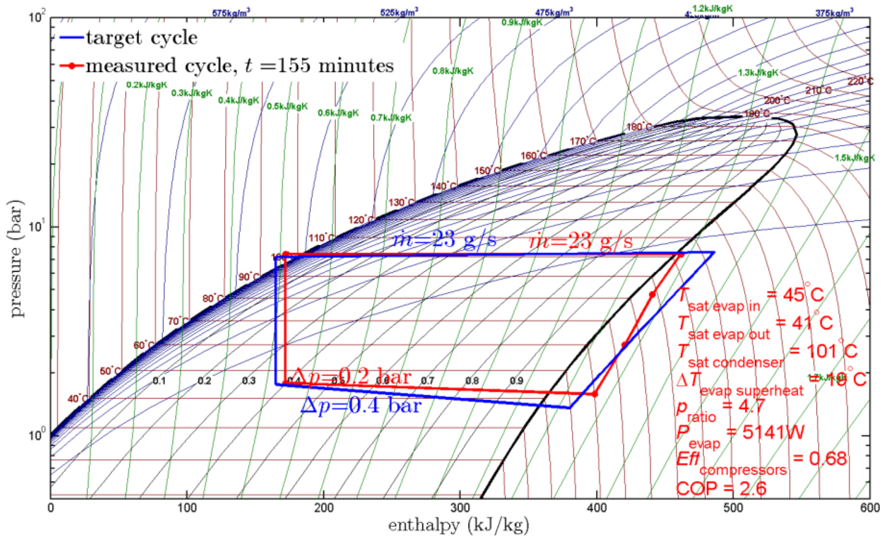
Nationaal Lucht- en Ruimtevaartlaboratorium

National Aerospace Laboratory NLR

## Executive summary

# Heat Pump Conceptual Study and Design

## Overall assessment and further work




**Problem area**
The classical method for spacecraft thermal control is that the waste heat is conducted to the radiator, where it is radiated to space. Therefore, the radiator temperature must be lower than the equipment temperature. A Heat Pump uses a compressor to raise the radiator temperature above the temperature of the equipment which results in a higher heat rejecting capacity compared to a conventional radiator with the same surface area.

**Description of work**
This report is an assessment of the heat pump project, and contains recommendations for further work.

**Results and conclusions**
In this project, a heat pump breadboard with novel high-speed turbo compressors has been designed and built. The breadboard demonstrates that the heat pump is very efficient: At the target setting (saturation temperature of 45°C at the evaporator, 100°C at the condenser, and a 'payload' heat input of 5 kW), the measured COP is 2.6, which is considerably higher than the requirement of 2. The test program could not be finished completely, because particles from the compressor sealing got stuck in its diffusor. The recommendations for future work include space qualifying the compressor electronics, increasing compressor lifetime, and designing a receiver that works in zero gravity.

**Applicability**
The heat pump developed in this project can be used for GEO satellites and Lunar applications.

**Report no.**
NLR-CR-2014-009

**Report classification**
UNCLASSIFIED

**Date**

**Knowledge area(s)**
Ruimtevaarttoepassingen

**Descriptor(s)**
Heat Pump
two-phase
compressor
ECS
refrigeration

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# Heat Pump Conceptual Study and Design

## Overall assessment and further work

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| | |
|---|---|
| Customer | European Space Agency |
| Contract number | ESTEC Contract No.4000105360/12/NL/EM |
| Owner | NLR + partner(s) |
| Division NLR | Aerospace Systems and Applications |
| Distribution | Limited |
| Classification of title | Unclassified |

Approved by:

| Author<br>Henk Jan van Gerner | Reviewer<br>Johannes van Es | Managing department<br>Michel Keuning |
|---|---|---|
| Date: | Date: | Date: |

## Summary

The classical method for spacecraft thermal control is that the waste heat is conducted to the radiator where it is radiated to space. Therefore, the radiator temperature must be lower than the equipment temperature. A Heat Pump uses a compressor to raise the radiator temperature above the temperature of the equipment which results in a higher heat rejecting capacity compared to a conventional radiator with the same surface area. For example, when a heat pump is used to lift the radiator temperature from 45°C to 100°C, the heat rejecting capacity of a GEO satellite can be increased with 83%. However, commercially available compressors have a high mass (40 kg for 10kW cool capacity), cause vibrations, and are intended for much lower temperatures (maximum 65°C) than what is required for the heat pump application (100°C). Dedicated aerospace compressors have been developed with a lower mass (19 kg) and for higher temperatures, but these compressors have a lower efficiency.

In the preliminary design report for this project, a heat pump system was presented with a novel electrically-driven turbo compressor system with a mass of 2kg and a higher efficiency than existing aerospace compressors. The selected fluid for the heat pump is isopentane. A detailed design for a heat pump breadboard was presented in the detailed design report for this project. The assembly of the breadboard, the test plan and procedures are discussed in the test readiness report, and the test results are discussed in the breadboard test report. This report is an assessment of the heat pump project, and contains recommendations for further work.

At the target setting (saturation temperature of 45°C at the evaporator, 100°C at the condenser, and a 'payload' heat input of 5 kW), the measured COP is 2.6, which is considerably higher than the requirement of 2. The COP is higher because the efficiency of the compressors is higher than expected, and the pressure drop in the evaporator is lower. During one of the tests, one of the compressors started to make a noise and showed a decrease in performance. After opening the compressor, it was found that particles from the inlet and outlet connection sealing got stuck in the diffusor of a compressor. As a result, the test program could not be finished completely.

The recommendations for future work include space qualifying the compressor electronics, increasing the lifetime of the compressor, a method to cool the compressor with its own fluid, and designing a receiver that works in zero gravity.

# Contents

# 1 Introduction

## 1.1 Background

The classical method for spacecraft thermal control is that the waste heat is conducted to the radiator, where it is radiated to space. Therefore, the radiator temperature must be lower than the equipment temperature. In addition, the amount of heat rejected by a radiator is directly driven by the surface temperature and surface area. In essence, the limiting factors of the radiator performance are the equipment operating temperature and launch volume. Therefore, in order to increase the heat rejection capability using standard heat transfer method, a deployable radiator can be used to increase the radiating surface area or the operating temperature of the internal units can be increased. Another method is to increase the radiator temperature without increasing the temperature of the internal components. This is the major characteristic of a Heat Pump which makes this product unique since it has the capability of raising the radiator temperature above the temperature of the equipment where more heat can be rejected compared to a conventional radiator with the same surface area.
However, this increase in heat rejection capacity comes at price of a large mass penalty; Commercial available compressors with a cooling capacity of 10 kW at a $\Delta T$ of 55 K have a very large mass (e.g. 40 kg or 80 kg when for redundancy two compressors are used). Furthermore, the extra mass caused by the electric power for the compressor (due to larger solar panels, battery, and power control unit) is approximately 125 kg (assuming 25 kg/kW). For this reason, it is of utmost importance to use a very lightweight compressor with a high Coefficient of Performance (COP).
These characteristics are combined in the miniature electrically-driven compressors made by Celeroton. These compressors run at ultra-high speeds of up to 500,000 RPM, which results in very low masses, approximately 2 kg for a cooling capacity of 10 kW.

In the preliminary design report for this project [1], a heat pump system was presented with a novel electrically-driven turbo compressor system with a mass of 2kg and a higher efficiency than existing aerospace compressors [5-7]. The selected fluid for the heat pump is isopentane. A detailed design for a heat pump breadboard was presented in the detailed design report for this project [2], and a review of the assembled breadboard was presented in the test readiness report [3]. The test results of the breadboard are discussed in [4]. This report gives an assessment of the project and gives recommendations for future work.

## 1.2 Heat pump working principles

A vapor compression Heat Pump consists of a compressor, a heat exchanger at the hot source (i.e. the evaporator), a heat exchanger at the cold sink (i.e. the condenser), and an expansion valve (see Fig. 1 for a schematic drawing). The working fluid enters the compressor as a superheated vapour. The compressor increases the pressure and the temperature of the vapour. The vapour then travels through the condenser, where the vapour is condensed into liquid and the heat that is stored in the vapour is released. The liquid flows through the expansion valve, where the pressure abruptly decreases (adiabatic expansion), causing a partial evaporation of the liquid and a drop in the temperature. The cold liquid-vapour mixture then flows through the evaporator where it absorbs heat and completely turns into vapour before entering the compressor.

