15TH IEA HEAT PUMP CONFERENCE
MAY 26–29, 2026 | HOFBURG
VIENNA 2026
DECARBONISATION THROUGH INNOVATION

Application of a two-stage centrifugal steam compression system as part of a cascade high temperature heat pump

Maximilian Kriese[1*], Elias Alami[1], Enrico Jende[1], Panagiotis Stathopoulos[1]

[1] Institute of Low-Carbon Industrial Processes of the German Aerospace Centre, Department of High Temperature Heat Pumps

Abstract

In this study the integration and operation of a two-stage centrifugal steam compression system as part of a cascade high temperature heat pump is described. The system will be implemented in a pulp and paper mill in order to use waste heat and provide steam at 6.1 bar(a), for drying purposes. At first, the integration concept of the demonstration case is described. The system to be developed should be capable of providing steam between 6.1 and 11.1 bar(a), which is needed for a broad range of paper products. Subsequently, that leads secondly and with respect to a future commercialization of the two-stage compression system to the requirement of a modular system in terms of performance as well as of the general setup (construction, drive, assembly etc.). Therefore, the piping and instrumentation diagram of the two-stage system is presented. This is followed by the findings and lessons learned while carrying out the design and operation of an existing steam centrifugal compressor. Based on that, the preliminary design as well as the optimization of the flow path is described, including the boundary conditions (material choice, design constrains, inlet conditions etc.). Lastly, steady-state and quasi-static simulations have been conducted for performance evaluation of the two-stage system. The discharge pressure required for a wide range of paper types can be achieved. Additionally, the quasi-static time series has been performed to check if the stages match while starting up the system. At first a base case was carried out to ensure a stable operation, which is followed by an optimization. It is shown, based on the optimization, a surge margin of 15% and a prevention of choke, can be achieved in any operating point.

Keywords: high temperature heat pump; steam compressor; centrifugal compressor; R718, process heat

1. Introduction

In recent years, various studies have evaluated the application potential of a high-temperature heat pump (HTHP) for different industries, see e.g. [1], [2]. Besides the food and beverage, the chemical and the metal processing industries (non-ferrous metals), in particular the paper industry has proven to be suitable for HTHP's. The paper industry is among the world's five largest industrial sectors in terms of total energy consumption. In concrete terms, this translates to 6% of global energy consumption and a share of 2% of global CO_2 emissions [3]. Most of the energy is required for the generation of process heat [4], mostly below 200 °C. HTHPs are highly suitable for decarbonizing this demand, although only a few prototype systems reach these temperatures today [5], which is why a demonstration case with the outlook of commercialization, as to be presented in this paper, is needed.

1.1. High temperature heat pumps for the pulp and paper industry

In order to develop a HTHP which is capable of delivery steam up to 200°C for a broad range of paper types, the specific integration requirements needed to be investigated. Table 1 lists the paper types that account for the largest share of the production volume [6]. The steam temperatures have been corrected upwards compared to [6], because the temperatures do not correspond to the specified pressures according to the steam state diagram. Subsequently, a two-phase mixture would be present instead of saturated or superheated steam. The paper types can be categorized in three groups, requiring superheated steam of 6, 8 and 11 bar(a).

* Corresponding author: maximilian.kriese@dlr.de

Additionally, to define the development goals of a HTHP, it is necessary to know the available waste heat of the process as well.

Paper drying is accomplished by passing the moist paper webs over steam-heated drying cylinders. Air is circulated around both the paper webs and cylinders to absorb the moisture. After the drying process, the moist air has a comparatively high energy content, which is currently mainly wasted. The decisive factor is the water content in the air respectively the dew point of the water. Depending on the process, waste heat between 60 to 70 °C is available. Subsequently, the temperature lift between 60 and 200°C defines the boundary conditions for the integration of an HTHP, neglecting any losses at first.

