Energy Saving Potential of a Temperature Test Chamber by implementing a Heat-Pump

Andreas Wagner
Bitzer-Professur, Germany, andreas_wagner@tu-dresden.de

Thomas Tannert
Bitzer-Stiftungsprofessur für Kälte-, Kryo- und Kompressorentechnik / Technische Universität Dresden, Germany, thomas.tannert@tu-dresden.de

Ullrich Hesse
ullrich.hesse@tu-dresden.de

ENERGY SAVING POTENTIAL OF AN ENVIRONMENTAL TEST CHAMBER BY IMPLEMENTING A HEAT-PUMP

Andreas WAGNER[1]*, Thomas TANNERT[2], Ullrich HESSE[3]

Technische Universität Dresden, Bitzer Chair of Refrigeration, Cryogenics and Compressor Technology
Dresden, Saxony, Germany

[1] phone: +49 351 463 32603, fax: +49 351 463 37247, andreas_wagner@tu-dresden.de

[2] phone: +49 351 463 32546, fax: +49 351 463 37247, thomas.tannert@tu-dresden.de

[3] phone: +49 351 463 32548, fax: +49 351 463 37247, ullrich.hesse@tu-dresden.de

* Corresponding Author

ABSTRACT

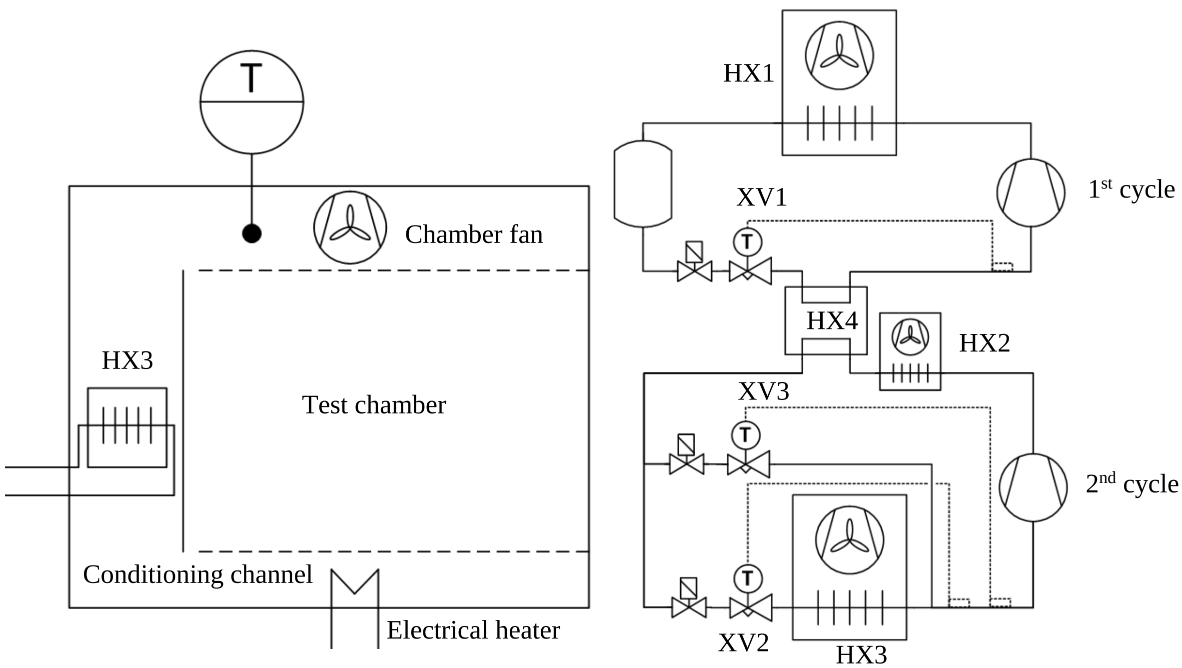
Energy saving becomes more and more important – also for the rather small industry sector of environmental simulation. So far the technical focus was mainly to realize challenging test procedures which are often gives as temperature profiles. The energy efficiency of the test devices are not specified by any standard. Nevertheless, especially test procedures with intermittent cooling and heating over a wide temperature range need a substantial electrical energy input. As state-of-the-art technology for cooling a vapor compression cycle and for heating an electrical heater is used.
In this work a standard environmental test chamber with a specified temperature range from -75 °C to 180 °C using a cascade cooling system and an electrical heater is investigated. As a first step the baseline chamber was investigated experimentally and the energy consumption was quantified. In the next step the refrigeration cycle was modified with a heat-pump capability as well as further cycle modification that indicated energy saving potential in preliminary studies. A comparison to the standard environmental test chamber is carried out and reveals substantial energy saving potential.

