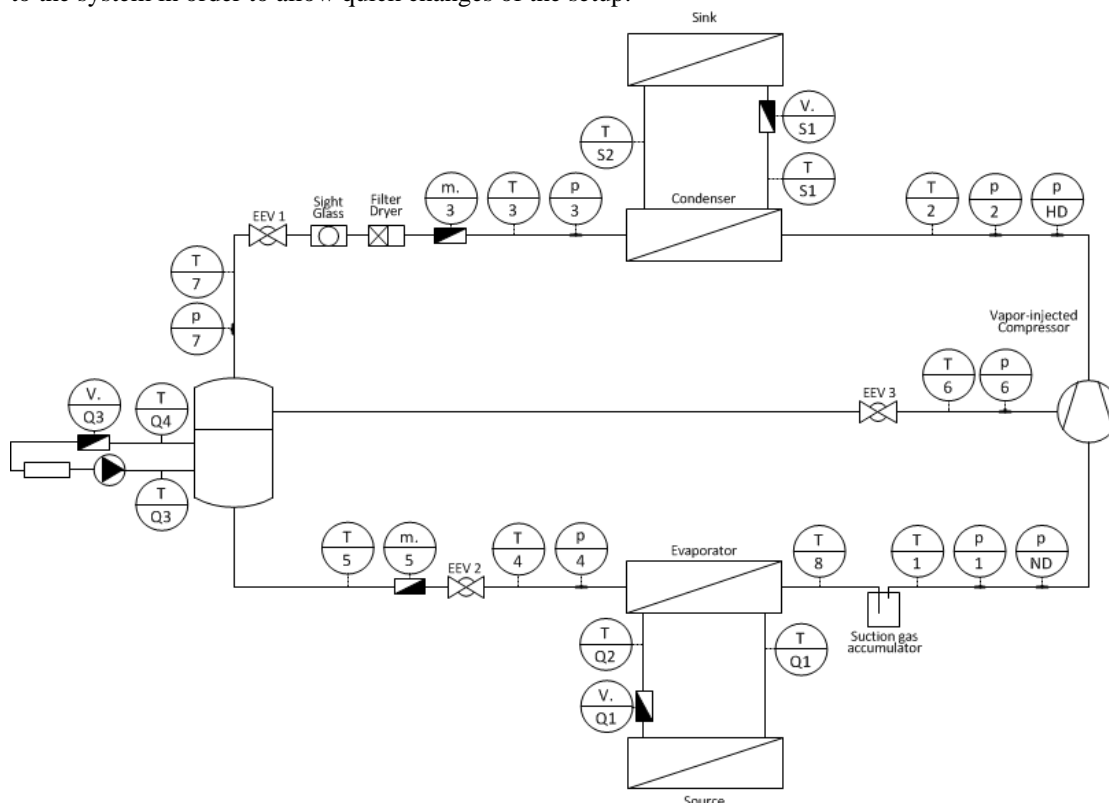
**Table 1:** Comparison of different systems with second heat source

System	$\gamma$	COP [-] (source 0°C, sink 45°C)				Comments
		$\beta=0$	$\beta=0.1$	$\beta=0.2$	$\beta=0.3$	
Open Economizer with 2 serial compressors (fig. 1)	0.4	3.4	3.7	4.1	4.7	Oil management problems solvable using compressor with intermediate injection Difficult to reverse cycle High refrigerant content High temperature lift of heat pump system possible
Open Economizer with 2 parallel compressors (fig. 2)	0.3	3.4	3.7	4.2	4.7	Oil management problems solvable using costly oil management system Difficult to reverse cycle High refrigerant content High temperature lift of heat pump system very difficult Highest cost (2 compressors and oil management)
Closed Economizer with 2 serial compressors (fig. 2)	0.4	3.5	3.8	4.2	4.7	Oil management problems solvable using compressor with intermediate injection Difficult to reverse cycle Low refrigerant content High temperature lift of heat pump system possible
Cascade system with heat injection at intermediate temperature level	0.7	3.4	3.7	4.0	4.4	No oil management problems Cycle reversal difficult but possible High temperature lift of heat pump possible, but no low temperature lifts High investment cost