Table 1 – Process heat demand for different paper types

Paper type	Drying temperature [°C]	Steam pressure [bar(a)]
Graphic paper	160	6
Corrugated board	185	11
Solid board	185	11
Folding boxboard	185	11
Sanitary paper	171	8
Moulded fibre	n/a	n/a
Special paper	160	6

In addition to the technical implementation, economic considerations are highly relevant as well. In order to define the economic efficiency of HTHP's, two facts are decisive. Firstly, the operation cost compared to natural gas boilers, which are mostly driven by the ratio of the natural gas to the electricity price, as this defines the minimum COP of HTHP. This value differs from country to country. Secondly, the investment cost, which can be reduced significantly if the system is modular. Modularization means a reduction of production effort due to economies of scale. According to [4] there are 842 paper mills (1157 paper machines) in the EU. Based on those numbers a prototype production is no longer considered economically, which is why a new modular heat pump system, as to be presented in this paper, needed to be developed.

2. Description of the demonstration case

In recent years various projects have started (e.g. SPIRIT [7], Push2HEAT [8] or SOLINDARITY [9]) to investigate how decarbonization of the pulp and paper industry can be achieved by applying HTHP's. In those projects the required process heat is only partially generated applying HTHP.

In the SPIRIT project a mechanical vapor recompression (MVR) system is applied to demonstrate that steam at a pressure of 6 bar(a) can be provided to the paper mill. However, the process steam is not generated by utilizing waste heat. In the Push2HEAT project, the waste heat from the hood is utilized on the one hand and a cascade heat pump is applied on the other, but only steam at 2 bar(a) is generated. In other words, the process steam generated does not reach the pressure and temperature level required to dry the various types of paper. In the SOLINDARITY project a different approach is applied by preheating the condensate before it's entering the boiler. Another special feature is the coupling of the HTHP with a solar-thermal system.

In contrast to the projects mentioned above, the EEETHOS project is the first of its kind aiming for a holistic integration of a HTHP into a paper machine. The integration concept as well as the requirements of the compressor technology applied, is described below.

2.1. Implementation setup

In Figure 1 the basic scheme of the cascade heat pump system is depicted, connecting three different technologies. The project partners are Felix Schoeller, HEATEN and the German Aerospace Center (DLR).

The cascade system consists firstly of a heat recovery system, secondly of a hydro-carbon heat pump and thirdly of a two-stage turbo steam compression system. The heat recovery system, for which Felix Schoeller is responsible, consists of a heat exchanger, which will be integrated in the hood of the paper machine and a water-glycol cycle. The dew point of the moist air will be at approx. 60 °C. In the current design state, a continuous temperature in the water-glycol cycle of 50 °C is considered. The delta results due to a limited effectiveness of the heat exchanger as well as some heat losses while transferring the heat. The same issues will occur while transferring the waste heat to the hydro-carbon cycle, which is why on the source side 40 °C are considered.

The hydrocarbon heat pump from HEATEN employs R600a (isobutane) as working medium and utilizes a piston compressor. Depending on the operating point, either 2 or 2.4 bar(a) of steam is to be generated at the sink. The difference arises from the requirement of a modular overall system to generate the necessary process heat between 6 and 11 bar(a) steam in the paper industry, as shown in Table 1. On the one hand, this enables an optimal operation of the two turbo compressors at peak efficiency in every operating point. On the other hand, there is a certain flexibility in operation in order to achieve the best possible overall performance (COP).

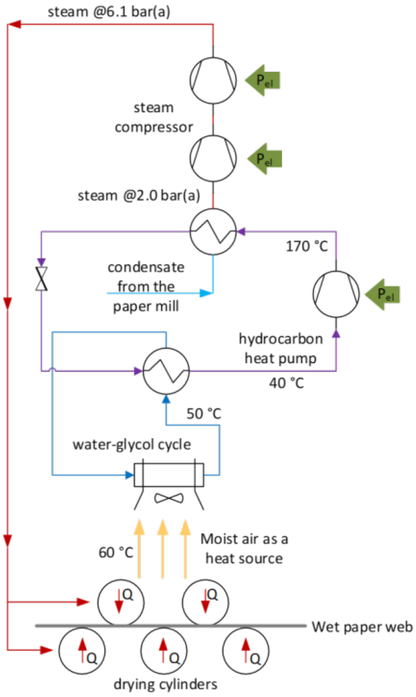

Figure 1 - Simplified integration setup of the cascade high temperature heat pump in a paper mill

The two-stage compression system of the DLR is in particular the scope of this paper. The centrifugal compressors operating in serial including an interstage cooling. Both stages have a nominal speed of 90.000 rpm and are especially aerodynamically optimized in terms of efficiency for this application. Sharing the same drive setup, meaning gear box, electric motor, frequency converter, skid etc., a modular system is to be constructed. This intends to reduce costs in the demonstration case on the one hand and increase the marketability of the compression system after the project ending on the other hand. Additionally, the two compressors and the subsystems (lubrication system, valves, armatures, interstage cooling, fittings and so on) are installed in a standard container.