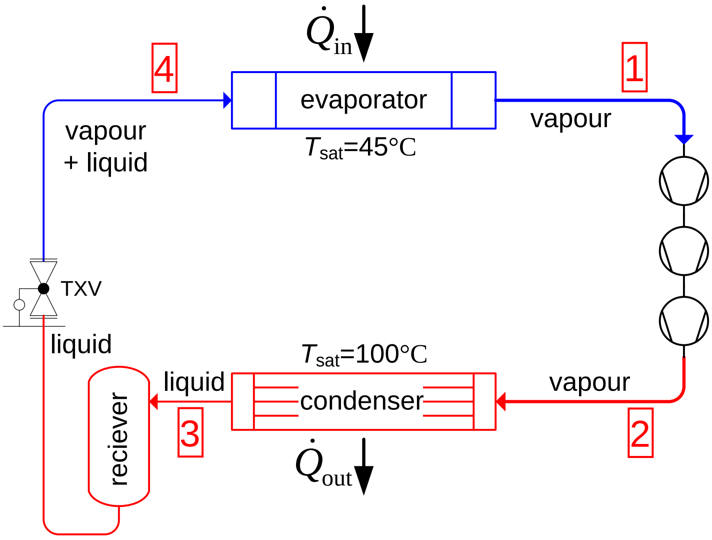


*Fig. 1 Schematic drawing of a basic vapour compression cycle*

The heat pump cycle with isopentane (R601a) as working fluid and where the saturation temperature is increased from 45 to 100°C is represented in the enthalpy-pressure diagram in Fig. 2. In the ideal cycle (solid line), the fluid leaves the evaporator and enters the compressor as saturated vapour. In an actual vapour compression cycle (dashed line), the vapour is slightly superheated (to 50°C) to ensure that the fluid is completely vaporized before it enters the compressor. In the ideal cycle, the compression process is isentropic, while in an actual cycle, the adiabatic efficiency of the compression process is approximately 60%. Furthermore, there is a pressure drop (which is assumed to be 0.4 bar) in the condenser, evaporator and transport lines of an actual heat pump.

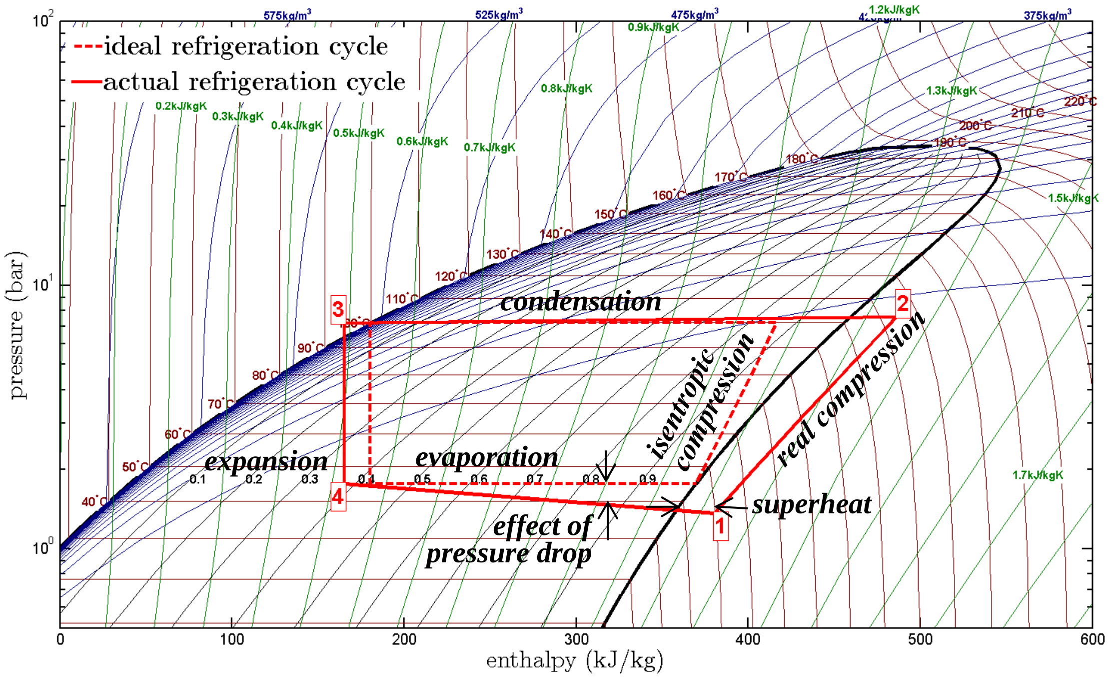


*Fig. 2 Pressure-Enthalpy diagram of a vapour compression cycle with isopentane (R601a) as refrigerant. The 'skewed dome' in the diagram is the two-phase region, i.e. the fluid in that region is a mixture of liquid and vapour. To the right of the dome, the fluid is vapour, to the left, the fluid is liquid. The green lines in the diagram are isentropic lines, the blue lines are isodensity lines, and the dark red lines are isothermal lines.*

## 1.3 Heat pump breadboard characteristics and fluid properties

The main characteristics of the heat pump breadboard are summarized in Table 1. Some properties of the selected working fluid are listed in Table 2.

| | |
|---|---|
| Saturation temperature in the evaporator | 45°C |
| Saturation temperature in the condenser | 100°C |
| Heat pump cooling capacity (evaporator heat load) | 5 kW |
| Total compressors power | ~2.5 kW divided over 3 compressors in a serial configuration |
| Condenser capacity | 7.5 kW (5 kW for heat load and 2.5 kW for compressor power) |
| Target COP | 2 |
| Working fluid | Isopentane (R601a) |
| Internal volume of system | ~3 liters total (~1 liter without the accumulator and receiver) |

*Table 1 Heat pump breadboard characteristics [1]*

| | |
|---|---|
| Saturation pressure at 45°C | 1.76 bar |
| Saturation pressure at 100°C | 7.20 bar |
| Triple point temperature | -161°C |
| Critical temperature | 187°C |
| Heat of evaporation $h_{lv}$ at 45°C | 0.33 MJ/kg |
| Vapour density $\rho_v$ at 45°C | 5.14 kg/m$^3$ |
| Liquid density $\rho_l$ at 20°C | 620 kg/m$^3$ |
| Liquid density $\rho_l$ at 45°C | 594 kg/m$^3$ |
| Liquid density $\rho_l$ at 100°C | 528 kg/m$^3$ |
| Thermal conductivity $k$ at 45°C | 0.10 W/(m K) |

*Table 2 Fluid properties of isopentane (R601a). The fluid properties are taken from REFPROP [8]*

# 2 Assessment of the results

## 2.1 Heat pump breadboard

In this project, a heat pump breadboard with three novel electrically-driven high-speed turbo compressors in a serial configuration has been designed and built. The refrigerant for the breadboard is isopentane (R601a). Fig. 3 shows a schematic layout of the heat pump breadboard. Isopentane enters the first compressor as a slightly superheated (to 50°C) vapour. Three compressors are used in a serial configuration to increase the saturation temperature of the fluid to the desired value (100°C) [1]. Before and after each compressor, the pressure and temperature is measured, so that the isentropic efficiency of each compressor can be calculated, as well as the efficiency of the total compressor system. After the compressors, vapour travels through the condenser, where the vapour is condensed into liquid and the heat that is stored in the vapour is released. The temperature of the condenser is controlled with a thermostat bath and cooling water. The liquid then flows into the receiver at the top, and out of the receiver at the bottom. After the receiver, the liquid flows through the expansion valve, where the pressure abruptly decreases, causing a partial evaporation of the liquid (to a vapour mass fraction of 0.36, see Fig. 2) and a drop in the temperature. The cold liquid-vapour mixture then flows through the evaporator where it absorbs heat and completely turns into vapour before entering the compressor.

A CAD drawing of assembled system is shown in Fig. 4 and a photo is shown in Fig. 5. A detailed description of the breadboard including a list of all the components is provided in [2].

