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COP tests of a Rotation Heat Pump

Andreas LÄNGAUER[(a)], Bernhard ADLER[(a)], Christian RAKUSCH[(a)], Karl PONWEISER[(b)]

[(a)] ecop Technologies GmbH
Vienna, 1230, Austria, office@ecop.at
[(b)] TU Wien
Vienna, 1040, Austria, karl.ponweiser@tuwien.ac.at

ABSTRACT

A completely new way of using an ecologically friendly, not flammable and nontoxic working gas for heat pump applications is realized in the Rotation Heat Pump (RHP) by using a Joule process instead of a Rankine Process. Substantiated by a working fluid which is always gaseous, it is possible to obtain a sensible heat transfer at sink and source.
The main problem to accomplish an efficient Joule process in heat pumps, the compressing of the gas, is based on a centrifugal potential which allows a compression efficiency of more than 99%.
Another major advantage of the Joule process within the RHP is the possibility to change the temperature range from -20 up to 150 °C without affecting the coefficient of performance (COP) considerable. The COP is mainly determined by the temperature spread between sink and source.
Latest tests on the first industrial RHP validate the function and possibilities of this new technology.
Keywords: Rotation Heat Pump, High Temperature, COP, Energy Efficiency, Centrifugal Potential.

1. INTRODUCTION

The main problems of high temperature heat pumps are related to the lubrication of the compressor combined with the aim to use nontoxic, not flammable and low global warming potential (GWP) working fluids. Considering these facts, the Rotation Heat Pump, which is based on a Joule process has been developed and is built now. A compression based on a centrifugal potential and a noble gas as working fluid, eliminate the named problems. Some general calculations and technical details should help to understand the function before the results of tests are shown. The setup of the test rig, including several points for measuring essential values will be described as well. A lookout for further concepts and possibilities of applications including different processes will finally be given.

2. MAIN PRINCIPLE, DESIGN AND CALCULATION

The basic design of a Rotation Heat Pump consists of a rotor on which the heat exchangers and pipes are mounted and a housing which encloses the rotor. The rotating domain is filled with an ecologically friendly working fluid, consisting of different noble gases in the gaseous system and water in the sink and source system. Further necessary components like electrical devices and pumps are mounted on the main rig. In Fig. 1 the complete design is shown while the upper case is removed (left) and the principle of the rotor including heat exchangers and pipes is represented (right).

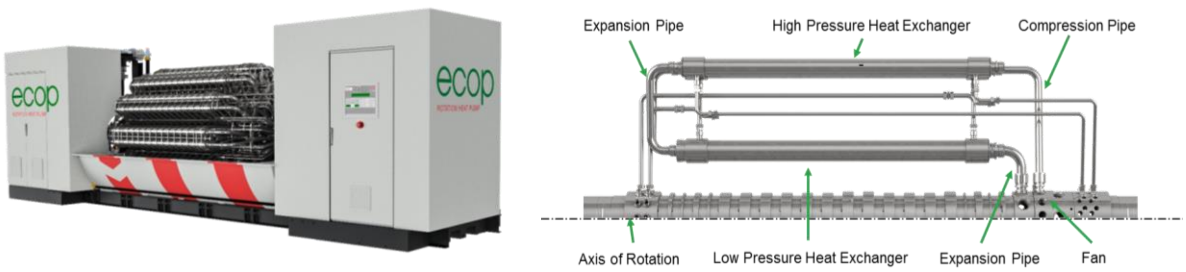


Figure 1: Design of a Rotation Heat Pump

In the following sections a short overview of the principle and function is given. In addition, the calculation of the COP of the Joule Cycle is given.