The system has a total thermal capacity of 1 MW. Furthermore, the technology readiness level (TRL) is to be increased from 5 to 7. In the end, the project is aiming for at least 2.000 operating hours so that operation and design can be optimized based on the experience gained.

2.2. Commercially available steam compression system

Considering the commercially available systems in [5], the development of new HTHP's is arguably quite straight forward. Only a few manufacturers can provide HTHP's or compression systems which can supply sink temperatures above 150 °C and using water / steam as it's working medium.

The compressor types used can be categorized into positive displacement and turbomachines, each of which has its own advantages and disadvantages for their application within HTHP's. The positive displacement compressors are able to deliver a comparatively higher-pressure ratio per stage [10] and are suitable for small and medium volumetric flow rates. The drawbacks are a comparatively larger design (volume and weight) and the higher costs per m^3/h. In addition, oil lubrication is required, which can contaminate the working medium and therefore oil separators need to be installed [11].

The turbomachines are in general more applicable in terms of volumetric flow rate in conjunction with their construction volume. Due to that they have a lower weight as well. Additionally, turbomachines are characterized by their high efficiency with constant load. However, the more complex start-up procedure and part-load behavior [12] is considered unfavourable. In addition, the achievable stage pressure ratio is lower in comparison to the positive displacement compressors.

Within the DLR, the available tools for turbomachine design as well as the know how are decisive factors in addition to the advantages mentioned above, why it was decided to use a turbo compression system for the demonstration case. To date, only a few companies have developed steam compressors to market maturity that fit the application profile. In [5], these include the turbo compressors from WSETurbo [13], Pillar Blower [14] and the MVR system from EPCON [15]. The compression system of WSETurbo is suitable for the application in terms of capacity, but the compressors are limited to sink temperatures of 160 °C. Pillar compressors are suitable in terms of thermal capacity as well as in terms of achievable sink temperatures. However, the pressure increases per stage are comparatively small, which is why only a multistage application can be considered. That is comparatively resource-intensive and therefore not considered suitable. The last example, the MVR system from EPCON, can be compared with the WSETurbo system in terms of the performance spectrum (power output) as well as of the achievable discharge pressure [16]. Additionally, there is also a limitation in the achievable sink temperature.

In addition to addressing the technical challenges associated with achieving the required sink temperatures and pressures, the topic of modularization is of significant relevance for the DLR. Additionally, to the economic figure in section 1.1, Marina et.al [17] have shown that a large number of HTHP systems with a thermal demand of 1 to 5 MW will be required in the upcoming years. If a COP of at least 2.5 is considered so that the systems can be operated economically, a multi-stage turbo compressor system of 150 / 300 / 600 kW drive power per stage is required.

For all these reasons, a new two-stage turbo compressor system needs to be developed and will be subsequently applied within a heat pump system.

2.3. Compressor system design and requirements

Optimal integration and operation of the compression system within the cascade heat pump requires the consideration of various boundary conditions while designing the individual components. Two special features are explicitly discussed in this paper.

1. Demonstration of multiple operating points
2. Transient operation using the start-up procedure as an example

(1) As mentioned above, system modularization is of particular relevance in order to enable market penetration after project ending. In the long term, this should save costs by applying the same cascade heat pump over and over again in different industrial processes. A maximum of 6 bar(a) steam is required in the Felix Schoeller paper mill, but as described in section 1.1, the paper industry requires for drying purposes different steam temperatures (respectively steam pressures - 6 to 11 bar(a)) Therefore, the compressor stages must be designed for two operating points, in particular from a mechanical point of view. Accordingly, the first stage is engineered to withstand a maximum outlet steam pressure of 7 bar(a), while the second stage is rated for 11 bar(a), accounting for a safety margin.

