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DECARBONISATION THROUGH INNOVATION

Experimental investigation of the Recuperative Two-Phase heat pump Cycle

Benedikt G. Bederna [a]*, Christiane Thomas [a]

[a] TUD Dresden University of Technology,
Schaufler Chair of Refrigeration, Cryogenics and Compressor Technology, 01062 Dresden, Germany

Abstract

The Recuperative Two-Phase Cycle (RTPC) is an approach developed at TU Dresden in recent years to increase the efficiency of thermodynamic cycles. The main difference to the state of the art is the use of an asymmetric working fluid, which enables a liquid two-phase recuperation with minimal entropy production, as the capacity flows on both sides of the internal heat exchanger are equalized by a phase change on the low-pressure side. The efficiency improvement in the heat pump mode of the cycle is achieved by reducing the isenthalpic throttling losses by subcooling the working fluid to a temperature close to the evaporation temperature. The design of a test bench is presented, with which the heat pump cycle is realized for the first time. The set-up consists of five main components and uses the asymmetric mixture propane-dodecane. A conventional high-speed rotary compressor is used; however, it is operated without oil and the compression takes place in the two-phase region. A particular challenge is to determine the circulating and local composition of the asymmetric working fluid. A specific solvent-based method is described for taking small samples from the liquid and the two-phase flow, which can be analysed in a gas chromatograph. The experimental results of transient ramp-ups and various stationary operating points of the heat pump process are shown. A maximum temperature lift of 33 K between heat source and heat sink is achieved, and the pre-calculated behaviour of the cycle could be found. Observations on the operation of the process are discussed, the influence of the compressor speed and the source temperatures is highlighted, and the significance of the composition of the working fluid is analysed. Finally, observations on the compressor operability are discussed and necessary follow-up investigations for the quantification of the observed phenomenon are derived.

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Keywords: Recuperative Two-Phase Cycle (RTPC); Heat pump cycle; Proof of concept; Not-in-kind; Two-phase compression

1. Introduction

The Recuperative Two-Phase Cycle (RTPC) is an approach developed at TU Dresden in recent years to increase the efficiency of thermodynamic cycles [1]. In 2023, its application for heat pump applications was first described [2]. A detailed work including the theoretical background, a calculation approach and different study cases are currently under review [3]. Further theoretical classifications are planned in the near future by the authors. The basic idea of the RTPC heat pump cycle is to minimise throttling losses caused by isenthalpic expansion through maximum subcooling. Usually, only a small portion of this subcooling heat can be used effectively. That is why, in the intended design, subcooling is achieved in an internal heat exchanger (recuperator). On the low-pressure side of this recuperator, the temperature of the working fluid flow upstream of the compressor is raised from the low-temperature level to the high-temperature level of the cycle by cooling down the liquid. However, this is only possible if both working fluid mass flows have the same heat capacity flow. Consequently, there can be no superheated steam on the low-pressure side, as it would be the case in internal heat exchangers of conventional vapour compression cycles. The design therefore demands that an evaporation with a simultaneous increase in temperature must take place on the low-pressure side. Accordingly, the temperature difference of the subcooling corresponds to the temperature difference during

the partial evaporation. Above, this temperature difference also corresponds to the temperature lift of the cycle. This liquid two-phase recuperation is the key idea of the new cycle concept.

With a pure substance the behaviour cannot be achieved. Even conventional mixtures do not exhibit such a strong temperature gradient in the two-phase region during an isobaric change of state. Instead, asymmetric refrigerant mixtures are used, which have a non-linear temperature characteristic in the two-phase region and a very large temperature glide. These consist of at least one refrigerant and one auxiliary fluid, whereby the auxiliary fluid has a significantly higher normal boiling point and a much lower molar fraction. An example of this is the mixture of 95 mol % propane and 5 mol % n-dodecane, which is illustrated in Fig. 1 in a T-s diagram.

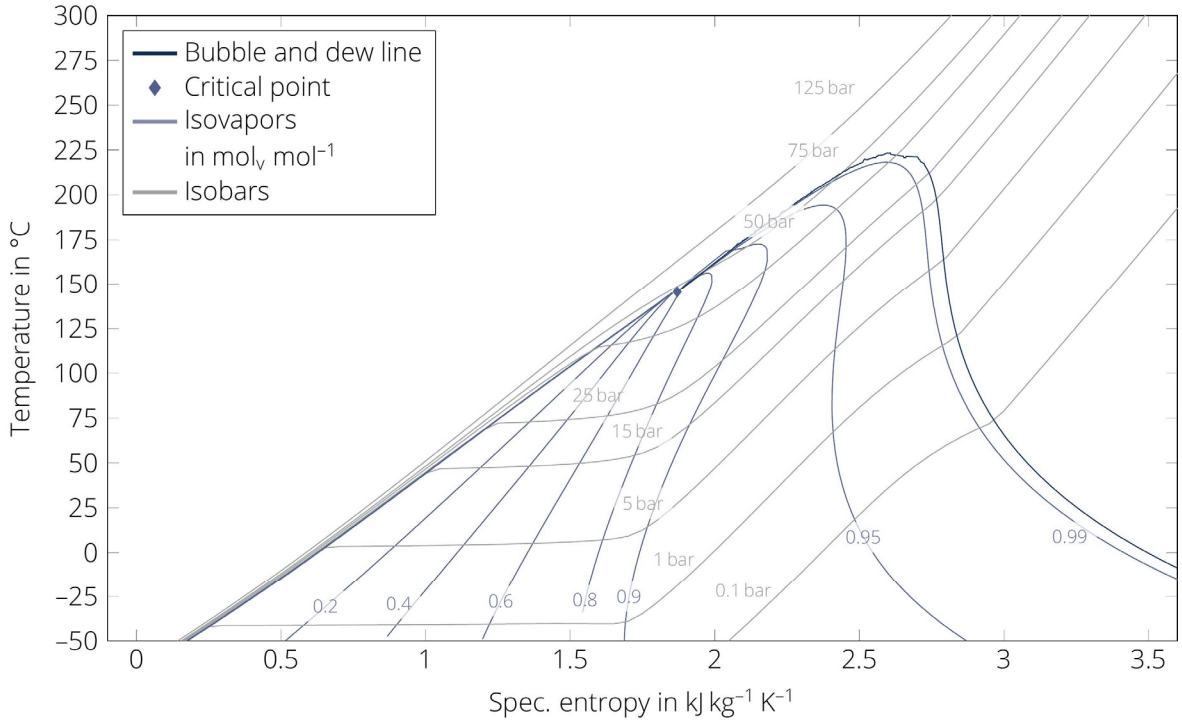


Figure 1. Illustration of the asymmetric 95 mol % propane and 5 mol % n-dodecane based on REFPROP 10 [4].