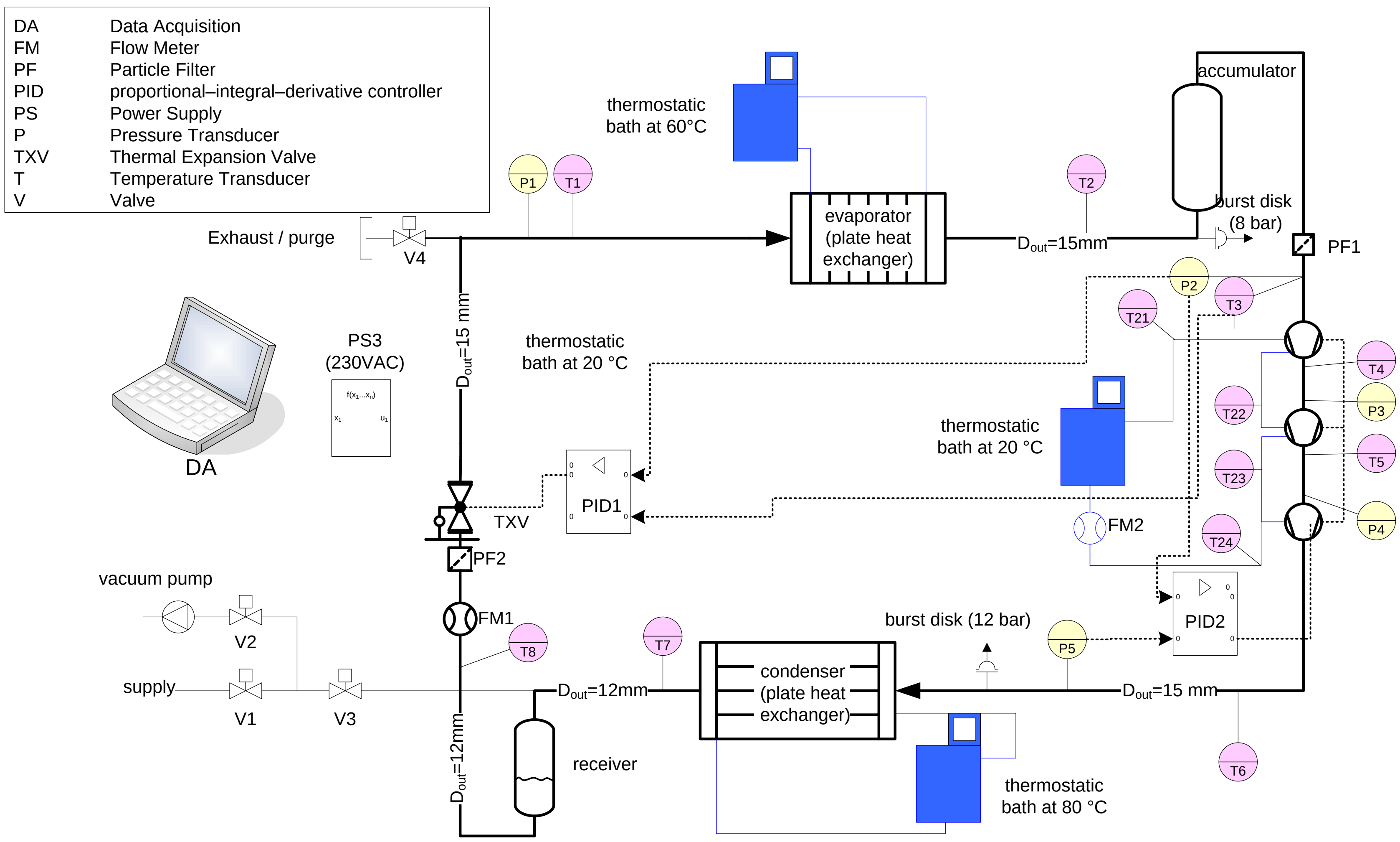


*Fig. 3 Schematic drawing of the heat pump breadboard*

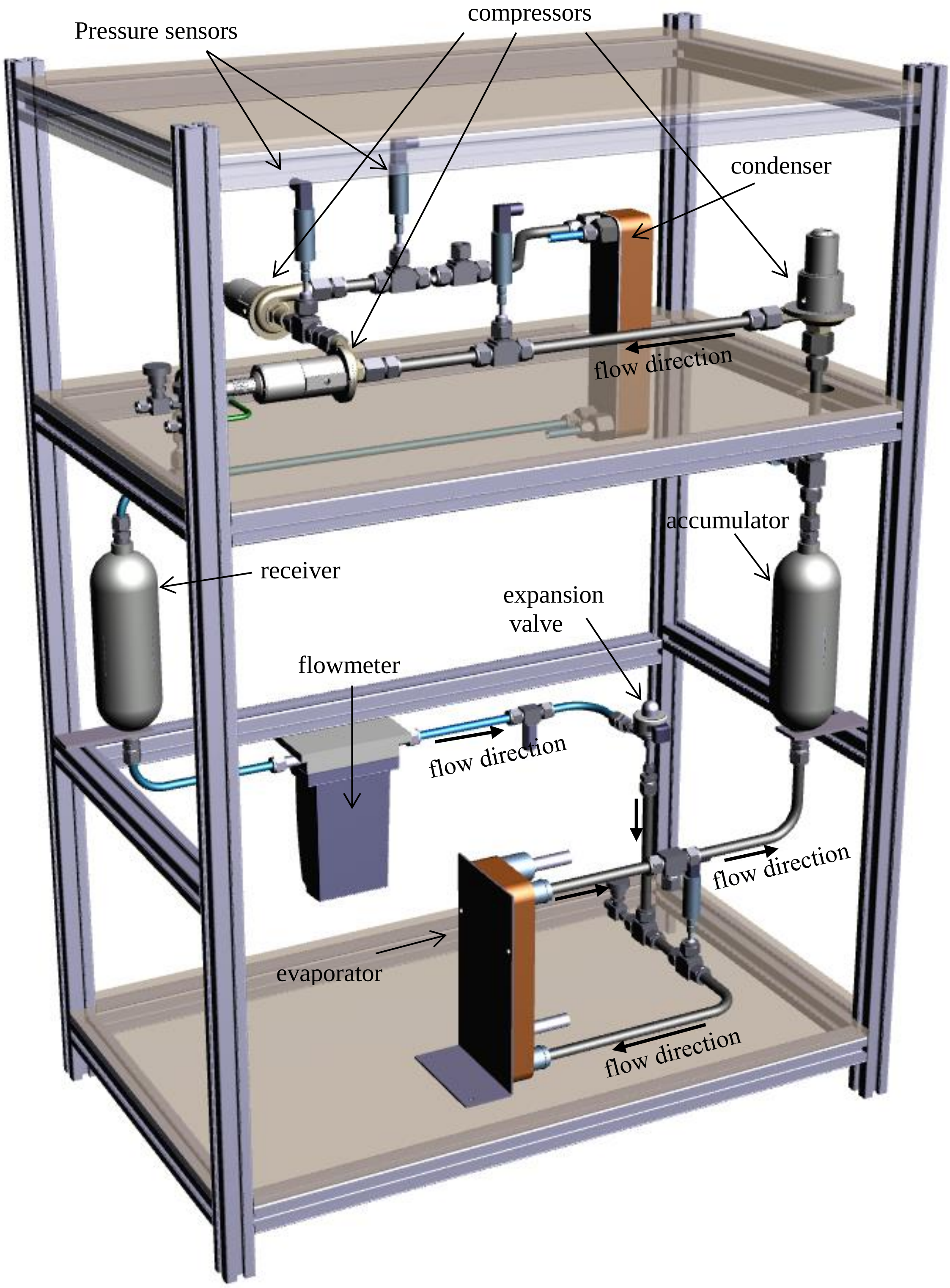


*Fig. 4 CAD drawing of the assembled system.*

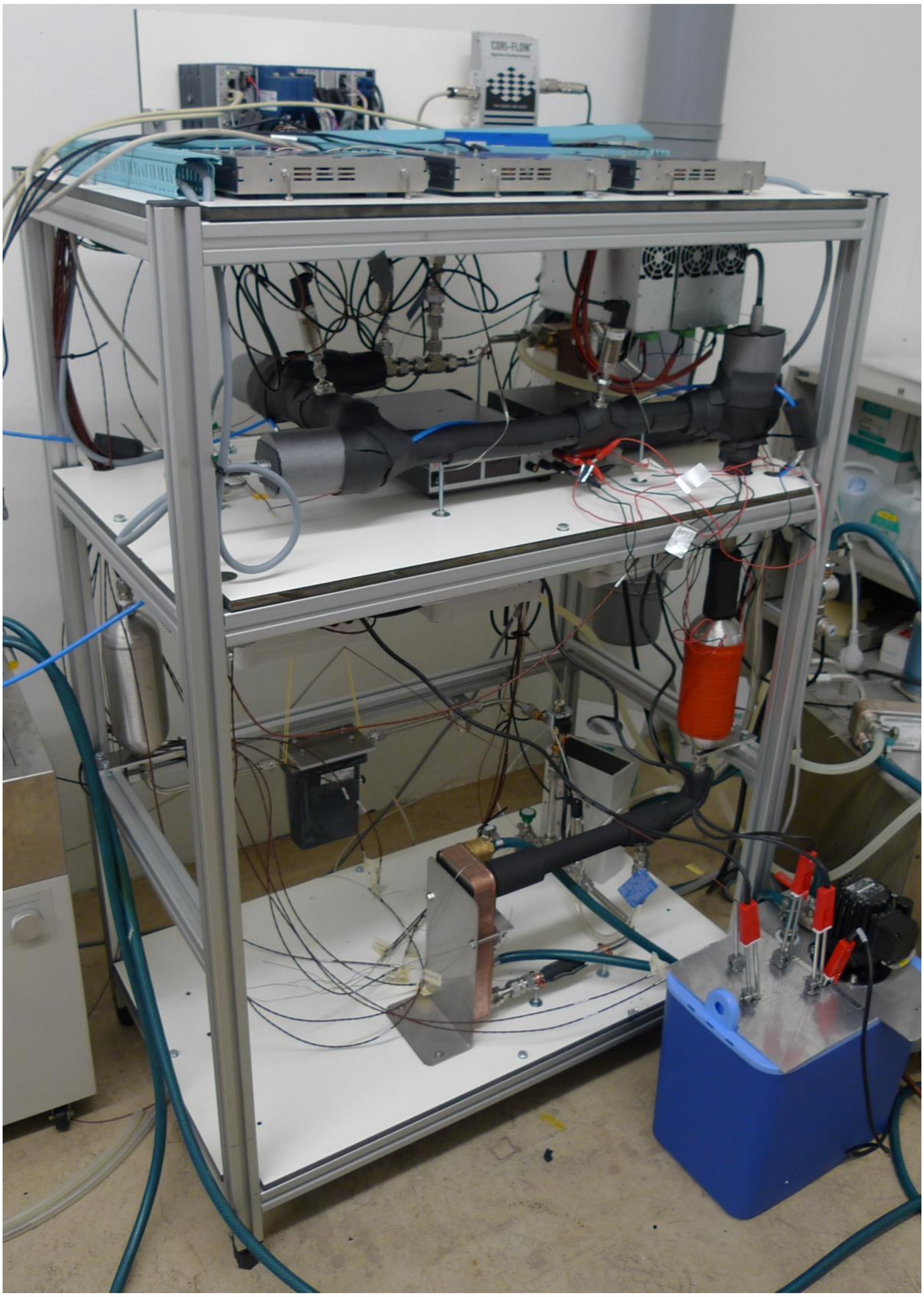


*Fig. 5 Photo of the breadboard.*

## 2.2 Heat pump compressors

### 2.2.1 *General design*

Three electrical-driven turbo compressors made by Celeroton are used in a serial configuration. For the first stage, an (slightly modified) existing CT-17-700 turbo compressor with a 3D impeller (see Fig. 6) has been used. For the second and third stage, a new turbo compressor with a 2D impeller (see Fig. 7) has been designed. The compressors run approximately at 180 kRPM. The design and manufacturing of the compressor is described more extensively in [6-8].