2.1. The Counter Clockwise Joule cycle in a Rotation Heat Pump

The main difference between conventional heat pumps and a Rotation Heat Pump is based on the thermodynamic cycle. The mainly used Vapor Compression Heat Pump Cycle, is replaced by the Joule Cycle. This has been described in various papers before e.g. Adler et. al (2011), Adler and Riepl (2014), Adler and Mauthner (2017). Since the process is not common used, a short description corresponding to the design is given.
In Fig. 2 the thermodynamic process is shown in a Temperature-Entropy diagram, including some characteristic points.
Starting at 1, the compression caused by the rotation of the working fluid containing system leads to increased pressure and temperature at point 2. During the process 2 to 3, thermal energy is transferred from the hot gas to the water of the sink via the high-pressure heat exchanger. The expansion from 3 to 4 is followed by the low-pressure heat exchanger, where the temperature of the gas is lower than the source temperature. Based on this temperature difference, sensible heat is transferred from the source to the gas and increasing its temperature (4-5). Going to the inner diameter from 5 to 5.1 the gas expands further going along with a temperature decrease. The final step to close the cycle is a pressure increase (5.1 to 5.2) provided by a fan. This is necessary to overcome the pressure losses due to the fluid movement and the divergence of the isobars with increasing entropy.

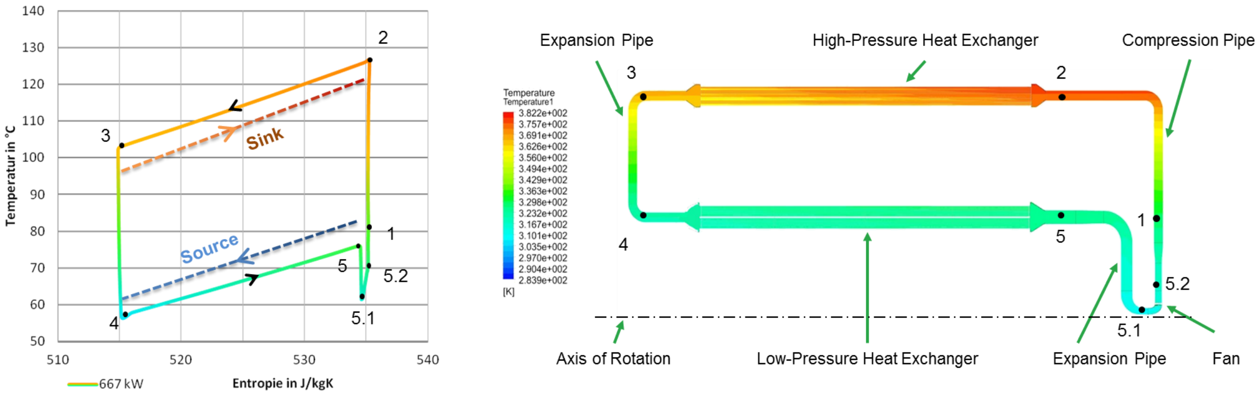


Figure 2: Joule cycle (left) and corresponding design (right) of a Rotation Heat Pump

While most applications use a compressor to increase pressure and temperature, which efficiency is essential for the COP of the system, the main compression in an RHP is based on a centrifugal potential. This is the key factor which leads to a very high efficiency of the cycle. For the main compression of the gas only frictional losses have to be compensated.
Fig. 3 shows the system including sink and source connections and flow paths.

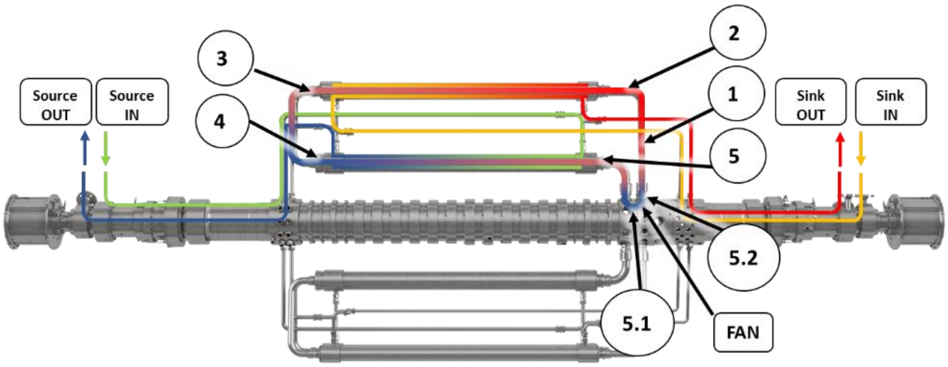


Figure 3: Rotation Heat Pump including connections and process-points

The described system correlates with the tested system. The results are shown in the chapter Testing.