1. INTRODUCTION

Environmental test chambers are used for individual stress tests of technical parts, assemblies or substances. These tests require predefined environmental conditions which may include temperature, humidity, sunshine or other desired conditions. Test procedures such as temperature-time profiles are usually defined by published and proprietary standards. Hence, environmental test chambers need to be designed for desired temperature ranges and necessary temperature changing speed to cover the required test procedures. Requirements on performance of environmental test chambers are defined in the IEC 60068-3-5 (International Electrotechnical Testing, 2001). State-of-the-art environmental test chambers use slightly modified vapor compression cycles for cooling and an electrical heater for heating.
Test procedures with many temperature changes cause high electrical energy consumption because of many cooling and heating operations. A new approach is to integrate a heat-pump system into the cooling cycle of the environmental test chamber. A heat-pump could heat up the test chamber more efficient than the electrical heater. Depending on the specified temperature range and the refrigerant it's assumed that the heat-pump can only replace the electrical heater in a subrange.
The heat-pump has advantages in efficiency during heating from temperatures below the ambient. In theory for this range the heat-pump doesn't have to generate any temperature lift. This paper presents a concept chamber with an integrated heat-pump system. In preliminary measurements the energy consumption is evaluated and compared to a baseline environmental test chamber.

2. STATE-OF-THE-ART OF ENVIRONMENTAL TEST CHAMBERS

The state-of-the-art design is explained for a common environmental test chamber, which builds the baseline for the experiments[1]. Figure 1 a) shows a simplified schematic diagram of the environmental test chamber. The relevant test chamber has a fixed volume of 280 l and is conditioned by a continuous, closed air cycle that is driven by a radial fan at the top. The air circulates through the test chamber and the adjacent conditioning channel. The conditioning channel consists of a finned tube evaporator and an electrical heater. The structure of the cooling cycle depends on the minimal applicable temperature. For a test chamber temperature down to -40 °C a single stage vapor compression cycle is used. A cascade cooling system can provide a chamber temperature down to -75 °C. The simplified[2] cascade system in Figure 1 b) shows two cooling cycles, which are thermally coupled via a cascade cooler (HX4). The first cycle is driven with 2.5 kg R452A and contains a receiver. R452A, as well as R449A, is a low-GWP alternative to replace R404A and R507 (Wagner *et al.*, 2017). The second cycle works with 0.5 kg R23 and uses an additional discharge gas cooler (HX2). The cycle doesn't have a receiver because of the high pressure level of the refrigerant. During standstills the refrigerant is superheated. Further a second expansion valve (XV3) applied to cool the suction gas during cooling from test chamber temperatures above 60°C.
The baseline environmental test chamber has a test chamber volume of 280 l and is specified with a temperature range from -75 °C to 180 °C and an average temperature changing rate[3] of 3 K/min. Both compressors have a nominal flow rate of 14.5 m^3/h. The expansion valve XV2 has a MOP of 3.1 bar (T'_{R23}(3.1 bar) = -60 °C, Lemmon et al., 2013). On a medium evaporation temperature of -70 °C the cascade system has a cooling capacity of around 2.88 kW[4]. The electrical heater has a capacity of 4 kW.



a) Environmental test chamber   b) Cascade cooling cycle

Figure 1: Schematic diagram of the state-of-the-art environmental test chamber and cooling cycle

[1] Type 3636/17, Feutron Klimasimulation GmbH
[2] For an easier understanding components for part load are not shown.
[3] IEC 60068-3-5 (International Electrotechnical Testing, 2001)
[4] Calculated with no subcooling, T_c = -20 °C, λ = 0.7, η_{is} = 0.7, T_{SH} = 30 K, T_{SG} = -10 °C