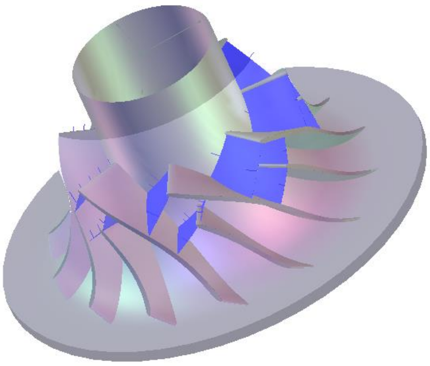

*Fig. 6: Drawing of the 3D-impeller for the first stage*

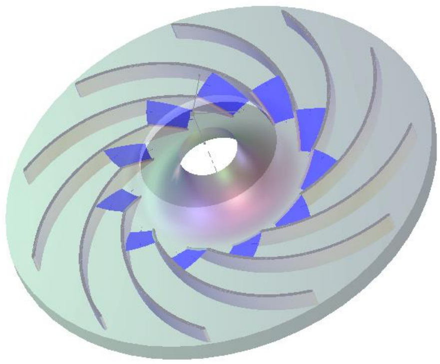

*Fig. 7: Drawing of a 2D-impeller for the second and third stage*

### 2.2.2 *Comparison with existing compressors*

In [1], a comparison has been made between commercial and aerospace compressors, and the compressors developed in this project. The results of the compressor comparison are summarized in Table 3. From this table, it can be concluded that the compressor developed in this project have:

- A mass of 2 kg, compared to >20 kg for other compressors with the same capacity
- A vibration level of <0.3 N compared to 40 N for commercial compressors
- An isentropic efficiency of 68%, which is higher than aerospace compressors (46%), and comparable to commercial compressors (60-70%)
- A condenser saturation temperature of 100°C, which is comparable to aerospace compressors, but higher than the maximum temperature of commercial compressors (65°C)
- No oil required for lubrication or sealing, compared to commercial compressors which require a few litres of oil.

| Compressor | Refrigerant | Mass for 10 kW cool capacity | efficiency | Max condenser temperature | Requires oil for lubrication and sealing | Estimated vibration level |
|---|---|---|---|---|---|---|
| Copeland ZB30KCE | R134a | 40 kg | 59.2% | 65°C | Yes, 1.9 litres | ~40N |
| Copeland ZB42KCE | R134a | 40 kg | 66.3% | 65°C | Yes, 1.9 litres | ~40N |
| Danfoss MLZ045 | R22 | 37 kg | 67.8%. | 45°C | Yes, 1.6 litres | ~40N |
| Danfoss MLZ045 | R404a | 37 kg | 70.3% | 45°C | Yes, 1.6 litres | ~40N |
| Danfoss The MLZ076 | R134a | 45 kg | 59.6% | 45°C | Yes, 2.7 litres | ~40N |
| Fairchild 54 mm Twin Screw | ? | 19 kg | <50%?? | 93.3°C | Yes | ? |
| Sundstrand LANTIRN compressor | R114 | 10 kg for 2.5 kW | Max 45.7% | 110°C | Yes | >> 0.3N |
| Honeywell F-22 compressor | ? | ? | ? | ? | ? | ? |
| Compressor developed in this project | Isopentane (R601a) | ~2 kg | 68% | 100°C | No | < 0.3N |

*Table 3 Comparison table*

## 2.3 Measured COP and compressor efficiency

Fig. 8 shows the measured heat pump cycle in a pressure-enthalpy diagram. At the target setting (saturation temperature of 45°C at the evaporator, 100°C at the condenser, and a ‘payload’ heat input of 5 kW), the measured COP is 2.6, which is considerably higher than the requirement of 2. This is because the pressure drop in the evaporator is lower (0.2 bar instead of 0.4 bar), and because the efficiency of the compressors is higher than expected (0.68 instead of 0.6). The measured efficiency of 0.68 of the compressors is considerable higher than existing aerospace compressors, and even higher than the efficiency of most of the commercially available scroll compressors that were analysed in the compressor comparison (see previous section).

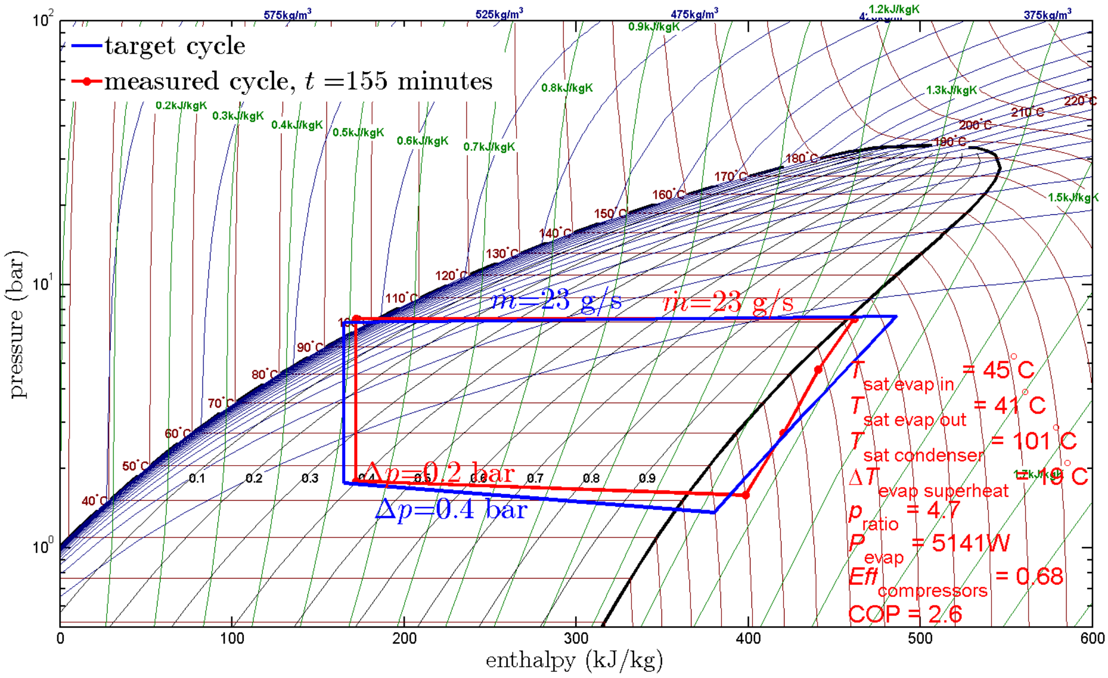


*Fig. 8 Measured (red) and target (blue) heat pump cycle in a pressure-enthalpy diagram. The measured cycle has a COP of 2.6, which is higher than the requirement of 2*

### 2.4 Noise in the Stage 3 compressor and subsequent investigation results

During a measurement, a loud noise was observed. Furthermore, the efficiency of the stage 3 compressor became lower than in previous measurements. For this reason, it was concluded that the stage 3 compressors could not be used anymore, and it was send back to Celeroton for inspection. The analysis at Celeroton showed that large particles are stuck in the diffusor (see Fig. 9 and Fig. 10 for a detail picture of the particles that were stuck in the diffusor). This resulted in blockage of some flow channels, which led to a surge at higher mass flows which probable resulted in the observed noise and vibrations. Also it is obvious that this resulted in a lower efficiency. The most likely origin of the particles are the Viton sealing rings that are used to seal the inlet and outlet connections of the compressors. In a subsequent project, a different sealing method for the inlet and outlet connections must be used. This is a relative simple modification.