2.2. Calculation of the Coefficient of Performance (COP) for a RHP

One of the most common dimensionless parameters for heat pumps is the coefficient of performance, since it represents the ratio of electrical power consumption and thermal power output. This in fact shows how efficient a heat pump is. In conventional systems the electrical power consumption is often measured at the electrical contacts of the compressor which means that the control system is not regarded. The thermal power output is based on the mass flow and increase of temperature at the heat sink in combination with the specific heat capacity.
To get a comparable value for a Rotation Heat Pump the electrical power consumption is based on the power of the fan engine plus the power of the rotor engine. The thermal power is calculated as mentioned above for conventional systems. Finally, the COP can be described as:

$$COP = \frac{\dot{Q}_{sink}}{P_{electric}} \quad \text{Eq. (1)}$$

where $\dot{Q}_{sink}$ is the thermal power at sink and $P_{electric}$ is the electrical power consumption, all given in W. Because of the fact, of having two devices for the electrical power consumption, the COP can be calculated using both as explained or just the electrical power for the fan. Using only the electrical power consumption for the fan for evaluating the COP, it can be seen as efficiency of the process itself. Therefore, it is named $COP_{Process}$ in the next sections and diagrams.

3. TESTING

In the following chapter the setup of the test system, the installed sensors, and the measurement system is described. This setup has been used for evaluating the afterwards shown results of the Rotation Heat Pump K7. The test rig is installed at the production site of the company where the RHP has been manufactured as well. It includes the RHP, heat exchangers, control valves, pumps, expansion tanks, coolers, sensors, and the whole pipe system.
Since there have been many different test scenarios, some specific and representative design points have been picked. They are shown in the following diagrams.

3.1. Setup of Test Rig, Sensors, and Measurement Procedure

For testing a Rotation Heat Pump no specific equipment compared to similar applications is necessary. Essential are for example temperature and mass flow sensors at sink and source as well as devices for measuring the electrical power consumption. The calibration of these sensors has been done before starting the tests. A specific list of relevant sensors and where they are located is given in Table 1. Additionally, Fig. 3 and Fig. 4 show the test rig and RHP K7 including sensor positions.