The non-linear temperature profile of the isobar is clearly visible, which corresponds almost exactly to an isotherm up to a vapour quality of approx. 85 to 90 % and then exhibits an almost linear temperature increase. This characteristic enables the formation of the aforementioned liquid/ two-phase recuperation. Until now, there has been no experimental evidence for this theory. Initially, this publication describes the design of a test bench for the proof of concept in section 2.1. Furthermore, modifications to the test bench are explained, which are intended to quantify the phenomena observed in the proof of concept more precisely (section 2.2). The results of this proof of concept are discussed in the results section (section 3).

2. Experimental approach

A two-stage approach was chosen for the experimental work. The first step was to verify the functionality of liquid two-phase recuperation with as little effort as possible. The exact behaviour of the cycle can only be predicted with a high degree of uncertainty. It was therefore unclear what structural arrangements would be required for its implementation. It was assumed from the beginning that the test bench would have to be modified and adapted to the cycle's behaviour after the initial tests. For this reason, the installation of expensive, precise and extensive measurement technology was dispensed with in the first phase. In the second step, based on the observations made, modifications have been carried out, and measurement technology was installed to enable a quantification of the observed effects.

2.1. Proof of concept

2.1.1. Cycle design and requirements

The simplest version of the RTPC is a five-component cycle. From an experimental point of view, implementing a heat pump cycle is easier than a power cycle, as only an expansion device is required in comparison to a pump. Therefore, a setup consisting of a recuperator, a compressor, a condenser, an expansion device, and an evaporator was chosen. As the objective is the proof of concept of a liquid-two-phase recuperation, it was decided to observe the transient ramp-up of the cycle. A temperature lift based on recuperation occurs when the temperatures of the heat source and the temperature of the heat sink diverge. For this reason, a tempered water bath was chosen as the heat source and an air volume as the heat sink. The air volume is located under a casing and is heated by the ongoing process (see Fig. 2).

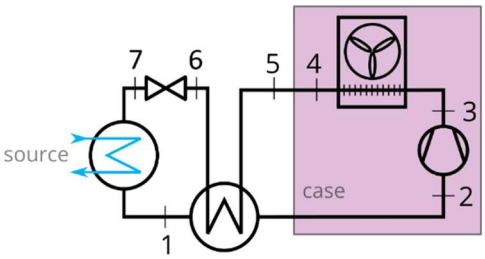
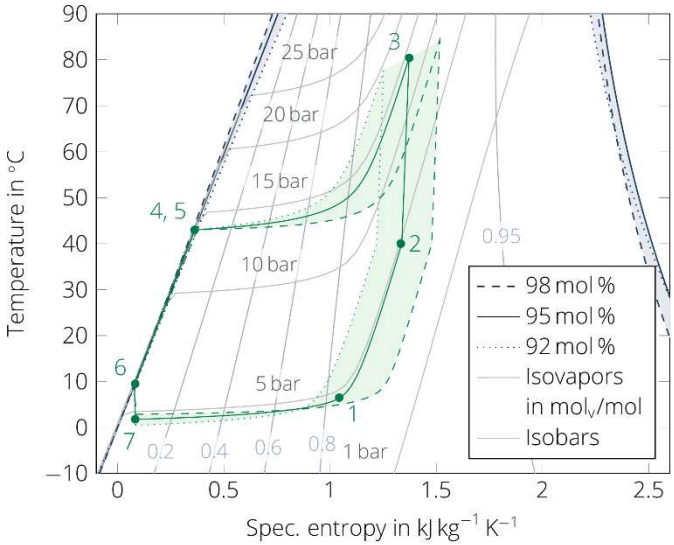


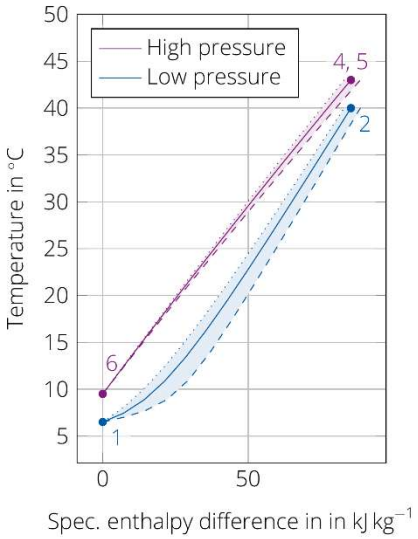
Figure 2. Concept of the test bench.

The intended end point for such a transient ramp-up is set at a water temperature of 8 °C and an air temperature of 40 °C. The reason for this is, on the one hand, to prevent the water in the evaporator from freezing and, on the other hand, because an air temperature of 40 °C already results in a very high compressor discharge temperature and thus an enormous thermal stress of the compressor.

For these boundary conditions, 95 mol% propane and 5 mol% n-dodecane (see Fig. 1) were identified as the most promising asymmetric mixture. Propane was chosen as the main fluid over isobutane because of its good volumetric cooling capacity at the temperatures in question. Furthermore, pressure measurement is more accurate due to the higher pressure levels. As an auxiliary fluid in combination with propane, n-dodecane achieves the highest expected COP (e.g. in comparison with shorter-chain hydrocarbons), which is related to the temperature gradient of the isobars in the two-phase region. The composition of 95-5 was chosen to balance cycle and operation aspects. A lower fraction of propane would lead to an undesirable increase of the temperature glide at low vapour quality. A higher fraction of dodecane, on the other hand, would result in a higher discharge temperature. This can be seen in Fig. 3, which shows the intended end points of the ramp-up for different compositions of the propane-dodecane mixture.



(a) Cycle profile range depending on the mixture composition.