(2) The start-up process in particular is considered in more detail below, because even considering a 24 / 7 operation, production interruptions due to paper tear occur almost on a daily basis. As soon as a paper tear is detected in the mill, the operation of combined heat and power plant (CHP plant) is shifted to part-load in order to generate just enough steam to prevent the drying cylinders from cooling down. As a result, the amount of waste heat required to operate the cascade heat pump is no longer available, which is why it has to be shut down.

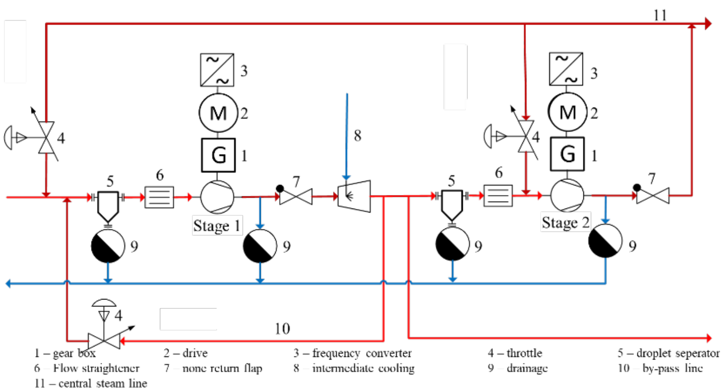

Figure 2 - Simplified P&I diagram of the two-stage steam compression system

It is therefore expected that the start-up procedure of the compression system is of particular relevance for a safe operation. In addition, the shortest possible start-up time with respect to an economical operation should be realized. Currently it isn't possible to estimate to which extent a continuous steam mass flow rate can be provided during the start-up of the hydro-carbon heat pump. Therefore, the steam compressors are initially started up using the central steam network. The start-up procedure is described as follows using the simplified piping and instrumentation diagram (P&I diagram) depicted in Figure 2, which includes the main components, relevant for the quasi-static simulation in section 4.2. The pipes are neglected for visibility reasons. The central steam network is represented by the central steam line (11), which is flooded with steam by the CHP power plant prior to the ramp-up of the cascade heat pump. At first the available steam is throttled to 2 bar(a) and fed to the first compressor stage. Due to the cold periphery, the steam is first passed through a droplet separator (5) to remove the condensate (9). This is followed by a flow straightener (6) to ensure an axial inflow and thus the best overall performance per stage. After compression, a non-return flap (7) will be integrated to prevent the compressor from becoming a turbine in the event of an emergency shutdown. Another condensate drain (9) will be installed upstream of the subsequent injection cooler (8), which is intended to limit superheating to 10 K before the second compressor stage. The injection cooler is initially not used during start-up in order to shorten the start-up time and limit the formation of condensate downstream. However, this cannot be completely prevented, which is why a droplet separator followed by a flow straightener will be integrated upstream of the second compressor as well. An almost simultaneous start-up of both compressor stages is currently considered. The ByPass line (10) for the first compressor is initially only intended for surge protection and not applied during the start-up procedure. A second ByPass is not required, as the additional steam required for surge protection can be used from the central steam line (11). Additionally, a second injection cooler is not considered to be necessary as the ratio of 1:10 of the injected steam mass flow rate of the HTHP compared to the generated steam mass flow rate of the CHP plant will not be exceeded. Subsequently, the temperature increase is not significant, which is why the injection cooler is neglected due do cost reasons.

3. Compressor design

In the following section, the design principles and assumptions / limitations for the design are presented at first. Secondly, a preliminary design study is presented. Subsequently, the setup of the compressors (drive, gearbox) and the aerodynamic optimization goals are described.

3.1. DLR's steam compression system

The DLR Institute of Low-Carbon Industrial Processes operates two prototype HTHP plants in Cottbus (lefthanded Brayton cycle based, called CoBra [18]) and in Zittau (lefthanded Rankine cycle based, called ZiRa [19]). Both systems utilize natural refrigerants, the CoBra uses air and the ZiRa uses water / steam. The compression system of the ZiRa is used as a blueprint for the compressor system to be developed for the demonstration case.