Figure 4: Overview test rig and RHP K7

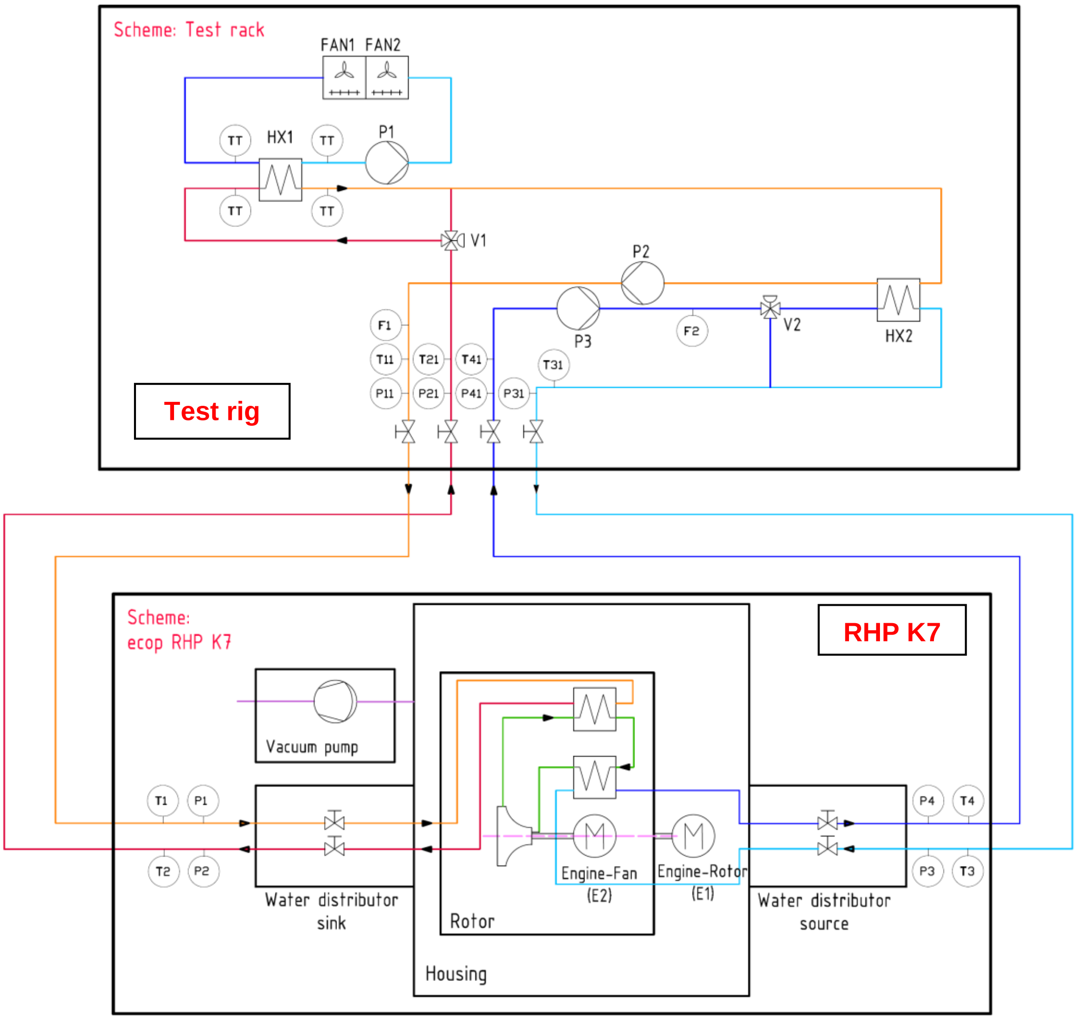


Figure 5: Simplified scheme of the test rig and RHP K7

These sensors are connected via a bus-system to the programmable logic controller where the measured values are logged in a text file to be used for further analyses and calculations afterwards. The electrical power consumption is evaluated via the frequency converters (FC) and the controller.

Table 1. Used sensors and their positions in the test system

Type of Sensor	Measurand	Position	Accuracy class
Pt 100	Temperature	Source in (T3)	A
Pt 100	Temperature	Source out (T4)	A
Pt 100	Temperature	Sink in (T1)	A
Pt 100	Temperature	Sink out (T2)	A
Magnetic-inductive	Mass flow	Source in (F2)	+/- 0,2 %
Magnetic-inductive	Mass flow	Sink in (F1)	+/- 0,2 %
Output of FC	Electrical power	Frequency converter (main rotor) (E1)	-
Output of FC	Electrical power	Frequency converter (fan) (E2)	-

The test procedure is divided in long-term-tests and short-term-tests where the first ones are considered for constant machine parameters for more than 8 hours. The short-term-tests are regarded to reach a specific design point and keep the state constant as long as it takes the system to be stable. This usually doesn’t exceed more than one hour, mostly it only takes a few minutes. The presented results are mainly from short term tests, where the duration of the test is given on the abscise in the diagrams. Due to some sensor noise, specific values like COP, thermal power and electrical power have been arithmetically averaged over 40 values (+/-20 values). The sampling rate of the logging is 5 seconds, so the averaged values display a mean over 200 s. If a value is averaged, it is mentioned in the diagram at the curve title with "_ave". Calculated values are related to the sensor signals and since there are given tolerances there is a possible deviation which is not considered and presented in this paper.