3. STRUCTURE OF THE CONCEPT CHAMBER

3.1 Thermodynamic Design

The concept chamber is based on the baseline cascade system. Figure 2 shows the design points in a T-s-diagram. The state points of the first and second cycle in cascade mode at the widest temperature spread show discharge temperatures below 90 °C. A heat-pump using only the first cycle is also shown. A condensation temperature of 60 °C can be generated with a discharge temperature of 82 °C and maximum pressure of 30 bar.
At this design point evaporation takes place on a temperature of 5 °C. For R452A this results in a pressure of 6.3 bar. A heat-pump using the whole cascade system brings numerous disadvantages compared to the single cycle heat-pump:

- $p_{0,2}$ of the second cycle: 28 bar, critical point at 25.9 °C and 46.99 bar
- Heat transfer in the cascade cooler needs additional temperature lift and compression work
- A missing receiver can cause a lack of refrigerant
- Modification in two cooling cycles

The thermopyhsical properties show that R23 doesn't have an applicable pressure level for a heat-pump. Using a cascade heat-pump also has a lower efficiency compared to a single cycle system. These facts bring the decision to develop a single cycle heat-pump system by extending the first cycle of the cascade.
The heat-pump uses the environment as heat source and the test chamber as heat sink. To use the test chamber as heat sink an additional heat exchanger has to be added. The additional heat exchanger also allows a single cycle cooling mode. According to the refrigerant's minimal boiling temperature[5] on ambient pressure the first cycle could provide a minimal test chamber temperature of -40 °C. A cooling process with only the first cycle should be more efficient than cooling with the cascade system. The state points of single cycle cooling are also represented in the T-s-diagram in Figure 2. It shows that even with an evaporation temperature T_0'' of -42.2°C and a condensation temperature T_c'' of 35 °C a moderate discharge temperature T_{DC} of 75 °C can be realized[6].

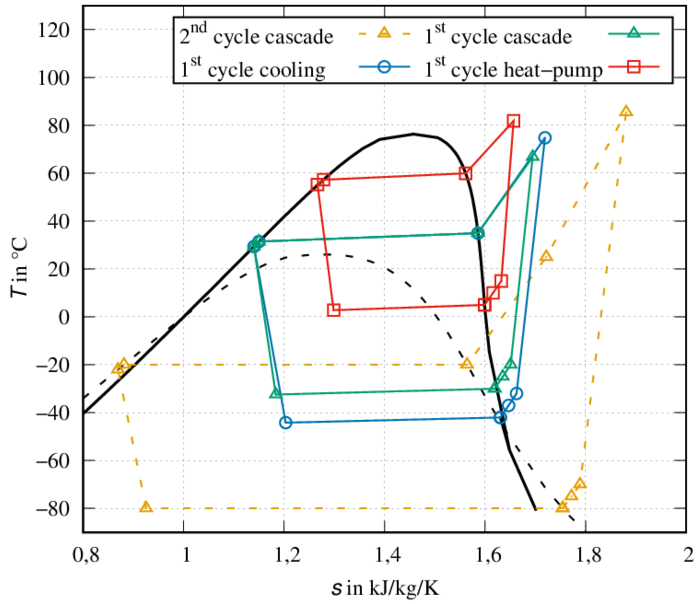


Figure 2: T-s-diagram of the modes of the concept chamber cycles

[5] T''_{R452A}(1 bar)= -42.2 °C, Refprop 9.1 (Lemmon et al., 2013)
[6] Calculated with, η_{is} = 0.7, T_{SG} = -32 °C.