*Fig. 9 Photo of the rotor and diffusor of the stage 3 compressor, where it can be seen that large particles are stuck in the diffusor*

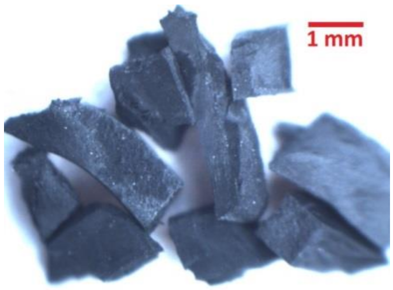


*Fig. 10 Detail picture of the particles that were stuck in the diffusor*

## 2.5 Summary for the test program

Because of the blockage of the stage 3 compressor with particles from the seals of the compressor (see previous section), only 5 out 8 items of the test plan have been carried out. The maximum cooling capacity test, short life time test, and the increase in cooling water temperature test have not been carried out. For the maximum cooling capacity test, it is assumed that the cooling capacity of 5 kW could be increased somewhat, because the cooling capacity of 5 kW was achieved with 175 kRPM, while the maximum RPM of the compressors is set to 200 kRPM. The compressor cooling water temperature used in the test was 25°C. The motor temperature is 80°C. Since the motors are allowed to have a temperature of 100°C, it is expected that the thermostat water temperature could be increased to 45°C. A summary of the test program is shown in Table 4.

| Test description | Test results | remark |
|---|---|---|
| Proof pressure test | successful | After modification of the compressors |
| Helium leak test | successful | The system has been helium leak tested 5 times |
| Filling of the breadboard with isopentane | successful | The system has been filled with isopentane 5 times |
| Start-up test | successful | The system has been started-up around 25 times. After some modifications to the compressor and condenser settings, the start-up is straight forward |
| COP test | successful, 2.6 measured | The measured COP of 2.6, is significantly higher than the requirement of 2 |
| Max cooling capacity test | not carried out | It is expected that the cooling capacity can be increased above 5 kW |
| Short life time test | not carried out | |
| Increase in cooling water temperature test | not carried out | It is expected that the cooling water temperature can be increased to 45°C |

*Table 4 Summary table for the tests that has been carried out*

# 3 Recommendations for future work

## 3.1 Space qualify compressor electronics

The converters that drive the compressors are not space qualified, not optimized for smallest mass and size, and not adapted to the final dc voltage bus of spacecraft. With a specific electronics design, all these drawbacks could be resolved. Celeroton directly has no experience in designing space qualified electronics, however the board of directors has contacts with companies which themselves design space qualified electronics and offer consulting in space qualified electronics. Furthermore, NLR has experience in designing and qualifying electronics for space applications.

## 3.2 Increasing the life time of the compressors

The compressor for this project phase have been equipped with ball bearings, as the goal of this project was to prove the thermodynamic feasibility. Therefore, the lifetime is not yet sufficient for space missions. Therefore, one main proposed future task is the replacement of the ball bearings with a fluid or magnetic bearing, which are both contactless, and therefore show no mechanical wear and a long lifetime. There are several technical challenges to solve such as fixation or levitation during rocket start (vibrations).
Since it was observed during the test phase of the heat pump breadboard that particles from the inlet and outlet connection sealing got stuck in the diffusor of a compressor, it is necessary to redesign the inlet and outlet sealing of the compressors. However, this is relative simple.

## 3.3 Cooling of the compressors with its own fluid

In the current breadboard, the motors of the compressors are cooled with water from a thermostat bath. It is proposed to cool the motors with isopentane from the heat pump. For this, in a first step the water of the thermostat bath can be increased to 45°C, and in a second step the cooling loop can be changed to isopentane. The best solution is to use ispentane coming from the expansion valve, since this fluid has a low temperature (45°C) and the pressure drop induced by the motor cooling circuit has no negative influence on the performance of the heat pump.

## 3.4 Using passive expansion valves instead of electronic expansion valves

The expansion valve regulates the superheat (e.g. 5°C) after the evaporator: If the superheat is too large, the valve opens more which results in a larger massflow and thus smaller superheat. Vice versa, when the superheat is too small, the valve closes more which results in more superheat. There are two basic types of expansion valves: Electronic expansion valves and thermostatic expansion valves. In the current breadboard, an electronic expansion valve is used

to control the superheat temperature. In an electric expansion valve, the valve opening is adjusted by an electric actuator via a PID controller that regulates the superheat temperature to the desired value. An electronic expansion valve can therefore be used for a wide range of fluids. However, the electronics required for an electronic expansion valve are undesired for a space application.
A thermostatic expansion valve regulates this superheat by use of a temperature sensing bulb filled with the same fluid as in the system. Thermal expansion valves are not available for every fluid, because the sensing bulb must be filled with the same (or similar) fluid as in the system. For the most common refrigerants (R22, R134a, R407c, R404a, ammonia), these valves are easily available, but there are no thermostatic expansion valves available for isopentane. Danfoss also makes thermostatic expansion valves for hydrocarbons, but not for isopentane. However, it is fairly easy to adapt an existing thermal expansion valve to isopentane and for this reason, it is proposed to use a thermostatic expansion valve in a future breadboard.

## 3.5 Designing a receiver that works in zero gravity

The receiver in the breadboard depends on gravity to ensure that only liquid is always present at the exit of the receiver. It is proposed to design a receiver that also functions in zero gravity (and preferably in all orientations), for example by using centrifugal forces.

## 3.6 Increasing the COP of the system

There are several methods for increasing the COP of a heat pump. These are discussed in the next sections.

### 3.6.1 *Increasing the compressor efficiency*

An obvious method to increase the COP of a system is by increasing the efficiency of the compressors. The measured efficiency of 0.68 is already very high, but there are measures that could allow for an even further increase of the efficiency.
Compressor stage 2 and 3 are realized as pure 2D impellers. These types of Impellers show a high influence of the tip clearance on the pressure ratio and efficiency. This is known from theory and could also be shown during first tests in air by adjusting the tip clearance. For such miniature, ultra-high-speed compressors, the tip clearance has a lower limit due to tolerances and vibrations. A shroud could lower the influence in the efficiency drop and in the required tolerances. Therefore, it is proposed to include also shrouded impellers in future analysis and redesigns.

### 3.6.2 *Reduce the pressure drop in the system by using parallel evaporators*

The pressure drop in the system, and especially the pressure drop in the evaporator, has a very large influence on the COP of the system. This is illustrated by Fig. 11, which shows the calculated COP of the heat pump, as a function of the pressure drop over the evaporator and condenser. In this calculation, the efficiency of the compressors is assumed to be 68%.

In the experiments, the pressure drop is 0.2 bar, and the COP is 2.6. However, in an actual application, the evaporator can exist of long tubing, and the pressure drop can become much higher. There are several methods to reduce the pressure drop in a system. The most straightforward method is to increase the diameter of the tubing. However, this also increases the volume of the system, which is undesired. Another method is to use parallel evaporator sections. With parallel channels, the diameter of the tubing can remain small, and it is therefore proposed to test parallel evaporators in a future breadboard.

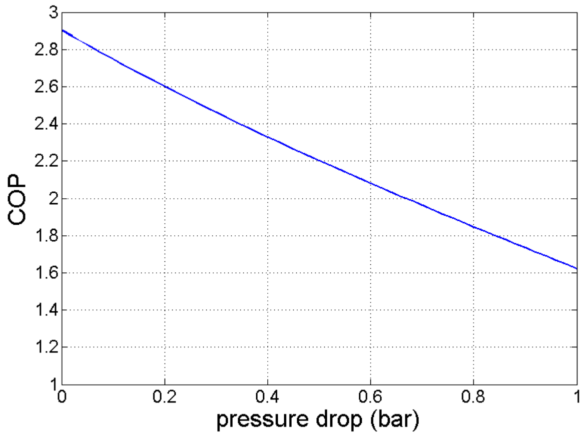


*Fig. 11 COP of the system as a function of the pressure drop over the evaporator and condenser. The efficiency of the compressors is assumed to be 68%.*

### 3.6.3 *Use additional heat exchangers and/or valves to increase the COP*

For a ‘wet’ refrigerant like isopentane, the two-phase dome is ‘tilted’ to the right (see Fig. 2). As a result, the liquid/vapour mixture that enters the evaporator has a relative high vapour mass fraction of almost 0.4. This means that a large portion of the latent heat of the fluid is not used in the evaporator. If the vapour mass fraction of the liquid/vapour mixture that enters the evaporator is reduced to for example 0.1, a much larger portion of the latent heat of the fluid is used in the evaporator, which reduces the massflow in the system and therefor increases the COP of the cycle. There several methods to achieve this:

- Using a Liquid-Suction Heat Exchanger (see Appendix A)
- Using Flash Economizers (Appendix B)
- Using Heat Exchanger Economizers (Appendix C)
- Using three cascaded cycles (Appendix D)

In Appendix A to D, these methods are analysed for the heat pump cycle with isopentane. With all four methods, the COP of the system can be increased by 17 to 21%. A Liquid-Suction Heat

Exchanger (LSHE) is the most simple method, and requires just one additional heat exchanger in the system. However, in practice there will be a pressure drop over the heat exchanger (this pressure drop is not included in the analysis, since it depends on the design of the heat exchanger), and this pressure drop can have a negative influence on the COP of the system (see Fig. 11). A system with Heat Exchange Economizers is more complex, but the vapour massflow through the heat exchangers is very small (just 14% of the total massflow) and as a result, the pressure drop over the heat exchanger will be very small.
The method with Flash Economizers requires two additional expansion valves and two separation vessels, which separate the liquid from the vapour. For a space application, centrifugal forces in the vessel can be used to separate the vapour from the liquid, and this could be a very lightweight solution.
It is proposed to make a trade-off between the different methods to increase the COP by adding heat exchangers and/or valves

## 3.7 Use a Liquid-Suction Heat Exchanger (LSHE) to obtain the required superheat for the compressors

It was observed in the measurements that due to the high efficiency of the compressors, the required superheat of the vapour that enters the compressors has to be rather larger. This superheat is inherent provided when a Liquid-Suction Heat Exchanger (see Appendix A) is used. An additional advantage of a LSHE is that it increases the COP of the heat pump. A LSHE does not provide superheating for the compressors during start-up, so this is an issue that has to be addressed.