3.2. Results of Testing

The testing of the Rotation Heat Pump showed that in terms of COP it is working in at least the same range or providing a higher COP as conventional heat pumps which are using standard refrigerants. Essential intensive thermodynamic properties like temperature of sink and source are shown as well as other process parameters like mass flow, electrical power consumption and rotational speed of the rotor.

Fig. 6 shows a simulation of the Joule Cycle for the first shown test case and set of parameters, based on real gas properties provided by the National Institute of Standards and Technology (NIST) database. The temperature curves of sink and source in relation to the heat transfer are simplified but in terms of heat transfer coefficient, logarithmic temperature difference and total heat transferred based on standard functions for heat exchangers.

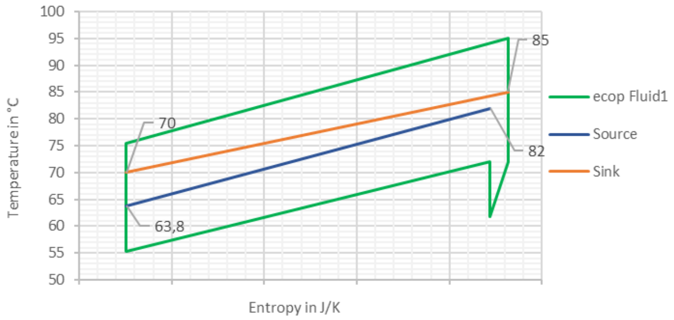
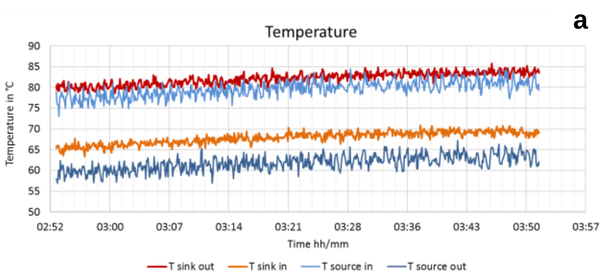


Figure 6: State diagram of the cycle, put up by calculated significant values of the process including the temperature of source and sink

The following diagrams, Fig. 7 and Fig. 8 show the results of short term tests using described formulas and measured values. This first example case illustrates the process at almost constant temperature of sink and source. The slow drift of the temperatures of around 5 K/h is caused by the heat exchanger HX1 in combination with the valve V1. Since it is a closed system and there is electrical power input, energy is finally transformed into heat which has to be transferred out of the system. The mass flow through HX1 is controlled using the valve V1 which is connected to the main control system. Because there is no feedback control and the value is set by hand, the exact value for constant temperatures couldn't be found for this test. All other values remain relatively constant. The rotational speed of the main rotor in this case was around 1200 rpm where it was possible to increase the temperature of the sink of approximately 15 K and decrease the source of around 15 K. The temperature spread between "T sink out" and "T source in" is around 3 K. This value depends basically on the rotational speed and the mass flow of sink and source as well as the mass flow of the working fluid. For this process and the given boundary conditions the calculated COP results in a value of around 6.



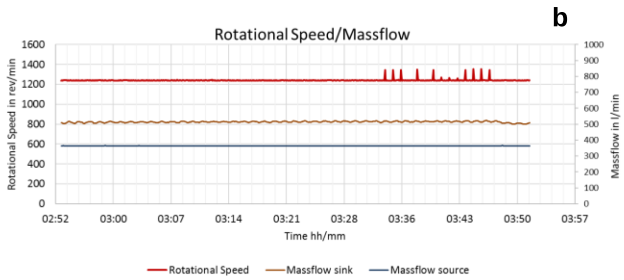


Figure 7: Temperature of sink and source (a), mass flow and rotational speed (b)