3.1 Structure and Component Design

The heat-pump is designed to have the same heat-load of 4 kW like the electrical heater. Therefore a refrigerant volume flow of 5.1 m^3/h is necessary[7]. Compared to the first cycle of the baseline environmental test chamber a smaller compressor is needed. In the concept chamber both compressors come with variable speed. They can provide the same volume flow like the baseline. The first cycle's compressor is a Bitzer 2-FES-2Y and supplies in cascade mode a volume flow 14.5 m^3/h at 70 Hz. The compressor of the second cycle is a Bitzer 2-DES-3Y and provides a volume flow of 14.5 m^3/h at almost 50 Hz (Bitzer Kühlmaschinenbau GmbH, 2012). Figure 3 shows a simplified[8] schematic diagram of the concept chamber. The evaporator of the second cycle (HX3) and the cascade cooler (HX4) are identical with those of the baseline climate test chamber. The structure of the additional heat exchanger HX5 is similar to HX3. HX5 is enlarged around 40 % by adding tube rows. It is also placed in the conditioning channel. The enlargement is done to realize higher heat loads during heat-pump mode compared with the cooling load the cascade system. The inlet flow of HX5 can go through distributor capillary tubes during cooling mode and through a manifold for heat-pump mode. Both modes provide a countercurrent flow. The condenser HX1 is a new dual flow construction. For condensation it has an inlet on top and an outlet on the bottom. The outlet on the bottom can be bypassed with distributor capillary tubes to work as an inlet during heat-pump mode. In that mode HX1 works as evaporator and has its outlet on the top. As a crossflow heat exchanger the change of the refrigerant flow direction doesn't have any effect on heat transfer rate, which will occur between cooling and heat-pump mode. HX1 is designed to have a heat load of 7.5 kW as condenser[9]. Table 1 shows the valve settings for different modes of the concept chamber. Solenoid valves and a check valve placed in suction lines can cause significant pressure losses, which degrease cooling and heating capacity and efficiency. There are three[10] solenoid valves in the suction line of the type Alco 540 RA 12. They come with a kv-value of 5.4 m^3/h and a minimal opening pressure drop of 0.05 bar (Emerson Electric Co., 2018). The first cycle contains 4.8 kg of R452A. HX3, HX4 and HX2 are equal to the baseline.

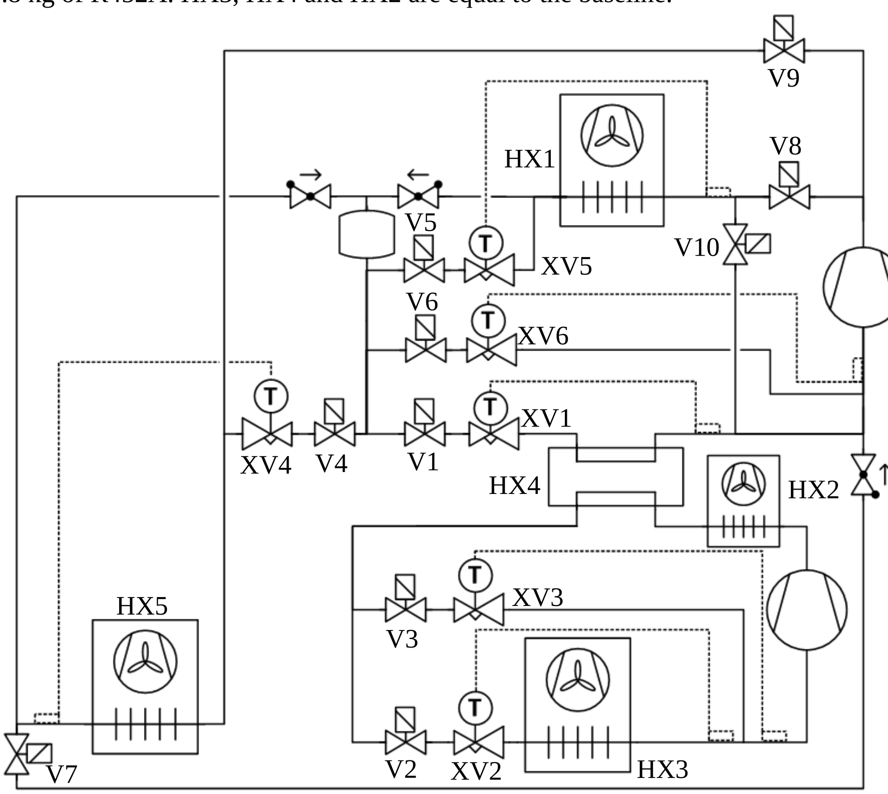


Figure 3: Simplified flow diagram of the concept chamber with additional heat-pump and single cycle cooling mode

---

[7] Calculated as shown in Figure 2 with $T_0 = 5$ °C, $T_C = 60$ °C, $T_{sub} = 0$ K, = 0.7, $\eta_{is} = 0.7$, $\Delta T_{SH} = 5$ K, $T_{SG} = 20$ °C

[8] Some elements for part load mode and air drying are not shown.