## 3.8 Make a full size breadboard

In the current breadboard, the evaporator and condenser that are used are not representative for a space application concerning volume, tubing length etc. This can have influences on the time dependent behaviour (e.g. during start-up) of the system. It is therefore recommended to make a full size breadboard.

## 3.9 Perform environmental (e.g. vibration and thermal cycling) tests

For a space application, a heat pump is subjected to severe vibrations (during launch) and temperature variation. It is there recommended to perform relevant environmental tests on the complete system or in an earlier phase on the most critical components (e.g. the compressors, electronics, and expansion valve).

# 4 Conclusions

In this project, a heat pump breadboard with three novel electrically-driven high-speed turbo compressors in a serial configuration has been designed and built. The refrigerant for the breadboard is isopentane (R601a). The breadboard demonstrates that the used heat pump method is very efficient: At the target setting (saturation temperature of 45°C at the evaporator, 100°C at the condenser, and a 'payload' heat input of 5 kW), the measured COP is 2.6, which is considerably higher than the requirement of 2. The compressors developed in this project have a much lower mass and higher efficiency than existing (commercial or aerospace) compressors.

The test program could not be finished completely, because particles from the sealing of the compressor got stuck in its diffusor

The recommendations for future work include space qualifying the compressor electronics, increasing the lifetime of the compressor, a method to cool the compressor with its own fluid, and designing a receiver that works in zero gravity.

## References


[1] H.J. van Gerner, *Heat Pump Conceptual Study and Design: Preliminary Design Report*, NLR-CR-2012-177 (2012)

[2] H.J. van Gerner, *Heat Pump Conceptual Study and Design: Detailed Design Report*, NLR-CR-2012-424 (2013)

[3] H.J. van Gerner, A. Pauw, G. van Donk, *Heat Pump Conceptual Study and Design: Test Readiness Report*, NLR-CR-2013-327 (2013)

[4] H.J. van Gerner, A. Pauw, G. van Donk, *Heat Pump Conceptual Study and Design: Breadboard Test Report*, NLR-CR-2014-007 (2014)

[5] Celeroton AG, *Heat Pump Conceptual Study and Design: WP 4100 Turbocompressor Preliminary Design* (2012)

[6] Celeroton AG, *Heat Pump Conceptual Study and Design: WP 4200 Compressor Detailed Design*, PR-6901-001 (2013)

[7] Celeroton AG, *Heat Pump Conceptual Study and Design: WP 4300 Turbocompressor Manufacturing*, PR-6901-002 (2013)

[8] Lemmon, E.W., Huber, M.L., McLinden, M.O. *NIST Standard Reference Database 23: Reference Fluid Thermodynamic and Transport Properties-REFPROP, Version 8.0*, National Institute of Standards and Technology, Standard Reference Data Program, Gaithersburg, 2007.


## Applicable documents


[AD1] Statement of Work for ESA Invitation To Tender AO/1-6751/11/NL/EM

[AD2] NLR proposal SID6393, *Conceptual heat pump study and design*, *issue 2*, dated January 24th 2012.

[AD3] H.J. van Gerner, *Heat Pump Conceptual Study and Design: Preliminary Design Report*, NLR-CR-2012-177 (2012)

[AD4] H.J. van Gerner, *Heat Pump Conceptual Study and Design: Detailed Design Report*, NLR-CR-2012-424 (2013)

[AD5] H.J. van Gerner, A. Pauw, G. van Donk, *Heat Pump Conceptual Study and Design: Test Readiness Report*, NLR-CR-2013-327 (2013)

[AD6] Celeroton AG, *Heat Pump Conceptual Study and Design: WP 4100 Turbocompressor Preliminary Design* (2012)

[AD7] Celeroton AG, *Heat Pump Conceptual Study and Design: WP 4200 Compressor Detailed Design*, PR-6901-001 (2013)

[AD8] Celeroton AG, *Heat Pump Conceptual Study and Design: WP 4300 Turbocompressor Manufacturing*, PR-6901-002 (2013)

# Appendix A Increasing the COP with a Liquid-Suction Heat Exchanger

For a ‘wet’ refrigerant like isopentane, the two-phase dome is ‘tilted’ to the right. As a result, the liquid/vapour mixture that enters the evaporator has a relative high vapour mass fraction of almost 0.4. This means that a large portion of the latent heat of the fluid is not used in the evaporator. When the liquid coming out of the condenser is subcooled to 60°C, the vapour mass fraction of the liquid/vapour mixture that enters the evaporator is reduced to 0.1 (see Fig. 13). As a result, a lower mass flow (and therefor lower compressor power) is required to achieve the same cooling capacity, and this increases the COP. A Liquid-Suction Heat Exchanger (LSHE) uses the cold vapour coming from the evaporator to cool the hot liquid from the condenser (see Fig. 12 and Fig. 13). As a result, the vapour that enters the compressor is at a higher temperature (92°C for the cycle in Fig. 13). The exit temperature of the compressor therefore also increases from 115°C without LSHE to 156°C with LSHE. The COP is increased from 2.04 without LSHE, to 2.41 with LSHE (i.e. an 18% increase in the COP). The drawbacks of a LSHE are the additional mass of the heat exchanger, and the higher compressor exit temperature.

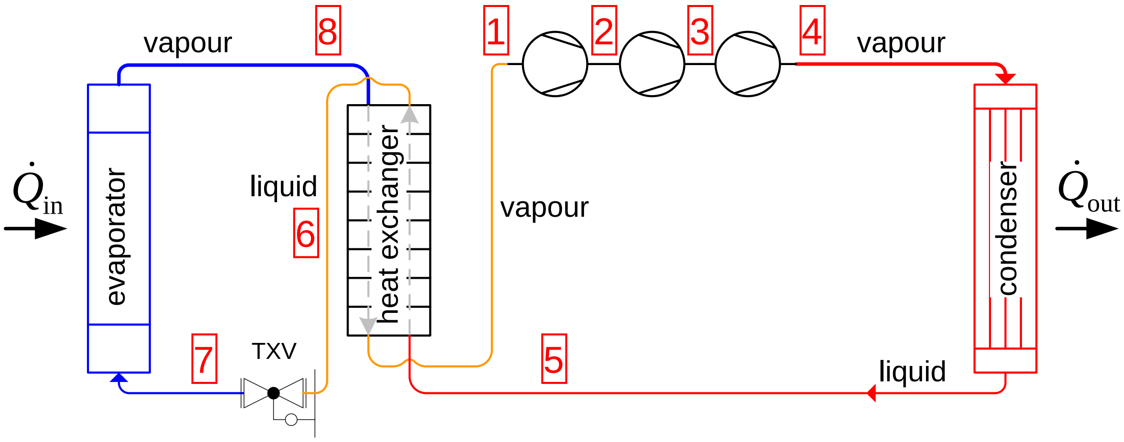


*Fig. 12 Schematic drawing of a vapour compression cycle with a Liquid Suction Heat Exchanger*