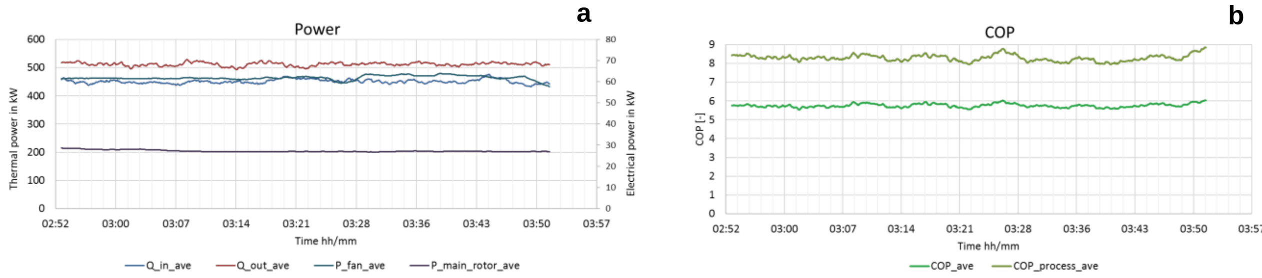


Figure 8: Thermal and electrical power (a), COP (b)

The second set of test results show the flexibility of the process over a given temperature range. This should only be an example, for the process itself it is possible to use a wider range. Fig. 9 and Fig. 10 show measured and calculated values over time. This curve shape can be realized by adapting the position of valve V1. So, for the descend of the temperatures till 05:50, the valve was more opened compared to the second half of the curve where the temperatures ascend. The rotational speed of the main rotor and the mass flow of sink and source are corresponding to the first example above. A different position of valve V2, which is controlling the mass flow of the source through the heat exchanger HX2, results in another temperature spread. In this case the spread between “T sink out” and T source in” is approximately 10 K. Because of less thermal power transferred, compared to the first case, the COP results in a value of around 5 as shown in Fig. 10. The small peak of the COP at 05:31 is caused by a short time when the electrical power consumption of the fan was less than before and afterwards. This may be caused by changed flow conditions in the gaseous system, especially at the fan. Anyway, it is clearly noticeable that the temperature level of the system can be adapted and changed during the process while the COP is not nameable affected.

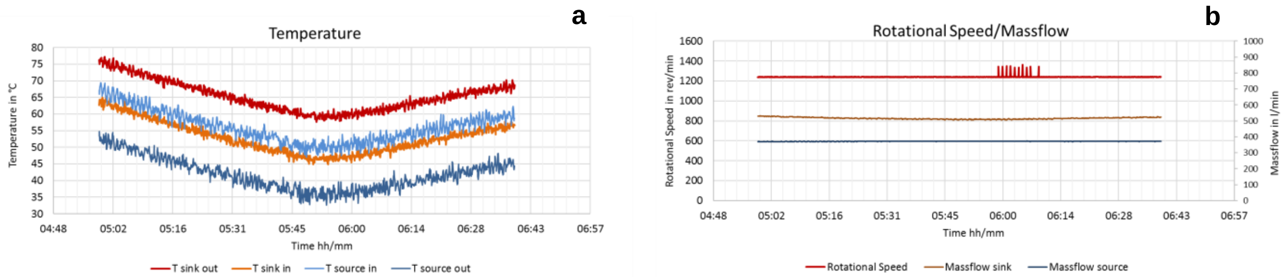


Figure 9: Temperature of sink and source (a), mass flow and rotational speed (b)

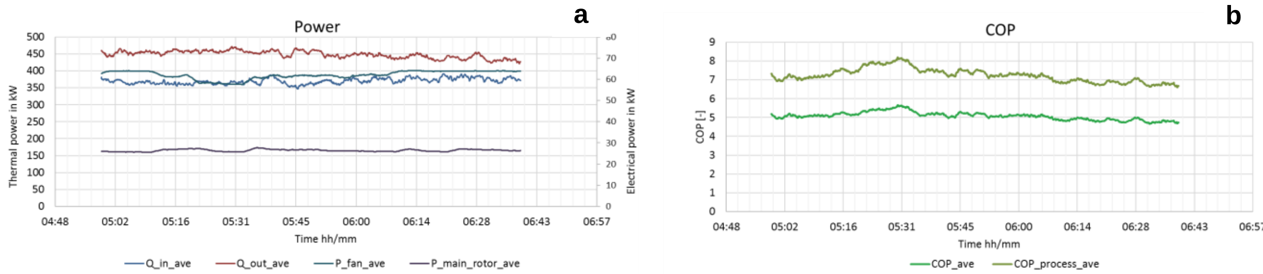


Figure 10: Thermal and electrical power (a), COP (b)