[9] $\dot{m}_{R404A} = 0.045$ kg/s, $T'_{R404A,out} = 33$ °C, $T_{R404A,in} = 65$ °C, $T_{air,in} = 25$ °C, $p_{air,\ in} = 1$ bar, $\dot{V}_{air,in} = 0.914$ m^3/s, full condensation

[10] Valve V7 and two more valves for part load mode and air drying feature are placed in the suction line.

Table 1: Switching table of the solenoid valves for the concept chamber

Mode	V1	V2	V3	V4	V5	V6	V7	V8	V9	V10
Cascade cooling	X	X						X		
Single cycle cooling				X	X		X	X		
Heat-pump					X				X	X

3. MEASUREMENTS

Both the baseline environmental test chamber and the concept chamber are tested in a cooling test procedure to check the comparability. In this test the concept chamber runs without using any new feature. Further the single cycle cooling mode is tested and integrated into the complete cooling procedure. A heating test procedure is driven to evaluate the energy consumption of the heat-pump mode. The results are compared to measurement data of a heating test procedure with the electrical heater.

4.1 Measurement Equipment

To measure the test chamber temperature the sensor (PT1000) of the environmental test chamber is read via RS232 interface. This brings the same conditions like in standard environmental test chambers. Further additional sensors connected to a measurement system are used. The additional temperature sensors for testing the baseline environmental test chamber are thermocouple type K, which have an accuracy of ±0.5 K after calibration. For testing the concept chamber thermocouples type T are used, which already bring an accuracy of ±0.5 K. For both measurements pressure sensors are used, which have an accuracy of ±0.050 bar after calibration. The energy consumption is measured with a Yokogawa WT130 digital power meter, which has an accuracy of ±(0.5 % of rdg + 18 W).

4.2 Comparison of the Baseline with the Concept Chamber at a cool down Test Procedure

To compare the baseline and the concept chamber a cooling test procedure from the highest to the lowest specified temperature (180 °C…-75 °C) is tested. The average ambient temperature on the baseline test is 19.5 °C and 20.5 °C at the concept chamber test. Figure 4 shows the test chamber temperature and the electric energy depending on time. When the test chamber temperature falls below 60 °C an acceleration of the cooling process can be observed. This happens because the suction gas cooling by the second expansion valve is deactivated. Going on from that position the full mass flow enters the evaporator. All in all temperature the energy consumption of the baseline environmental test chamber and the concept chamber fit quite well. The concept chamber needs 4 % less time and 6 % less energy for cooling down. A reason for that might be the larger orifice assembly in the expansion valve of the second cycle (XV2) of the concept chamber.