The theoretical efficiency of the four compared systems is very similar, with the cascade system falling slightly behind. Since it also features high cost of the system, it was not investigated in further detail. Based on aspects like cost and oil management the second version with the parallel compressors was also ruled out. The systems with open and closed economizer and serial compressors were chosen for further investigation. In order to simplify the oil management a scroll compressor with vapor injection and variable speed drive was used as the compressor for the system, replacing the two separate compressors in series. In this way high efficiency of the circuit and reasonable investment costs can be combined. Even though the system shows most advantages when applied to an air source heat pump with its high temperature lifts, the prototype system presented here was a brine to water (geothermal) heat pump. Major reason for this choice is the ease of testing the system, while still being able to proof the concept.

### 3. PROTOTYPE AND MEASUREMENT SYSTEM

Figure 3 shows the laboratory setup of the breadboard system with open Economizer that was built to test the system concept. Due to the availability of components, the refrigerant R410a was chosen. On the bottom of the schematic we can see the evaporator, a plate heat exchanger which was heated by a brine circuit with electrical resistance heating. The refrigerant then passes through suction line accumulator and compressor with vapor injection to the condenser that was implemented as a plate heat exchanger cooled by water. After passing the refrigerant mass flow meter and the first expansion valve EEV1 the refrigerant then enters the OptiRef. As we can see on the left side of the schematic, heat is injected into the OptiRef from the second heat source again with an electrically heated water loop. Inside the OptiRef the refrigerant is then split into a liquid part that passes through EEV2 to the evaporator again. The gaseous refrigerant leaves the OptiRef on top and is then injected into the injection port of the compressor. EEV 3 is used to regulate the injection flow for different test conditions, but should not be necessary for implementations in the field, since it lowers the efficiency and adds to the investment cost. Several shut off valves (not shown) and a 4-way reversing valve (not shown) were added to the system in order to allow quick changes of the setup.



**Figure 3:** Schematic of the test setup with open economizer including sensor locations

For the measurements a series of temperature, pressure, flow rate and power sensors were installed at several locations of the system as shown in the measurement plan (fig. 3). Table 2 presents a list of the sensors and their measurement uncertainties. As data acquisition unit a NI C-Rio platform was used sampling the data every 5 seconds and storing them to a text file.

**Table 2:** Sensors and their according uncertainties

Name	Description	Measurement range	Uncertainty <sup>*)</sup>
T	Temperature measurement, T-type thermocouple	-20C to +120C	± 0.5°C
$\dot{V}$	Volume flow rate, magnetic inductive sensor	0.1 – 3 m <sup>3</sup> /h	± 3 % m.v.
$\dot{W}_{el}$	Electrical power, - 1 Phase, true RMS - 3 Phase, true RMS for compressor power	220 V / 10 A 380V / 10 A	± 0.2 % f.s. ± 0.2 % f.s.
p	Pressure sensor, piezoelectric sensor	0 to 10 bar (low p) 0 to 50 bar (int. and high p)	± 1% f.s. ± 1% f.s.
$\dot{m}$	Refrigerant mass flow meter, Coriolis sensor	0 – 150 g/s	± 0.2% m.v.

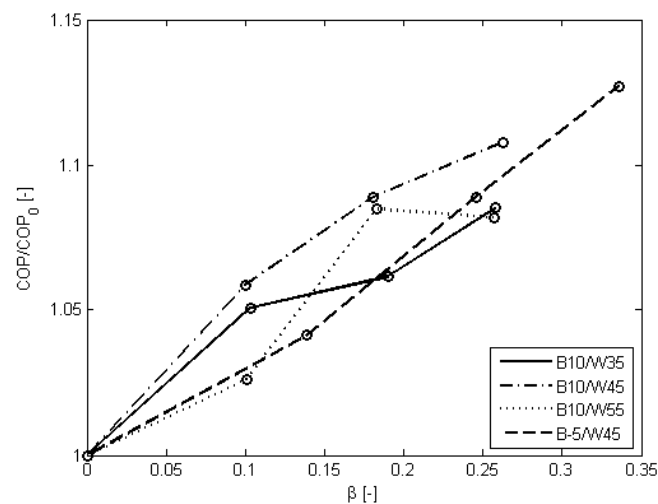
<sup>\*)</sup> m.v. (of measured value); f.s. (of full scale)