The last example shows the step response of the system, therefore the transferred heat between sink and source was adjusted in a relatively short time (<1min). The variation takes place at 04:26, when the temperature of “T sink in” and “T source in” approach each other. This test scenario could be a Dirac jump because more or less power is requested at the sink in a short time period. The jump shown in Fig. 11 (a) results in a decrease of thermal power, Fig. 12. Because of the averaging the step is displayed as a ramp and not instantaneously as the temperatures show. Additionally, the COP drops from around 6 to 5 which is the result of less thermal power transferred while the electrical power consumption remains almost constant.

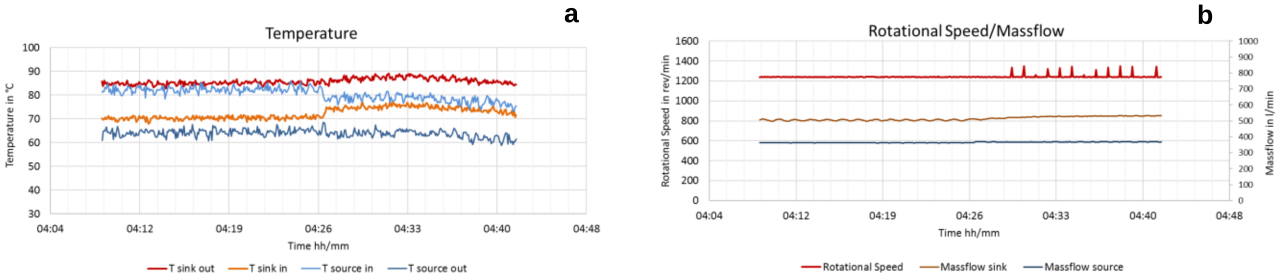


Figure 11: Temperature of sink and source (a), mass flow and rotational speed (b)

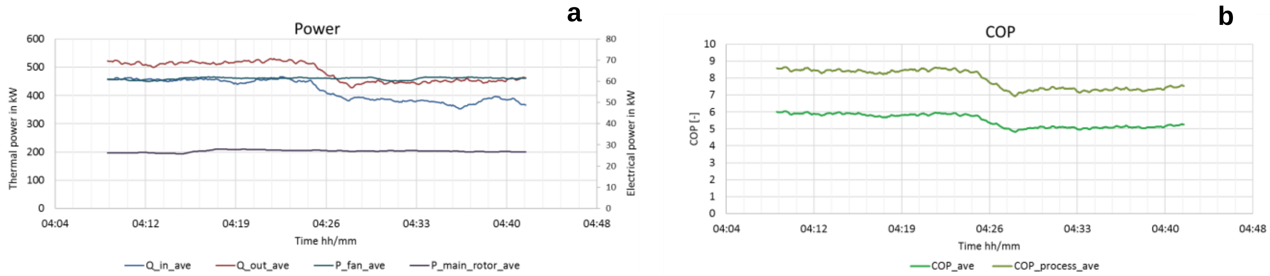


Figure 12: Thermal and electrical power (a), COP (b)

One more test was done at a higher rotational speed of the main rotor in combination with changing temperatures at sink and source. The case, shown in Fig. 13 and Fig. 14, describes a drift from around 68°C up to around 91°C of the sink outlet temperature at a speed of 1400 rpm. The mass flow of sink and source is controlled to be at a constant value of 450 l/min. Compared to the previous cases the power for the main rotor is slightly higher at this speed. This is caused by the increased losses of for example bearings and sealings. The temperature drift for this case has been done by changing the value of valve V1 in a way that less heat is transferred out of the system. In terms of the COP there is no significant change during the whole time of the test case, including the drift to a higher temperature level, starting at 05:02, as well. Because of the heat transferred and a higher temperature spread between sink and source, more electrical power is needed as shown in Fig. 14. This affects primary the power of the fan because of changed conditions of the process. For a closed cycle it is necessary to compensate the divergence of the isobars which in fact results in a higher pressure increase to be provided by the fan.