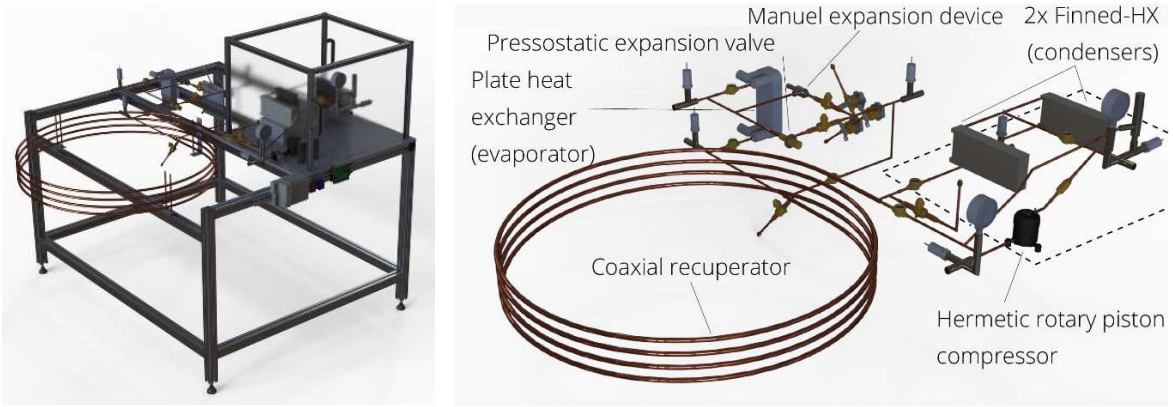
(b) Temperature profile range in recuperator.

Figure 3. Calculated cycle at the end of a ramp-up with different compositions of the working fluid mixture.

The diagram clearly shows that, unlike in the state of the art, compression must take place in the two-phase region. This must be taken into account when selecting the compressor. In addition to the cycle shape, the Figure also illustrates the effect of a possible composition shift. Composition shift refers to the effect whereby the circulating and local composition in the working fluid differs from the composition of the fluid that was filled into the system [5]. In this case, due to the widely differing properties of propane and dodecane, it is likely that dodecane-rich liquid will accumulate in certain areas (such as in the compressor) and that this dodecane mass will no longer be available for circulation. This would result in an increasing circulating composition of propane. Fig. 3 shows that a composition shift effects the vapour quality of the state points and the temperature profiles of the cycle considerably. Particular attention should therefore be paid to this aspect during the evaluation and quantification. Another effect that can influence the composition shift is the different solubility of the two working fluid components in the lubricating oil. For this reason, the process should be operated oil-free. This must also be taken into account when considering the compressor.

2.1.2. Components and arrangement

In order to implement the cycle under the required boundary conditions, a test bench was developed, as shown in Fig. 4. An overview of the main components is given in Table 1.



(a) Rendering without isolation. (b) Component arrangement.

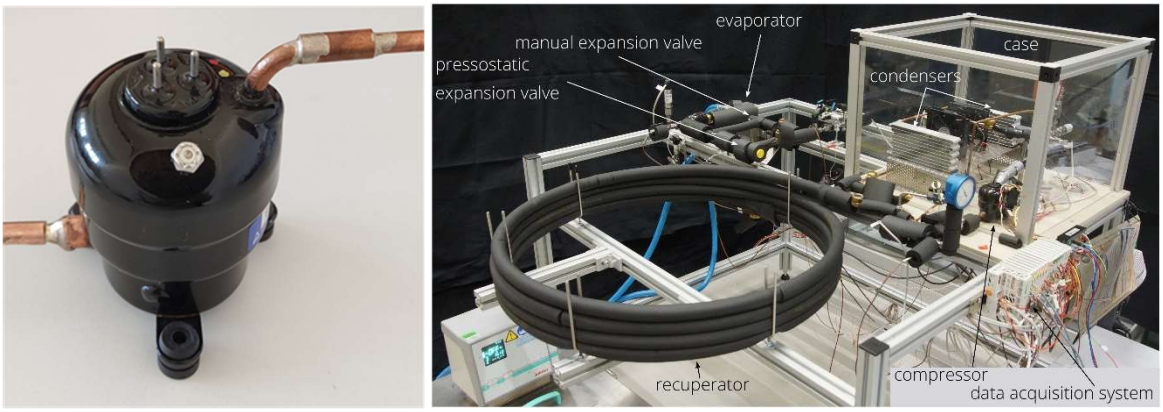
Figure 4. Component arrangement of the test bench.

Table 1. Used equipment in the test bench.

Component	Type	Manufacture	Identification	Specification
Compressor	Rotary piston	Aspen	Q4-24-5111	1.4 cm^3
Recuperator	Coaxial	In-house		Inner: DN 6 Outer: DN 10 Length: 10 m
Evaporator	Plate heat	DieTerHoeven	B3-12-12	
Condenser	Finned microchannel	Alcoil	C3.3x7x1.25S CU-stock	2x in parallel
Manual expansion device	Flow metering valve	Fitok	MSSS-ML6-A	$C_v = 0{,}004$
Automatic expansion device	Presso-static	Honeywell	AEL 0.5	
Heat source	Tempered bath	Julabo	FP50-HL	Water filling

A conventional plate and finned-microchannel heat exchanger were selected as an evaporator and condenser. To keep the temperature difference low during condensation, the heat transfer surface was increased by using two identical heat exchangers in parallel. The recuperator and compressor presented a particular challenge. The recuperator is an in-house manufactured coaxial heat exchanger (tube-in-tube) with a very large radius. Evaporation takes place in the inner tube. The geometry was chosen to influence the two-phase flow as little as possible and thus make the existing flow pattern predictable. Furthermore, it has a slight incline to ensure that, liquid working fluid flows back from the compressor suction port towards the evaporator during standstill.

A compact, speed-controlled rotary piston compressor from the company Aspen was chosen. Two key aspects were essential for this decision. Under normal operating conditions, rotary pistons carry a considerable amount of oil, which provides lubrication and sealing between the vane and the cylinder in the working chamber. This means that a certain degree of liquid tolerance can be assumed. The compact design was chosen in order to minimise the separation effects of liquid volumes inside the compressor. To enable the compressor to suck in a two-phase flow, the liquid separator and the damping windings were removed from the compressor (Fig. 5 (a)). It has already been mentioned that any interaction between the working fluid mixture and a lubricant for the compressor should be avoided as far as possible in order to minimise the number of unknown impact parameters. For this reason, the compressor oil was flooded with oil only once and drained out before installation in the system. This meant that only an initial lubricating film was present. Since the test rig was not designed for continuous operation, the associated risk was considered acceptable.



(a) Rotary compressor before the assembly. (b) Test bench in the laboratory.

Figure 5. Photos of the test bench.

A conventional thermostatic expansion valve cannot be used as the expansion device, as there is no superheating before the compressor. For this reason, a presso-static expansion valve was installed, with the spring set firmly to a pressure of approx. 4.7 bar. This corresponds to the intended evaporation pressure. In addition, a flow metering valve was installed in parallel as a manual expansion device to adjust further operating pressures. The entire test bench is shown in Fig. 5 (b). The results achieved with this test bench configuration are documented in section 3.