## 4. MEASUREMENT RESULTS AND DISCUSSION

### 4.1 Compressor with vapor injection

After setting up the system and doing some optimizations on the control system, the performance of the heat pump was measured for several different operating conditions. Temperatures of the main heat source were adjusted from  $-5$  to  $10^{\circ}\text{C}$  and for the heat supply from  $35$ - $55^{\circ}\text{C}$ . The temperatures of the second heat source were set according to the needs of the system and ranged from  $20$ - $30^{\circ}\text{C}$ . For each test, the capacity of the compressor was adjusted to  $10$  kW heating capacity, when no heat is added from the second heat source. The rotational speed was then kept constant and the heat input from the second heat source was increased from  $0$  kW in three steps up to  $3$  kW. The coefficient of performance (COP) was then calculated considering compressor and controller power consumption but not the power consumption of any pumps. Figure 4 shows the according results. The COP at different operating conditions is shown normalized to the value of the COP with no heat from the second heat source ( $\text{COP}/\text{COP}_0$ ) with respect to the normalized heat input from the second heat source  $\beta$ . With increasing heat addition from the second heat source we can see an improvement of the COP. At a heat input from the second heat source of approximately  $25\%$  we can see performance improvements of  $8$ - $12\%$  depending on operating conditions. According to the simulations we expected an increase of  $20$ - $25\%$  for the given operation. Additionally, the temperature of the second heat source was up to  $30^{\circ}\text{C}$ , instead of the expected  $20^{\circ}\text{C}$ , which reduces usability of the system in the field considerably, since the aim is to keep the temperature of the second heat source low. A quick alteration of the system to the closed economizer version according to figure 2 (right hand side) showed very similar results. The effect of the heat injection from the second heat source showed performance improvements ranging from  $7$ - $10\%$ , but showed some control instabilities. This led to the conclusion, that the open economizer is a slightly better solution than the closed version. Still, the far lower than expected efficiencies could not be resolved using the vapor injected scroll.

After close analysis of the results in collaboration with the compressor manufacturer the reason for the unsatisfactory results could be determined. Since the vapor injection of the used Scroll compressor is not designed for the higher flow rates of injected refrigerant, the compressor pockets of the scroll are too small to absorb the injected refrigerant for the given application. Therefore, the system experienced a high pressure build up at the injection port. This high pressure drop then led to the need of high temperatures of the second heat source. Due to the construction of the scroll, no solution to this problem could be found, which led to an improved system design using two compressors in series.



**Figure 4:** Measured COP values from the first prototype with vapor injected scroll compressor. Heating capacity is  $10$  kW in each operating point when there is no heat from the second heat source ( $\text{COP}_0$ ).