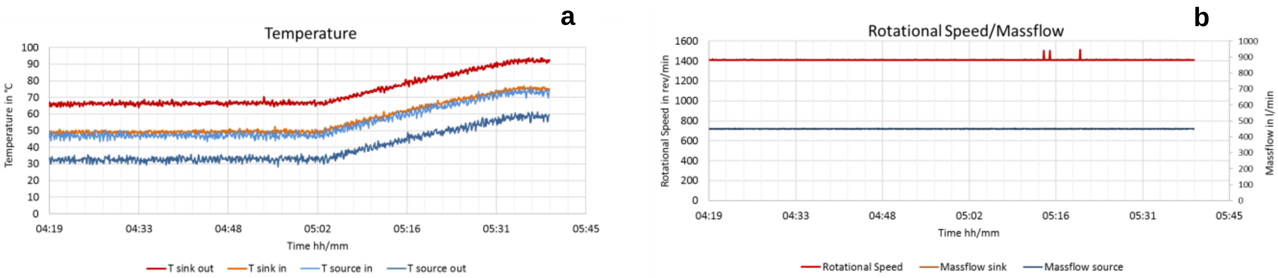


Figure 13: Temperature of sink and source (a), mass flow and rotational speed (b)

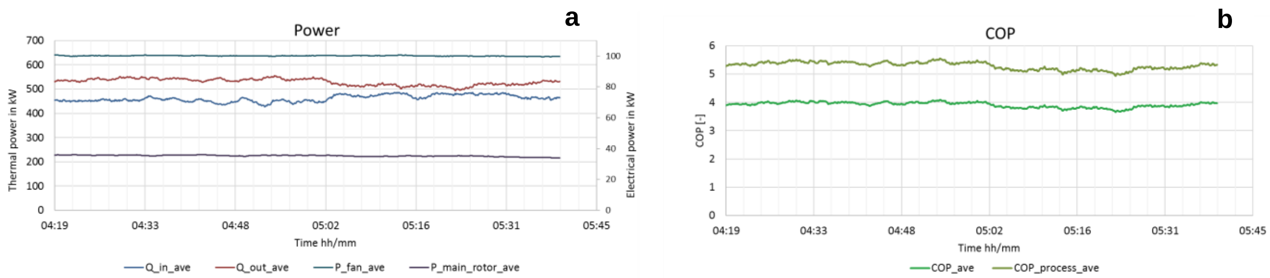


Figure 14: Thermal and electrical power (a), COP (b)

4. CONCLUSIONS

The operation and testing of the RHP showed that the principle and theory of the Joule Cycle is working for this application as predicted, as well as that it is very effective and opens a wide range of further heat pump applications not used right now. Not only the possible higher temperatures but also the sensible heat transfer are unique as well as the usage of a Joule Cycle for a heat pump. This means in combination with an environmentally friendly working fluid where the GWP=0, which is nontoxic and not flammable that by use of the presented machine the CO_2 emissions of many applications can be considerably reduced in a save way.
The described system of the test rig and the Rotation Heat Pump were used for evaluating different COP values and show the dependence on different parameters. Actual test cases include results for constant boundary conditions and stationary operation, variation of the temperature range and system responses for a step at the sink and source inlet temperature. A simulation based on analytical calculations shows good accordance with the test results. Based on the tests shown in this paper, some more test cases will be scheduled where even more knowledge about the Rotation Heat Pump and its function will be generated. This includes cases for high temperature applications as well as different scenarios for industrial applications.

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