2.2. Redesign for detailed quantification

Following the successful proof of concept, the test bench was modified. The aim of this modification is to quantify the cycle more accurately. The flow diagram of the redesigned test bench is shown in Fig. 6. The basic structure of the heat exchanger assemblies was retained. However, the temperature and pressure measurement technology were replaced with more accurate sensors (see Table 2). This conversion also included improving the positioning of the sensors in order to minimise the effects of heat and pressure losses. Furthermore, a Coriolis measuring device was added, which can be used to determine both the mass flow and the density. The lack of mass flow determination was a significant disadvantage during the proof of concept, as power output could not be proven without it. The aim of the density measurement is therefore to obtain a continuous measurement of the circulating composition. This is possible with the working fluid propane-dodecane, as the densities are very different. At 16 bar and 10 °C, propane has a density of around $517\ g\ l^{-1}$ and dodecane of $758\ g\ l^{-1}$ [4].

Additional service ports were also installed to simplify filling and removal. The particular challenge arises from the fact that dodecane cannot be evacuated from the system due to its low vapour pressure at room temperature.

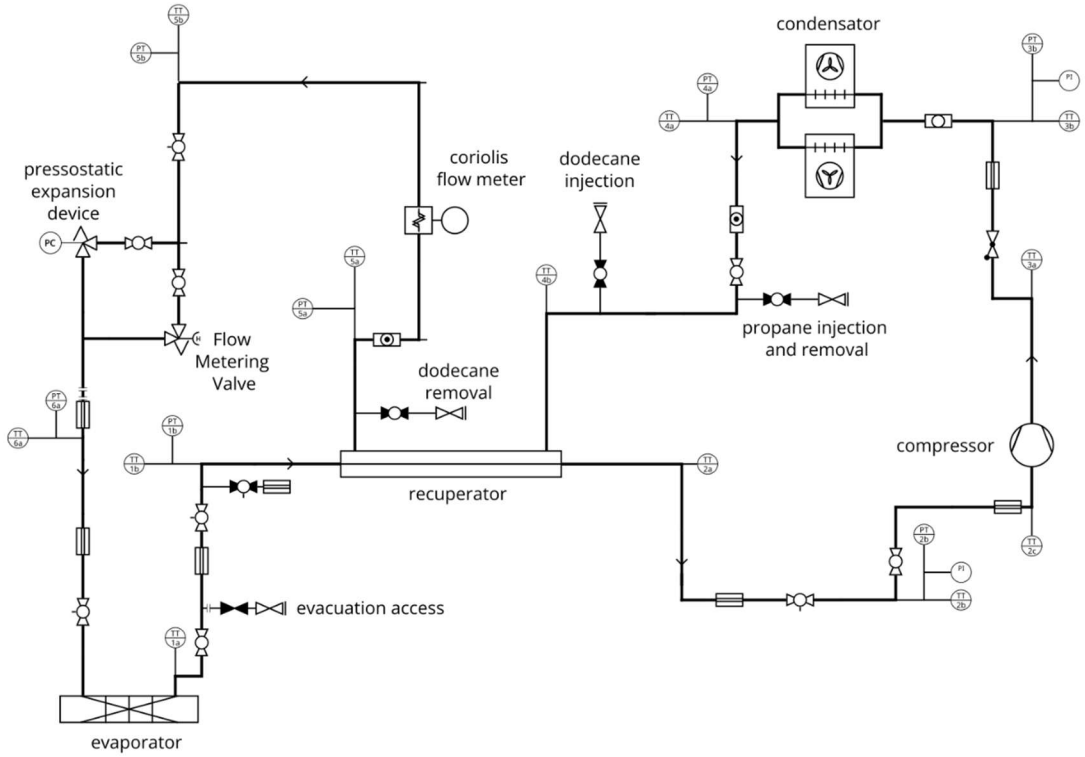


Figure 6. PID of the redesigned test bench.

Table 2. Intended sensor equipment of the redesigned test bench.

Parameter	Principle	Manufacture	Type	Tolerance	Specification
Temperature	Resistive	Heinz Messwiderstände	PT1000	$\pm 0.1\,°C$ at $0\,°C$	ø 1 mm, four-wire
Pressure	Piezo-resistive	Keller	PAA-33X	$\pm 0.1\%$ full scale	0...16 bar, 0…30 bar
Density and mass flow	Coriolis	Anton Paar	L-Cor 8000	$\pm 3\ g\ l^{-1}$, ±0.2%	

2.3. Approaches for quantification of the composition

One finding from the experiments on the proof of concept is that only limited information can be gained from observing the two-phase flow in conventional refrigerant sight glasses. Although it was possible to observe liquid and vapour fractions, no reliable conclusions could be drawn about their flow and flow pattern. For this reason, it was decided to replace the regular sight glasses with glass tubes. These tubes are made of borosilicate glass and were purchased from Schott. They have the same geometric data as the copper tubes and are approved for pressures up to 28 bar. They are sealed in conventional clamp ring fittings, with the cutting and clamping rings being replaced by PTFE ones. It should be noted that the sight glass tubes are covered by an acrylic glass tube with an outer diameter of 44 mm during installation as an additional safety measure during operation. A total of seven sight glass tubes are planned for the circuit. Fig. 7 illustrates their installation in the circuit.

The most complex new features on the redesigned test bench are the sampling valves. These are used to collect samples from the liquid and two-phase flows for subsequent analysis in a gas chromatograph mass spectrometer (GCMC). A GCMS of the company Shimadzu is available in the laboratory, which has a peak area repeatability of < 1% RSD. The aim is to determine the circulating composition and local compositions at individual locations in the cycle. A valve for two-phase sampling is shown in Fig. 7 as well. The valves are an own development and are manufactured in-house. The aim was to sample a representative cross-section of the working fluid flow, whereby no relevant mass may be removed from the process. At the same time, the process must not have to be interrupted by the sampling.