### 4.2 Two compressor solution

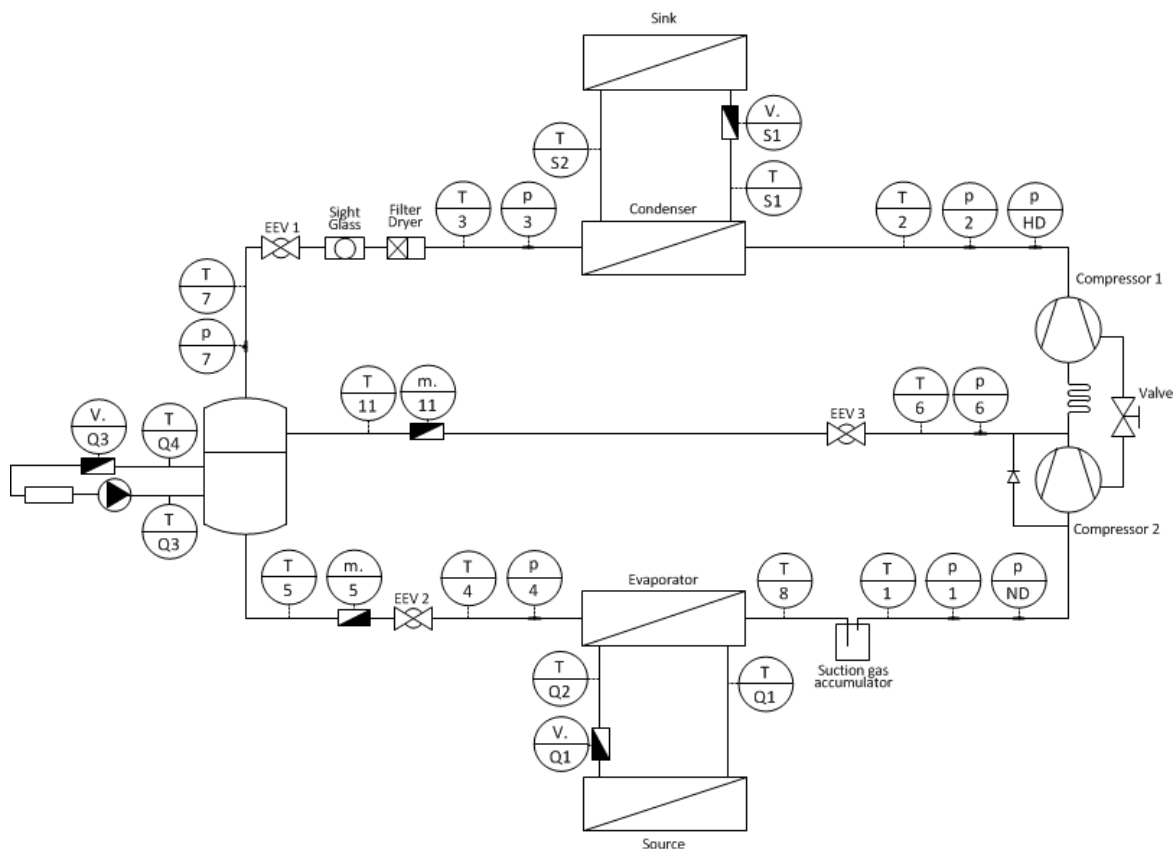
Since the scroll compressor with vapor injection is not suitable for the given application, a bread board system with two independent scroll compressors was built as shown in figure 5. In order to allow mixing of the injected refrigerant with the discharge gas of the low pressure stage compressor, an approximately  $2$  m long refrigerant tube was used between mixing point and inlet of the high pressure stage compressor. In addition an oil management system was used that connects the oil sump of both compressors. This connection line is controlled using a solenoid valve that opens one minute after shutdown of the compressors and closes before the compressors are started again. In this way oil equalization of the compressors is conducted during each shutoff

period of the system. No adverse effects resulting from oil shifting from one compressor to the other could be found for runtimes of up to 4 hours.

As with the first prototype, the system was run in the beginning without heat addition from the second heat source. In this way the functionality could be proven and the optimum refrigerant charge could be determined. Afterwards the second heat source was added again and the effects on heating capacity and COP investigated. Figure 6 shows the increase of the heating capacity of the heat pump, normalized to the heating capacity without heat injection from the second heat source  $\beta$ . The results are presented as a function of the normalized heat input from the second heat source for three different operating conditions and compared to the simulations results:

- B-10/W55 (brine source in  $-10^{\circ}\text{C}$ , heat supply  $55^{\circ}\text{C}$ ) with an intermediate pressure of  $p_{\text{int}} = 15,6$  bar and a temperature of  $24^{\circ}\text{C}$  for the second heat source
- B-10/W55 with an intermediate pressure of  $p_{\text{int}} = 13.2$  bar and a temperature of  $21^{\circ}\text{C}$  for the second heat source
- B0/W55 with an intermediate pressure of  $p_{\text{int}} = 14.7$  bar and a temperature of  $21^{\circ}\text{C}$  for the second heat source
- Simulation results from the design phase at B-10/W55 with an intermediate pressure of  $p_{\text{int}} = 13.2$  bar

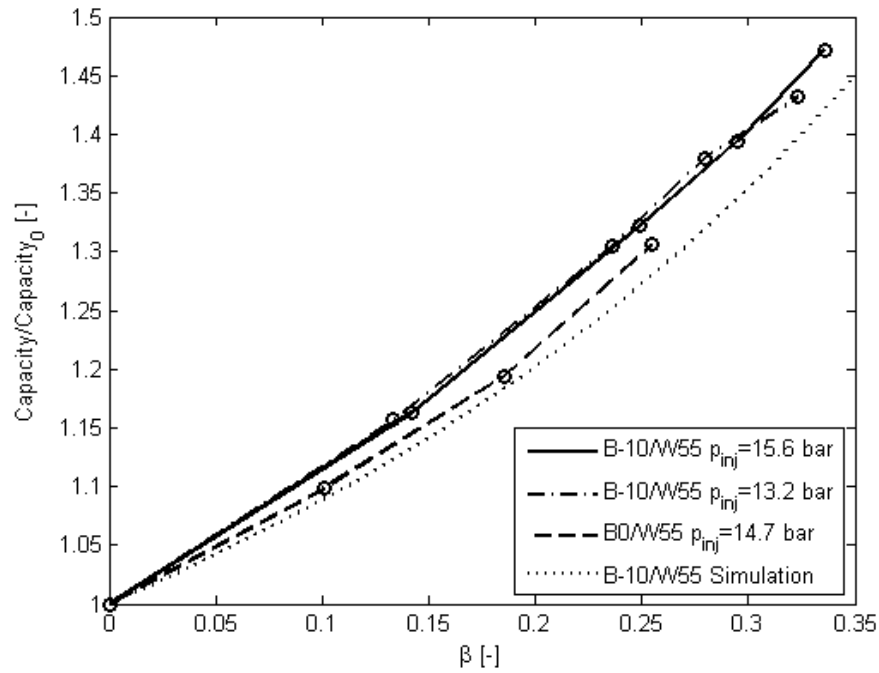
Figure 6 shows that the addition of heat of the second heat source leads to a significant increase in heating capacity. If 30% of the heating capacity is added at the second heat source the heating capacity of the system increases by approximately 35% for all cases. These measurement results match the simulation very well.



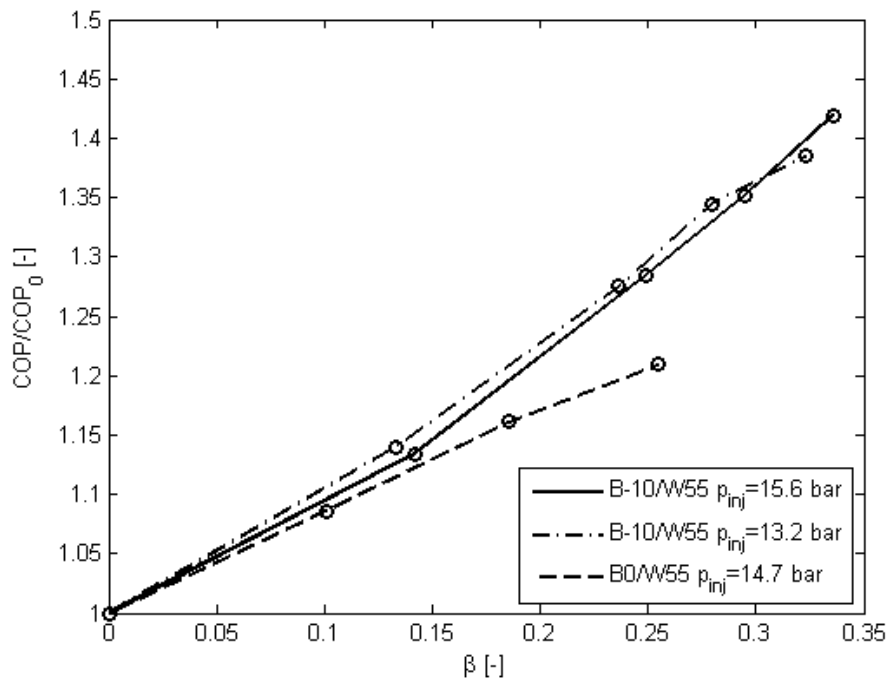
**Figure 5:** Schematic of bread board system with two independent compressors.