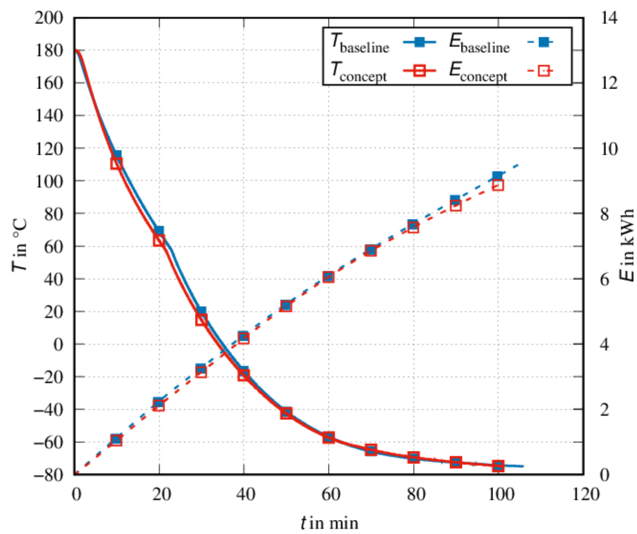


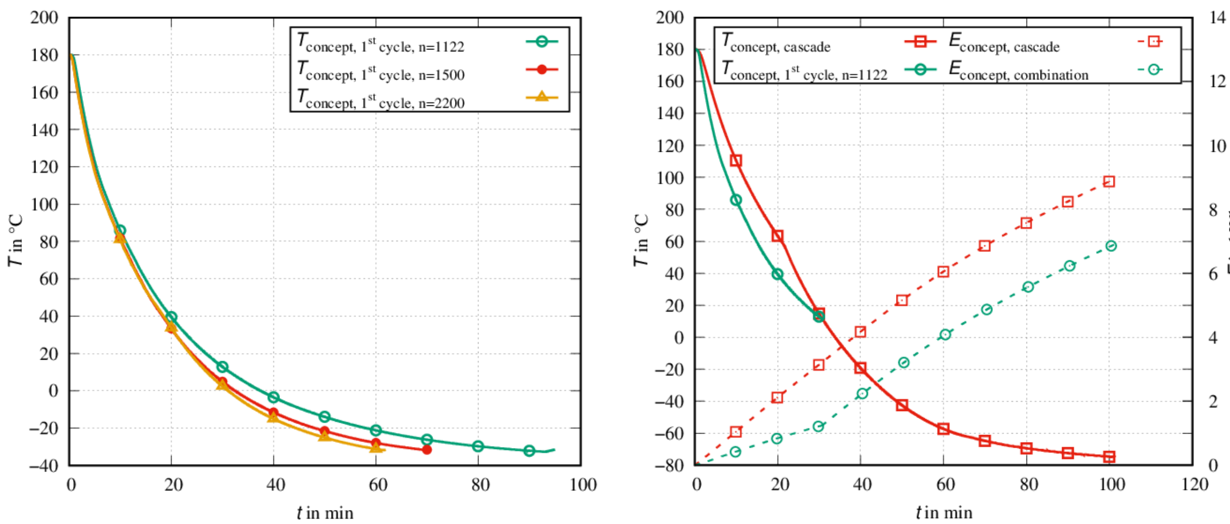
Figure 4: Cool down test procedure of the baseline environmental test chamber and the concept chamber

4.3 Evaluation of the single Cycle Cooling mode of the Concept Chamber at a cool down Test Procedure

To evaluate the single cycle cooling mode a cooling test procedure starting from 180 °C is driven. The cooling procedure is tested with different rotational speeds of the compressor. Figure 5 a) shows the test room temperature over time. As expected the temperature decreases faster with increasing speed of the compressor. In this configuration it is not possible to cool the test chamber below -31 °C. The reason for that are the solenoid valves in the suction line, which start closing partly when the pressure drop falls below 0.05 bar. Using valves with a smaller kv-value avoids uncontrolled closing because of higher pressure losses. Higher pressure losses reduce the cooling capacity, so larger flowrates are necessary to achieve lower test chamber temperatures.

Reducing the number of valves in the suction line can be a solution. This would ban some additional functions. Another solution is to find active working valves, which don't need a minimal pressure loss to open.

Even with the actual configuration energy can be saved during a cooling procedure. Figure 5 b) shows the combination of the single cycle cooling mode with a rotational speed of 40 Hz and the cascade cooling mode. The temperature profiles show, that single cycle cooling is faster than the cascade cooling in the test room temperature range from 180 °C down to 60 °C. The reason therefore are higher mass flow rate values resulting from higher suction gas pressures. To evaluate the energy savings in context of the complete cooling procedure the single cycle cooling mode is combined with the cascade cooling mode. In the example of Figure 5 b) the single cycle cooling mode is used until both temperature curves cross (test chamber temperature of 11 °C). Until that point 61 % of energy can be saved. During cooling from 180 °C to -75 °C the energy consumption can be reduced by 25 %.



a) Rotational speed variation of the compressor at single cycle cooling mode

b) Integration of single cycle cooling mode into the full scale cooling procedure

Figure 5: Concept chamber using single cycle cooling mode at a cool down test procedure