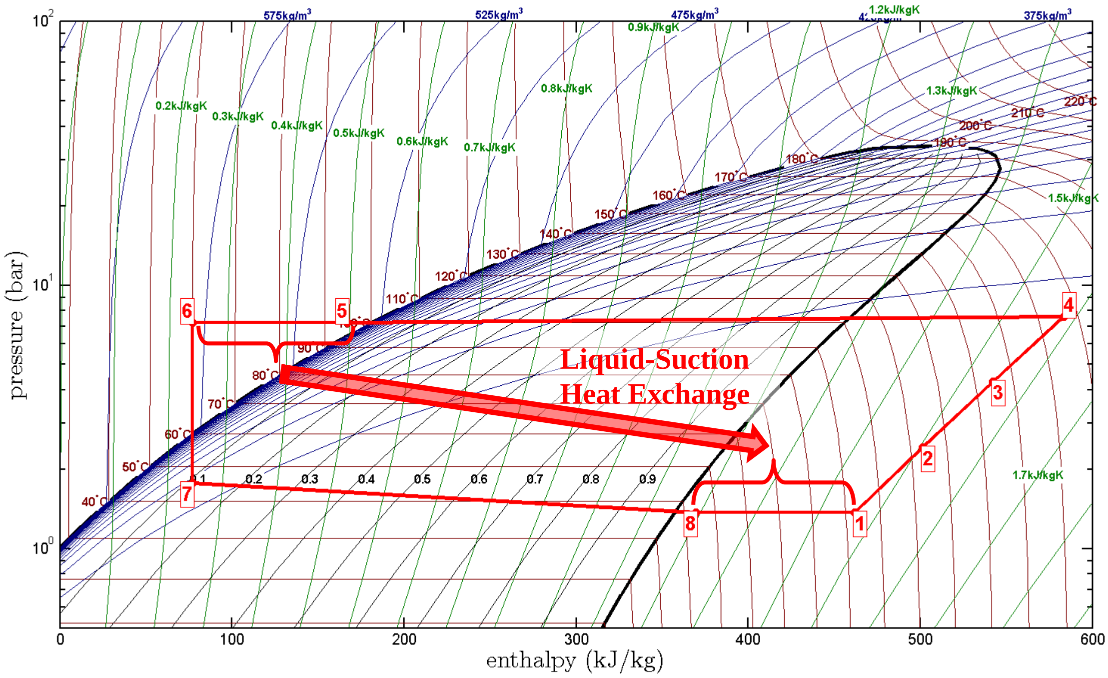


*Fig. 13 Vapour compression cycle with a Liquid-Suction Heat Exchanger. The refrigerant is isopentane*

## Appendix B Increasing the COP with flash economizers

For a ‘wet’ refrigerant like isopentane, the two-phase dome is ‘tilted’ to the right. As a result, the liquid/vapour mixture that enters the evaporator has a relative high vapour mass fraction of almost 0.4. This means that a large portion of the latent heat of the fluid is not used in the evaporator. The vapour mass fraction of the fluid that enters the evaporator can be reduced by not expanding the fluid with one valve, but by using multiple valves (see Fig. 14). After the first expansion valve, the mixed vapour/liquid flow is separated into vapour and liquid. For terrestrial applications, this is usually achieved in a tank due to gravity. However, for a space application, centrifugal forces can be used to separate the vapour from the liquid. The vapour (which is 14% of the total mass flow for the cycle depicted in Fig. 15) is then injected between the second and third compressor stages. The liquid (86% of the total mass flow) is directed to a second expansion valve. After the second expansion valve, the mixed vapour/liquid flow is again separated into vapour and liquid, and the vapour is injected between the thirst and second compressor stage. The liquid is directed to the third expansion valve. The vapour mass fraction of the liquid/vapour mixture that enters the evaporator is reduced to 0.1 (compared to 0.4 for a basic cycle, see Fig. 2). This means that a much larger portion of the latent heat of the fluid is used in the evaporator, which increases the COP of the cycle. For the cycle with isopentane, the COP is increased from 2.03 without flash economizers, to 2.46 with flash economizers (i.e. a 21% increase in the COP).

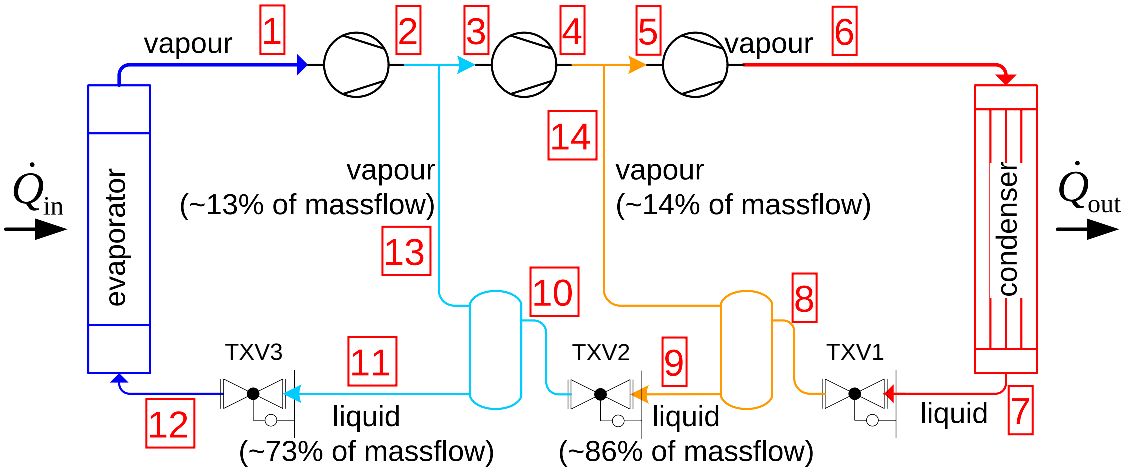


*Fig. 14 Schematic drawing of a vapour compression cycle with a Flash Economizer*

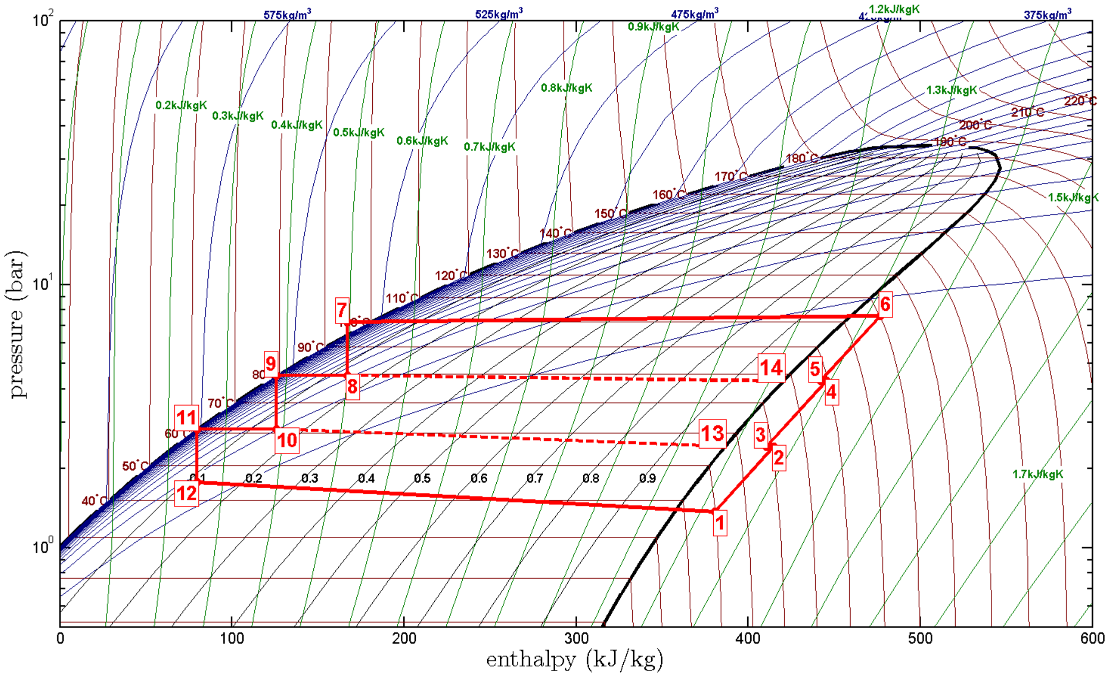

*Fig. 15 Vapour compression cycle with a Flash Economizers.*

## Appendix C Increasing the COP with heat exchanger economizers

For a 'wet' refrigerant like isopentane, the two-phase dome is 'tilted' to the right. As a result, the liquid/vapour mixture that enters the evaporator has a relative high vapour mass fraction of almost 0.4. This means that a large portion of the latent heat of the fluid is not used in the evaporator. The vapour mass fraction of the fluid that enters the evaporator can be reduced by not expanding the fluid with one valve, but by using multiple valves and heat exchangers (see Fig. 16). Before the first expansion valve, the liquid flow is divided into a main flow (86% of the massflow for the cycle depicted in Fig. 17) and a secondary flow (14% of the massflow). In a heat exchanger, the main flow is cooled by the secondary flow. When the secondary flow absorbs heat from the primary flow, it evaporates. The secondary vapour flow is then injected between the second and third compressor stage. Before the second valve, the flow is again separated, and the main flow is further cooled to 62°C before it enters the third expansions valve. As a result of the stepwise cooling of the main flow, the vapour mass fraction of the liquid/vapour mixture that enters the evaporator is reduced to 0.1 (compared to 0.4 for a basic cycle, see Fig. 2). This means that a much larger portion of the latent heat of the fluid is used in the evaporator, which increases the COP of the cycle. For the cycle with isopentane, the COP is increased from 2.03 without heat exchange economizers, to 2.39 with heat exchange economizers (i.e. a 17% increase in the COP). Because the vapour flow through the heat exchangers is very small (~14% of the total flow), the pressure drop over the heat exchanger will be very small.