A similar comparison was carried out also for the coefficient of performance COP with varying degrees of heat addition from the second heat source. Figure 7 shows the according results. As can be expected the increase in COP is larger for the operating conditions with high temperature lift than for the lower temperature lift. Again the increase of performance is approximately 35% with the second heat source delivering 30% of the heating capacity. As we can see no significant difference in performance is achieved when using varying intermediate pressure levels.





**Figure 6:** Measured heating capacity from the prototype with two scroll compressors. Heating capacity for each operating condition is referenced to the state with no additional heat input from the second heat source.



**Figure 7:** Measured COP values from the prototype with two scroll compressors. COP for each operating condition is referenced to the state with no additional heat input from the second heat source.

These results confirm the theoretical analysis and the system simulations and show that it is possible to build the system with standard components from the market (with exception of the OptiRef). The downside of the results of this study is that the costs of the system increases significantly with the two-compressor solution and the according oil management. In addition, the operating envelopes of the compressors limit the operating conditions of the heat pump significantly. Larger systems using different compressor technology might show better cost efficiency. Besides heat pumping systems also applications in the field of refrigeration are possible.

## 5. CONCLUSIONS

A heat pump system with two heat sources at different temperature levels has been simulated and a laboratory prototype has been built and tested. In a first step a system with open Economizer was built using a variable speed scroll compressor with vapor injection port. While the function of the system could be proven, the expected COP and heating capacity increase could not be proven. At a heat source temperature of 0°C and a heat sink temperature of 45°C, the temperature of the second heat source needed to be at approximately 30°C in order to allow an 8-12% increase in COP. This means that the intended exergy reduction could not be fulfilled and the applicability for field installations is reduced.

It was found in close discussion with the compressor manufacturer, that the major problem of the system was the fact that the compressor is unable to swallow the large amount of injected refrigerant at intermediate pressure due to its internal design. A second system design with the same compressor and a closed economizer system that allows two-phase injection into the compressor showed very similar results but with slightly lower performance improvements.

Therefore, a system was built using two independent compressors with according oil management. This system then showed the desired results with respect to COP and heating performance. At standard operating conditions (source 0°C and sink 45°C) efficiency and heating capacity improvements of up to 35% could be demonstrated when the second heat source was at approximately 20°C. Therefore the system is technically feasible, but the costs are significant due to the need of two compressors and an according oil management system. From this perspective the target market should be large scale heat pump or refrigeration applications with two heat sources.

## 6. NOMENCLATURE

Symbol		Greek	
COP	coefficient of performance [-]	$\beta$	ratio of heat from 2 <sup>nd</sup> heat source to supplied heat
$\dot{m}$	mass flow rate [kg/s]	$\gamma$	volume ratio of high press. to low press. compressor
$p$	pressure [kPa]		
$\dot{Q}$	heat rate [W]	Indices	
SPF	seasonal performance factor [-]	el	electrical
$T$	temperature [°C, K]	in	in
$\dot{V}$	volume flow rate [m <sup>3</sup> /h]	inj	injection
$\dot{W}$	power [W]	sink	heat sink
		supply	heat supply

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