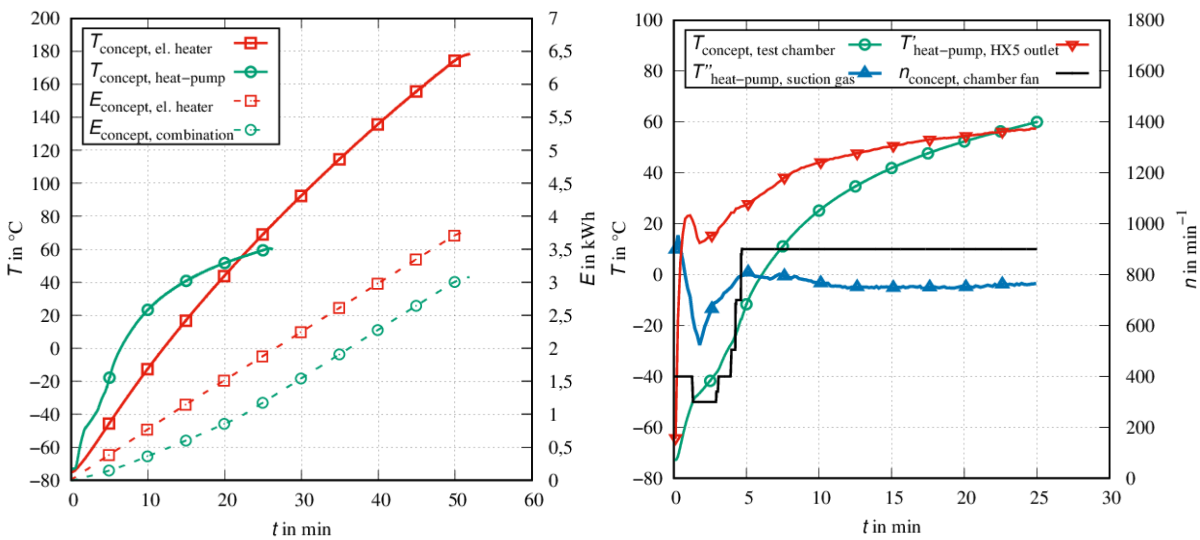
4.4 Evaluation of the Heat-Pump Mode of the Concept Chamber at a Heat up Test Procedure

For the evaluation of the heat-pump the concept chamber is cooled down to -75 °C and then switched from cascade mode to heat-pump mode in accordance with Table 1. During heat-pump mode the compressor runs with a rotational speed of 37 Hz and an average ambient temperature of 20.3 °C. The rotational speed of the chamber fan is reduced from 900 1/min to 400 1/min to minimize the pressure drop in HX5. For comparison a heat up test procedure with electrical heater was measured at an average ambient temperature of 19.9 °C. Figure 5 a) shows the test chamber temperature and energy consumption over time. At the beginning of the heat up test procedure with heat-pump the temperature profile of the test chamber shows an abrupt increase. This results from relocation processes of the refrigerant during switching. After the abrupt temperature increase the heating process slows down, because the condensing pressure degreases. The limitation of the maximum opening degree of the expansion valve in combination with the low condensing pressure causes a low evaporation pressure. The low evaporation pressure results in a low mass flow.

Figure 5 b) shows the temperature of saturated liquid at the outlet of HX5, the temperature of saturated vapor in compressor inlet and the suction gas temperature. In answer to the degreased condensation pressure the chamber fan rotation speed is reduced. The effect can be seen in the ongoing heating process and the increasing heating rate up to a test chamber temperature of -10 °C (Figure 5 a)). From that point the heating rate is decreasing. There are several possible reasons for a reduction of heat load at heat-pump mode on increasing test chamber temperatures.

1. The specific condensation enthalpy degreases with an increasing pressure.
2. There is no complete condensation in HX5 because of an underload of refrigerant.
3. There is no complete condensation in HX5 because of condensation in the line to the receiver and in the receiver.

Point 1 is defined by the thermophysical properties of R452A and can only be changed by using a refrigerant with a more constant specific condensation enthalpy. During the measurement bubbles in the sight glass at the outlet of the receiver are observed, so an underload of refrigerant in accordance with point 2 is probable. At Figure 5 b) it can be seen, that from 22.5 min the refrigerant temperature at the outlet of HX5 is higher than the test chamber temperature. At least from this time a non-complete condensation happens. The necessary refrigerant mass depends on the intended maximum test chamber temperature in heat-pump mode and the corresponding evaporation temperature. If there is an underload of refrigerant mass ongoing at least from 22.5 min, much higher heat loads and efficiencies can be achieved by adding more refrigerant. Point 3 couldn't be verified. Therefore it might be an option to improve the insulation of the tubing between HX5 outlet and receiver inlet and to insulate the receiver.
In the actual state the heat-pump allows a reduction of energy consumption of 43 % during the heat up procedure from -75 °C to 55 °C. Over the full temperature scale it's still a reduction of 19 %.