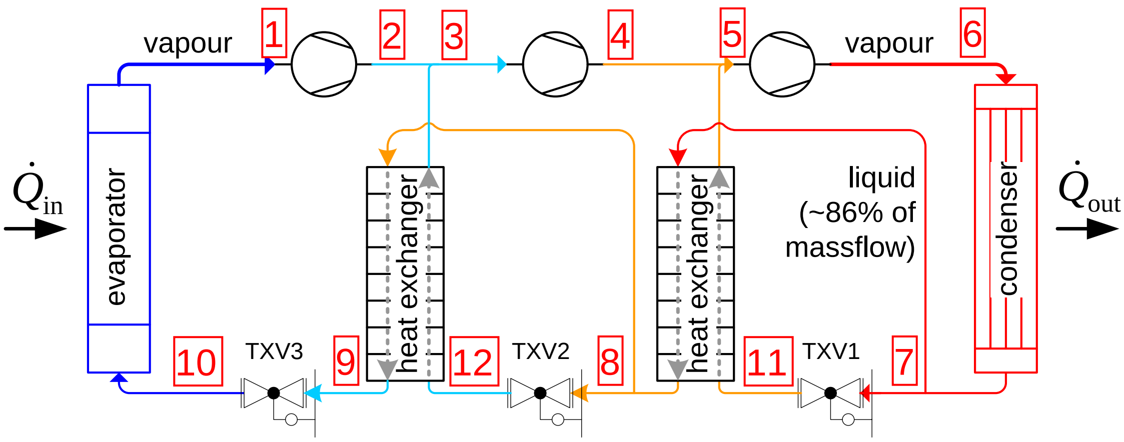


*Fig. 16 Schematic drawing of a vapour compression cycle with a Heat Exchanger Economizer*

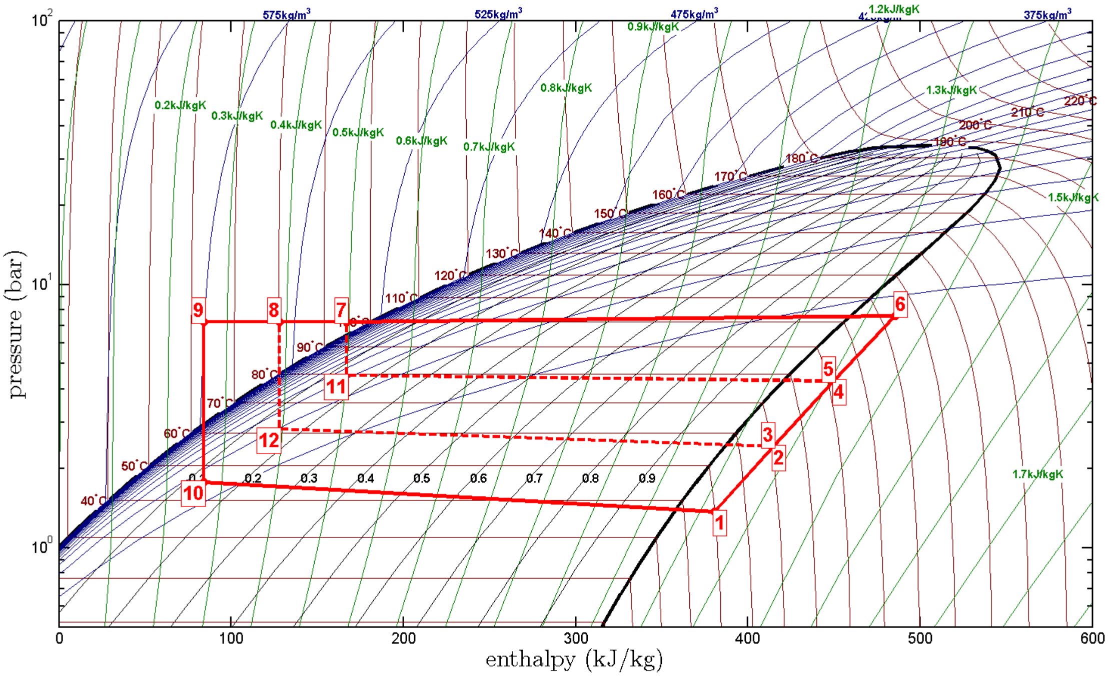


*Fig. 17 Vapour compression cycle with a Heat Exchanger Economizer.*

## Appendix D Increasing the COP with cascades

The heat pump cycle can be separated in three cycles (see Fig. 18). Cycle b is thermally connected to cycle a, and cycle b is thermally connected with cycle c. With three cascades, the vapour mass fraction of the liquid/vapour mixture that enters the evaporator is reduced to 0.1 (compared to 0.4 for a basic cycle, see Fig. 2). This means that a much larger portion of the latent heat of the fluid is used in the evaporator, which increases the COP of the cycle. For the cycle with isopentane (see Fig. 19), the COP is increased from 2.03 with one cycle, to 2.47 with 3 cascaded cycles (i.e. a 21% increase in the COP). In Fig. 19, isopentane is used in all three cycles. However, it is also possible to use different refrigerants for each cycle.

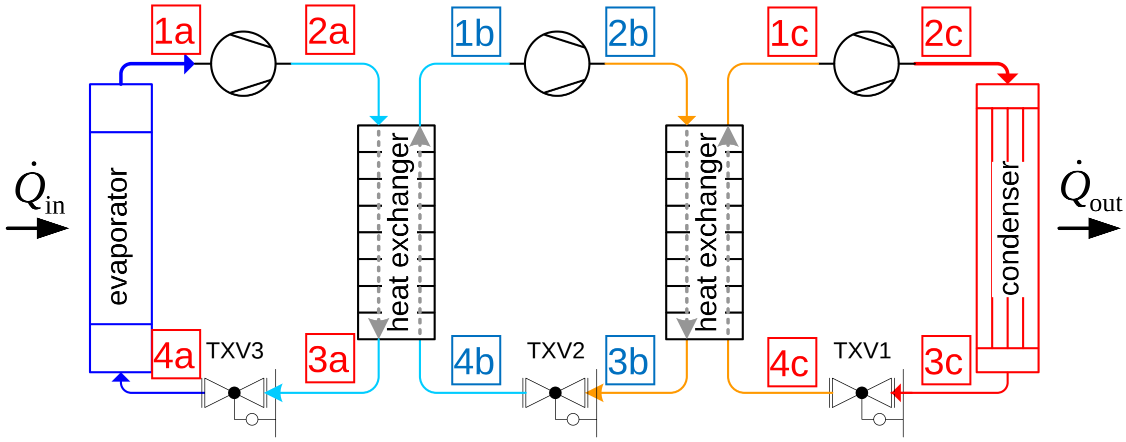


*Fig. 18 Schematic drawing of a vapour compression cycle in three cascades*

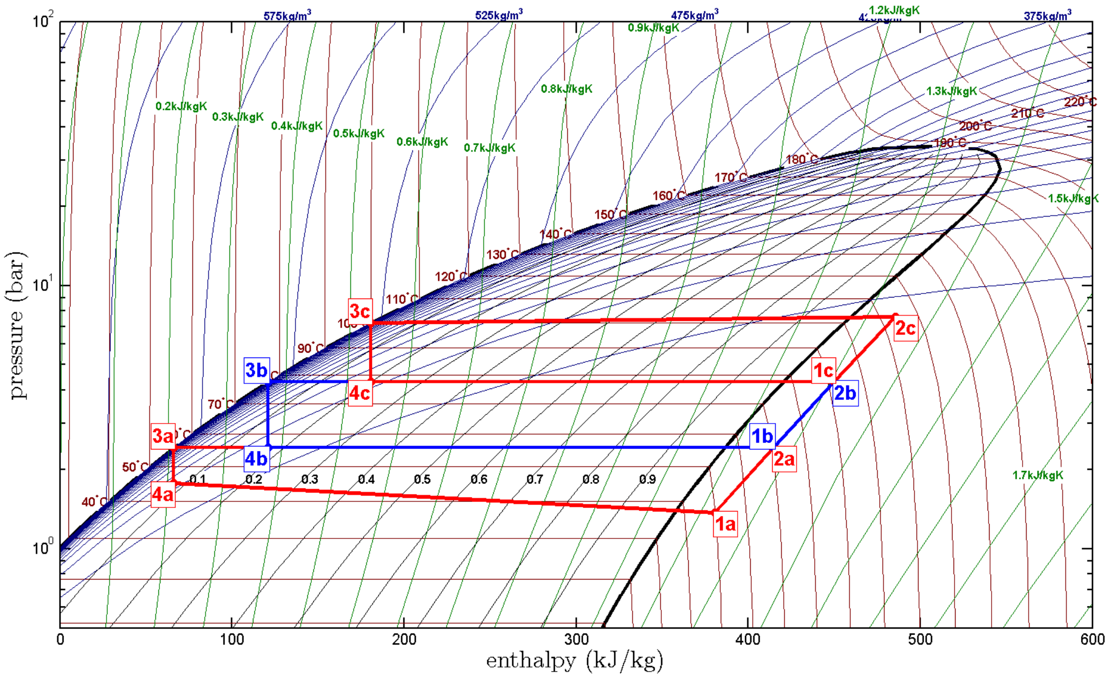


*Fig. 19 Vapour compression cycle in three cascades.*