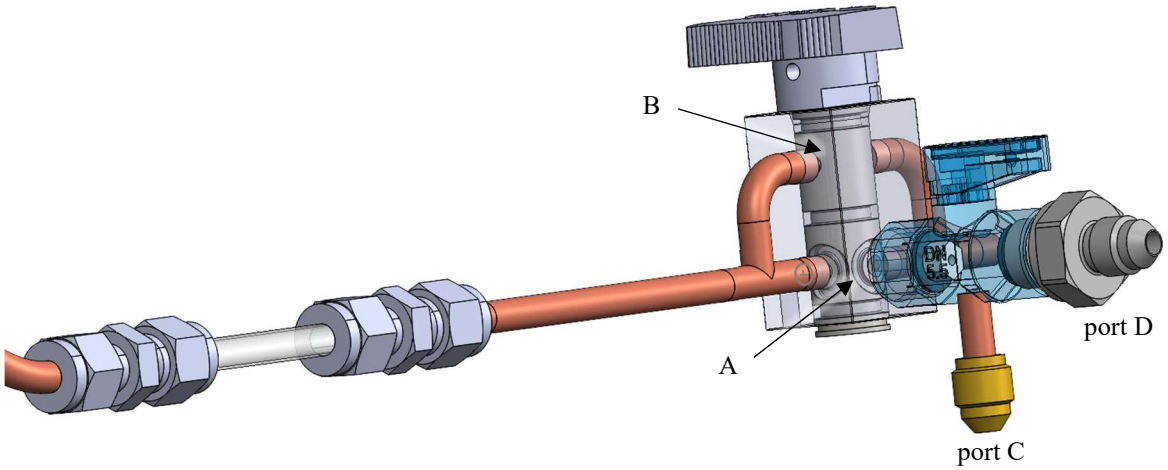


Figure 7. Rendering of the sight glass tube and the two-phase sampling valve at measurement point 2.

The non-inversive sample can be taken with a double-pass plug valve (see transparent illustrated component in Fig. 7). Either the lower (A) or upper (B) passage is open. The internal volume of passage A represents the sample volume. In the case of a liquid sample, this corresponds to 115 µl, and in the case of a two-phase sample, 176 µl. The sample can be taken from the passage A via a side bore. For the purpose of its extraction, a side valve with two connections is installed. Port C is used for the evacuation of the sample volume before and after the sample collection, and the port D allows the sample to be removed using a gas-tight syringe. The design for taking liquid samples differs not only in terms of sample volume. There is a bore and a side valve on both sides to enable the sample volume to be flushed.

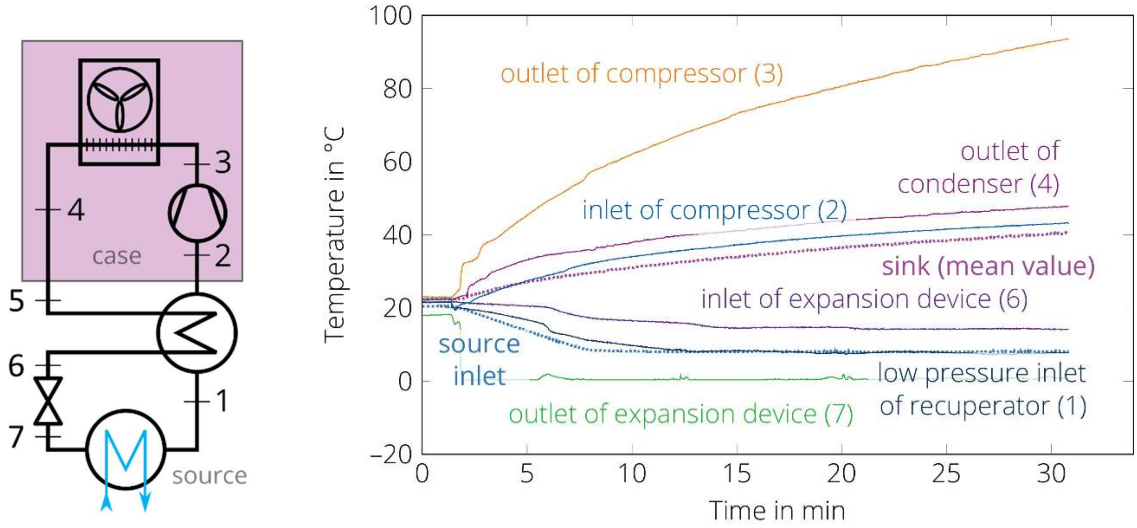
A particular challenge with asymmetric mixtures is the homogenisation of the sample. Homogenisation is necessary to prevent separation of the mixture components during injection into the GCMS. In the literature, an often-used approach is to expand the sample into an evacuated cylinder (e.g. [6, 7]). Furthermore, it is possible to heat up the sample. Neither can be applied in the present case, as the auxiliary fluid has a very high boiling point and very low vapour pressure. The sample of the asymmetric mixture is always in a two-phase state at ambient pressure and temperature. A direct liquid injection via a gas-tight syringe is not possible because the pressures needed would have to be too high at room temperature. The boiling point of the propane-dodecane mixture to be analysed, on the other hand, is above 100 °C and therefore too high for the injection port of the GCMS available. Furthermore, a homogenisation by reducing the pressure below the ambient pressure is not helpful, neither, because the vapour pressure of the mixture at room temperature is as low as 2 mbar. For this reason, a solvent-based method is proposed. Sampling from the two-phase flow is carried out as follows. First, the passage A of the plug valve, and thus the sample chamber, is perfused. The side valve is evacuated via port C. A gas-tight syringe containing a solvent is connected to port D of the side valve. The plug valve cylinder is then turned by 90°, separating the sample volume from the working fluid circuit. The flow passes through the passage B now. The sample mass expands via the side bore into the evacuated area of the side valve. The solvent is now also injected into this area. The respective volumes or masses are selected so that the entire working fluid quantity dissolves in the solvent, creating a homogeneous liquid phase. After a pressure balance has been achieved, a liquid sample can be taken from this phase and introduced into the GCMS. A correspondingly larger quantity of solvent must be used for the liquid sampling. Preliminary tests on this method have already been carried out.

3. Results and Discussion

The results shown in section 3 were obtained on the basis of the test bench described in section 2.1. The focus was providing qualitative evidence of the fundamental technical feasibility of liquid two-phase recuperation. Quantitative conclusions cannot be drawn on this basis yet, particularly because the mass flow was not measured during these tests. Although the compressor is only used to generate a pressure difference in this experiment and is not itself the subject of investigation, some observations regarding its operation can be shared, as it is not specified for oil-free, two-phase compression.