a) Heat-pump mode and electrical heating

b) Temperatures during heat-pump mode and rotational speed of the test chamber fan

Figure 6: Heat-pump mode and electrical heating of the concept chamber at a heat up test procedure

CONCLUSION

In this work a heat-pump is integrated into an environmental test chamber. The new concept chamber is built up. It is based on a state-of-the art environmental test chamber with a vapor compression cascade cycle and an electrical heater and supports test chamber temperatures from -75 °C to 180 °C. An additional heat-pump function is integrated in the first cycle of the cascade system, which represents a new feature. This modification also allows running the cooling mode by using only the first cycle. Both additional modes shall bring significant energy saving potential in comparison to a baseline environmental test chamber.

The new concept chamber is compared with a baseline environmental test chamber within a cool down procedure. Without using the new features the concept chamber provides similar results to the baseline chamber. Integrating the single cycle cooling mode into the cool down procedure brings up an energy saving potential of 25 %. The temperature range and efficiency of the single stage cooling mode can be increased by reducing the number of solenoid valves or by using valves with a lower pressure loss and without a minimal opening pressure drop. The heat-pump mode is tested within a heating test procedure starting form a test chamber temperature of -75 °C. It is compared with a heating test procedure using only the electrical heater. The heat-pump substitutes the electrical heater for the heating process within a test chamber temperature range between -75 °C and 55°C. In that temperature range 43 % of electrical energy can be saved. For a full scale heat up procedure the electrical heater is still necessary. In combination with the electrical heater the heat-pump still brings up an energy saving potential of 19 % over the full temperature range.

During the heating process using the heat-pump, an uncomplete condensation can be verified. Reasons therefor can be found in a deficit of refrigerant mass or a partial condensation in the receiver. A larger amount of refrigerant mass and an additional thermal insulation of the receiver can cause a higher heat loads and higher efficiency.

All in all a heat-pump mode can increase the efficiency of temperature test chambers during test procedures with many temperature changes. Further investigation on optimizing the heat-pump and methods for part load control are recommended.

NOMENCLATURE

The nomenclature should be located at the end of the text using the following format:

HX	heat exchanger	(-)
MOP	maximum operation pressure	(bar)
rdg	reading	(-)
XV	expansion valve	(–)
E	energy	(kWh)
kv	flow factor	(m^3/h)
$\dot{m}$	mass flow	(kg/s)
n	rotational speed	(min^{-1})
p	pressure	(bar)
T	temperature	(°C)
T'	temperature of saturated gas	(°C)
T''	temperature of saturated liquid	(°C)
V	valve	(-)
$\dot{V}$	volume flow	(m^3/h)
λ	volumetric efficiency	(-)
η	efficiency	(-)

Subscript

0	evaporation
c	condensation
in	inlet
is	isentropic
SH	superheat
SG	suction gas#
Sub	subcooled
TC	test chamber

REFERENCES

Bitzer Kühlmaschinenbau GmbH, 2012, Documentation semi hermetic reciprocation compressors, Retrieved from https://www.bitzer.de/shared_media/documentation/kp-104-1.pdf

Emerson Electric Co., 2018, 200RB 240RA 540RA, Retrieved from https://climate.emerson.com/documents/200rb-240ra-540ra-en-3758098.pdf

International Electrotechnical Testing, 2001, IEC 60068-3-5 Environmental testing - Part 3-5: Supporting documentation and guidance - Confirmation of the performance of temperature chambers.

Lemmon, E.W., Huber, M.L., McLinden, M.O. 2013, REFPROP, NIST Standard Reference Database 23, Version 9.1, Boulder, USA.

Wagner, A., Langebach, R., Hesse, U., 2017, Systematische Suche und Erprobung von Ersatzkältemitteln für Anwendungen in der Klimasimulation bis -40 °C, DKV-Tagung, Bremen.

ACKNOWLEDGEMENT

The author appreciates the collaboration with the project partner Feutron Klimasimulation GmbH.