3.1. Ramp-up tests and proof of concept

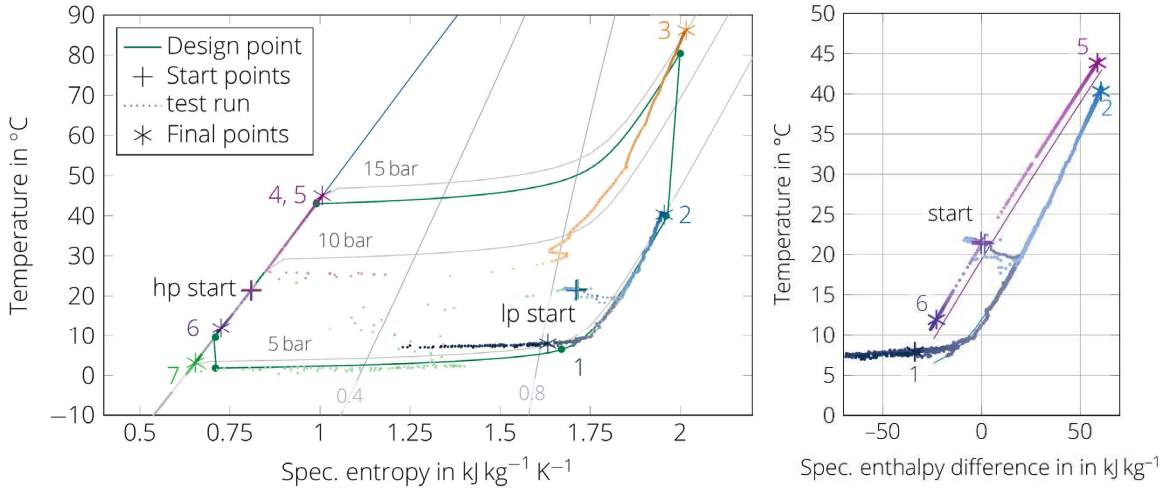
Initially, ramp-up tests were conducted to observe the development of liquid two-phase recuperation. Fig. 8 shows the temperature profiles for a test with a compressor speed of $4000\ min^{-1}$.



(a) Simplified flow chart (b) Temperature measurement data

Figure 8. Temperature development during a transient ramp-up test with $4000\ min^{-1}$.

During the experiment, the source temperature was decreased from ambient temperature to 8 °C. The air temperature under the case of the test rig is a result of the energy input by the condenser and compressor and increases during the experiment. The temperature at the outlet of the expansion device is kept almost constant during the experiment due to the presso-static control, while the other temperature curves are either related to the water or air temperature. This already indicates that a recuperation takes place and already confirms the basic idea of a transient ramp-up of the cycle, as intended. The experiment had to be stopped before reaching a stationary state due to the high compression discharge temperature. Therefore, another experiment at 3000 min^{-1} was performed. After about 90 minutes the temperature limit was reached. To demonstrate that a liquid-two-phase recuperation is actually occurring during this ramp-up, the thermodynamic states are calculated for every second. However, this requires the assumption of an effective composition. In a first step, the assumption is made, that this composition corresponds to the initial composition of 95 mol% propane and 5 mol% dodecane. The resulting T-s diagram is shown in Fig. 9.



(a) Profiles of all state points of the cycle.

(b) Profiles in the recuperator.

Figure 9. Transient evolution of the cycle during a ramp-up test of 90 minutes calculated with the initial composition of 95 mol% propane and 5 mol% dodecane. The colour intensity serves as the time axis. It increases with the duration of the experiment.

The time axis is illustrated in the diagram by the intensity of the colours for the individual state points. The darker the colours, the further the experiment has progressed. Since the high-pressure (hp) side is separated from the low-pressure (lp) side by valves, there are two starting points. At ambient temperature, the ramp-up for the hp side begins at the boiling line, while for the lp side it begins in the two-phase region. The diagram also gives the originally calculated cycle as a solid line. Fig. 9 (a) shows how the points of state approach the design points as the experiment progresses, forming a parallelogram. However, due to a greater temperature difference between the working fluid and the air in the condenser, higher temperatures – and consequently higher pressures – are reached than anticipated in the design cycle. Fig. 9 (b) shows the specific heat exchanger diagram of the recuperation. Particularly relevant is the fact that the temperature difference of the subcooling in the liquid phase of the recuperator corresponds to the temperature difference during evaporation on the low-pressure side of the recuperator (both are approx. 33 K). Equal temperature differences represent the thermodynamically optimal case (counter flow heat exchanger). However, as can be observed in Fig. 9 (b) for the end points of the ramp-up, the energy balance along the recuperator is not fulfilled assuming the initial composition,which can be expressed by

$$0 \neq h_1(z_{init}) - h_2(z_{init}) + h_5(z_{init}) - h_6(z_{init}). \quad (1)$$

Heat losses cannot adequately explain this effect, as the recuperator is insulated and its average temperature level is almost the same as the ambient temperature. Instead, a composition shift is considered to be the reason. The compressor housing thus presumably acts as a separator for liquid droplets rich in dodecane, which consequently accumulate in the compressor sump and are therefore no longer part of the working fluid circulation. The following approach is therefore taken. For the time steps, a circulating composition z^* is determined based on the assumption of

$$0 = h_1(z^*) - h_2(z^*) + h_5(z^*) - h_6(z^*). \quad (2)$$

As the vapour-liquid equilibrium is of crucial importance for the calculation, the revised interaction parameter of propane-dodecane from [8] were used for these calculations rather than the standard Refprop interaction parameters. The resulting change in composition over time is shown in Fig. 10.

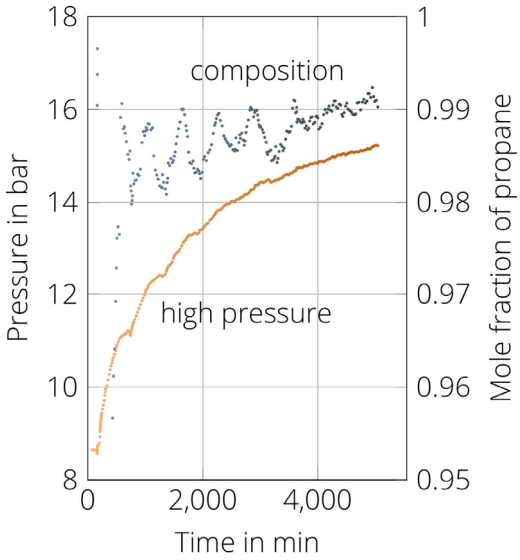


Figure 10. Composition evolution during a ramp-up test at 3000 min^{-1} calculated based on the energy balance of the recuperator.

It becomes clear that, during the experiment, the composition shifts to a value between 98 mol% and 99 mol%. The fluctuations in composition can be explained by slight variations in high pressure level. Due to the changing composition during the ramp-up, a third axis is added to the T-s diagram, which describes the fraction of propane in the working fluid circulation. This diagram is shown in Fig. 11.

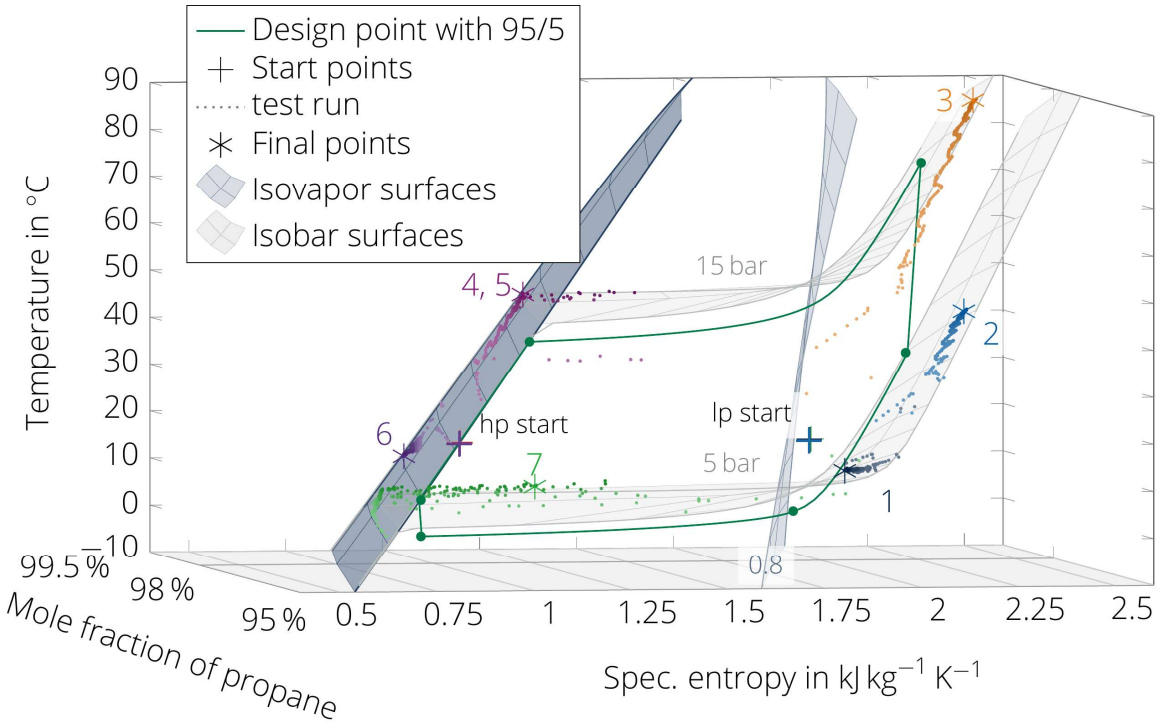


Figure 11. Transient evolution of the cycle during a ramp-up including a changing circulating composition calculated based on the energy balance of the recuperator. The colour intensity increases with the duration of the experiment and represents the time axis.

Despite this shift, the basic cycle shape is preserved, as the asymmetric behaviour of the mixture is even stronger with the higher fraction of propane.
Generally, the experiment shows that the mixture evaporates on the low-pressure side of the recuperator and that compression takes place in the two-phase region. Moving liquid working fluid could also be observed in the refrigerant sight glasses installed before and after the compressor. From these observations, it can be concluded that liquid two-phase recuperation can technically be realised by employing an asymmetric mixture.

3.2. *Stationary operating points and composition shift*

In addition, several stationary operating points were recorded at the end of the ramp-up at different compressor speeds and source temperatures. The same analysis approach (Eq. 2) was used here. Fig. 11 shows the steady-state points and the calculated circulating compositions are shown next to point 1. For orientation purposes, the T-s diagram indicates the isolines for the minimum and maximum composition observed at 2800 and 2600 min^{-1}, respectively, and in addition the design cycle with the composition 95-5. As expected, lower compressor speeds result in a reduced high pressure, as the compressor's volumetric efficiency and the mass flow rate decreases. Tests were also carried out at 2400 min^{-1} and 2200 min^{-1}, resulting in final pressures of 13.99 and 12.89 bar respectively. However, these points revealed the limitations of the evaluation method based on Eq. 2 and the fundamental limitations of the test rig. Calculating a composition based on the energy balance would result in non-physical conditions at the state point 5. The cause could be an insufficient mass flow rate, for which the assumption of the energy balance no longer applies. However, the most important finding from this speed variation is that the liquid two-phase recuperation is still present and that the cycle seems to adapt to the external boundary conditions due to the varying composition. To further evaluate this statement, the source temperature was then varied slightly for a fixed speed of 3000 min^{-1}.

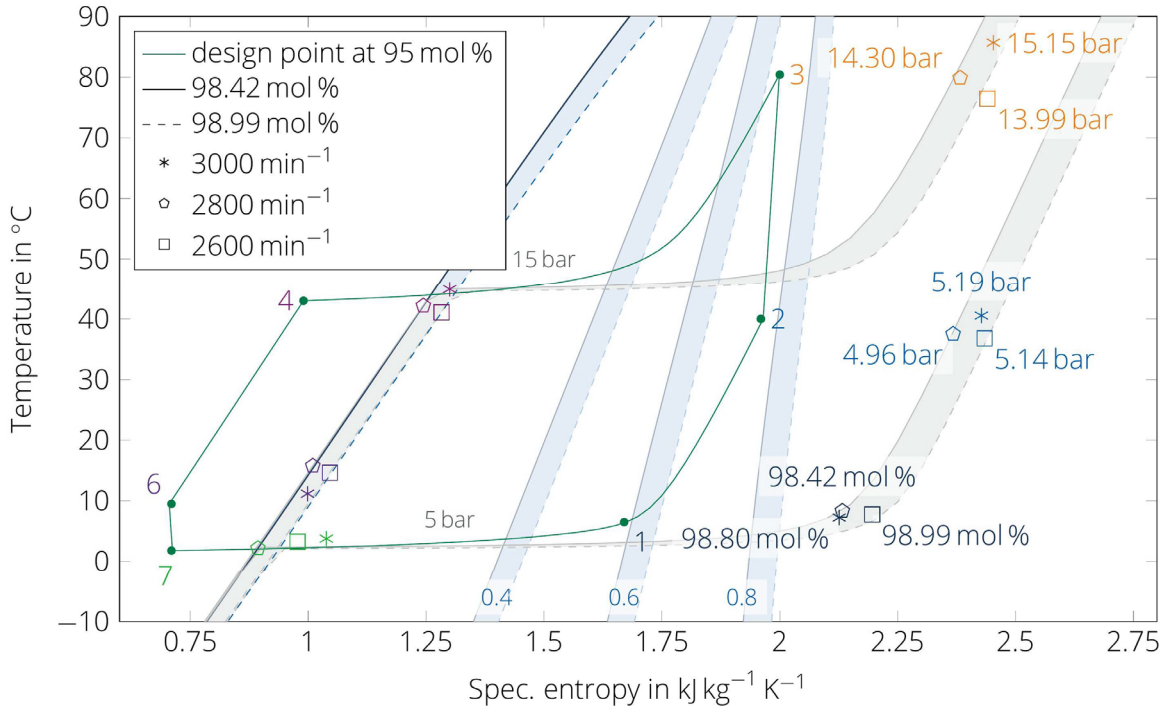


Figure 11. Steady-state point at different compressor speeds.

In three cases, the presso-static expansion device was used and only the water bath temperature was decreased from 8 °C to 7 °C and to 6 °C. In the fourth case, the source temperature was left at 8 °C, but the evaporation pressure was increased using the manual valve to 5.53 bar. Surprisingly, there was no noticeable change in the cycle behaviour. Even the higher low-pressure level during manual control did not lead to a breakdown of the cycle. This can only be explained by an even more asymmetrical behaviour of the working fluid, as occurs when a higher proportion of propane is circulating. Unfortunately, even in these cases, calculating the composition using the energy balance yields non-physical values; however, a general trend towards circulating propane fractions of over 99 mol% is conceivable. This observation is extremely significant for two reasons. Firstly, it demonstrates the adaptability of the cycle and, secondly, a higher evaporation pressure at the same source temperature means lower specific compressor work. The composition shift could therefore lead to a more efficient RTPC in a technical implementation than assumed in the calculations. The next step is the quantitative investigation of this observations.

3.3. Two-phase operation of the compressor

The operating conditions for the compressor were initially considered extremely challenging, as it was started up with only an initial lubricating film and two-phase compression had to be achieved. However, the compressor has not failed to date, and even after approximately 15 hours of operation, no acoustic degradation is detectable. On the contrary, a number of interesting qualitative observations have been made. When the compressor is switched off, re-expansion takes place. As a result, a hot liquid wave flows back from the compressor suction port into the recuperator. The liquid has to come from the working chamber itself and not from an additional separator, as the suction line extends directly into the working chamber. Another qualitative observation concerns liquid suction. If little liquid reaches the compressor when it starts up, for example because only a small amount of working fluid was present on the low-pressure side, this leads to a higher discharge temperature and lower achievable high pressures during subsequent operation.

These observations suggest that the liquid has a positive effect in the compressor. One explanation for this could be that accumulated quantities of liquid in the working chamber have a cooling and lubricating effect. This is particularly conceivable because a dodecane-rich liquid has a significantly higher viscosity than pure propane. While the dynamic and kinematic viscosity of liquid propane at 40 °C is 0.08 mPas and 0.177 mm^2s^{-1}, that of dodecane is 1.06 Pa s and 1.44 $mm^2\ s^{-1}$, which is higher than that of water (0.65 mPas and 0.66 mm^2s^{-1}) [4]. This could also explain why the compressor is still operational even without lubricating oil. However, it should be noted that the compressor manufacturer originally specified a viscosity of around 69 $mm^2\ s^{-1}$.

4. Conclusion and outlook

This publication describes the first experimental realisation of the Recuperative Two-Phase Cycle (RTPC). This is a new approach of thermodynamic cycles. The fundamental principle is a liquid two-phase recuperation, which is enabled by using an asymmetric working fluid mixture. When applied to a heat pump cycle, this should reduce isenthalpic throttling losses. The following aspects of the experimental work were presented:

1. The construction of a basic test bench is explained. It is composed out of five main components and charged with the asymmetric working fluid 95 mol% propane and 5 mol% dodecane. The simple form of the RTPC that is being targeted in the test bench requires an almost oil-free and two-phase compression process.
2. This test bench was used to carry out the proof of concept for liquid two-phase recuperation. Observations were made which indicate that a strong composition shift is present, however, this shift does not affect the feasibility or the fundamental cycle shape.
3. These observations were used to derive modification and supplementation measures that can be used to quantify the qualitative findings of the proof of concept.
4. To investigate the composition shift, a sampling approach was presented that is based on the use of a solvent to homogenise the sample.

The next step will be to carry out the quantification tests on the redesigned test bench.

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References

[1] Bederna B.G., Hesse, U. (2021). Vorrichtungen und Verfahren zum Durchführen von Kreisprozessen. DE102021114326.

[2] Bederna, B. G., Barta, R. B., Thomas, C. (2023). Thermodynamic concept of a novel recuperative two-phase heat pump cycle. 26th IIR Int. Congress of Refrigeration. https://doi.org/10.18462/iir.icr.2023.0588

[3] Bederna, B. G.,Thomas, Ch. Introduction and thermodynamical investigation of the Recuperative Two-Phase Cycle. Applied Thermal Engineering, https://doi.org/10.1016/j.applthermaleng.2026.130898

[4] Lemmon, E. W., Bell, I. H., Huber, M. L., & McLinden, M. O. (2018). NIST Standard Reference Database 23: Reference Fluid Thermodynamic and Transport Properties-REFPROP, Version 10.0, National Institute of Standards and Technology. https://doi.org/https://doi.org/10.18434/T4/1502528

[5] Chen, J., Kruse, H. (1995). Calculating Circulation Concentration of Zeotropic Refrigerant Mixtures, HVAC&R Research 1 (3) 219–231. https://doi.org/10.1080/10789669.1995.10391320

[6] Xu, X., Liu, J., Cao, L., & Li, Z. (2012). Local composition shift of mixed working fluid in gas–liquid flow with phase transition. Applied thermal engineering, 39, 179-187. https://doi.org/10.1016/j.applthermaleng.2012.01.064

[7] Jeong, Y., Lee, S., & Kim, M. S. (2025). Local composition shift of refrigerant mixtures and its effect on cooling performance in a heat pump system. Energy Conversion and Management, 333. https://doi.org/10.1016/j.enconman.2025.119809

[8] Schnelting, L. , Bederna, B. G., Raabe, G., Römer, U., Thomas, Ch. Bayesian Calibration of HEOS Mixing Models and Impact on Thermodynamic Cycles. ChemRxiv. 2025, https://doi.org/10.26434/chemrxiv-2025-vnm16