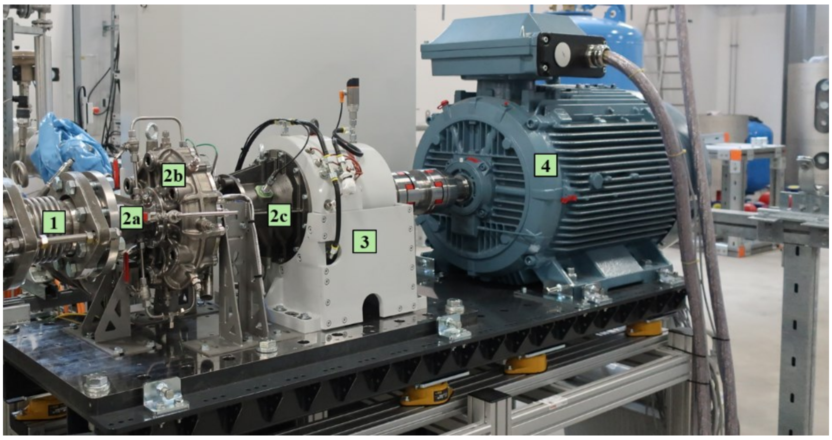

Figure 3 - Steam compressor stage for a high temperature heat pump

In Figure 3 the third compressor stage of ZiRa's multistage compression system is depicted, mounted on the DLR's test rig. From left to right, one can see upstream of the compressor inlet a compensator (1) that mitigate forces due to thermal expansion while operating the system. The compensator is directly connected to the compressor housing, which consist of three parts. The first two (2a & 2b) have been separated because of their weight. The purpose of (2a) is to direct the flow to the compressor leading edge. (2b) contains the stage (impeller, diffusor, volute) as well as the sealing and bearing system. The compressors shaft is directly coupled to the gearbox. The coupling needs to be lubricated, which is why a third housing (2c) is required. The gearbox (3) has a transmission ratio of 1:23.3. The max. speed of the drive motor (4) is 3000 rpm, subsequently the max. rotational speed of the compressor is 70.000 rpm, which is considered insufficient (low volumetric flow rate) and needs to be adopted for the two stages to be developed in the EEETHOS project. The compressor shown above has been designed for the following inlet conditions:

- Inlet pressure: 7.6 bar(a)
- Pressure ratio: 2.2*
- Inlet temperature: 185 °C
- Mass flow rate: 0.4 kg/s
- Working medium: steam

*The aerodynamic flow path of the stage has been optimized applying CFD. The desired discharge pressure is 15.5 bar(a), but to account for pressure and heat losses as well as deviations between CFD and measurements the optimization goal was set higher. This approach will be applied for the two-stage system presented in this paper as well.

The compressor commissioning / testing has happened in summer 2025 with air at first, and steam is to be followed as next. Nevertheless, some lessons learned / conclusion based on the design, commissioning and test phase can already been drawn and subsequently used for the compressor design of the EEETHOS project:

- The separation of the housing into two components (2a) and (2b) raises costs. This compressor is considered as a prototype, but reducing the number of components subsequently reduces the manufacturing effort, which is beneficial in terms of modularization.
- An asymmetric volute design (2b) would beneficial and lead to a:
  - Less complex back plate design
  - Less effort would have been necessary to account for the pressure level
  - Less manufacturing costs due to a reduction of components
- Balancing is necessary to ensure a stable and safe operation. Vibrations can occur on the impeller, shaft, gearbox, drive motor and so on. Therefore, it is recommended to have multiple options for placing weights or reducing it in order to ensure low-vibration level while operation
- Although, one oil type is used, two lubrication systems are applied. One for the gear box and one for the clutch and bearing system. That is due to a temperature limitation of the bearing of the gear box. This can be sort out comparatively easy. The application of just one lubrication system is recommended due to:
  - Less control effort
  - Less construction to ensure that the oil is refeed to each of the oil tanks
  - Less money is spent
- The comparatively high pressure requires:
  - A precise axial / radial thrust compensation in order to reduce the bearing load, which raises the compressors lifetime
  - Additional material of the housing (raises the weight of the compressor). Therefore, a suitable basement is required in order to avoid vibrations
- For controllability reason a measurement of rotational speed is recommend. In the example describe above the optical sensor is placed in the housing (2c). Due to the oil distribution, the visibility is limited and the measurement is not accurate. Therefore, the sensor should be placed on the slow rotating side (only if a gearbox with a fixed transmission ratio is applied).

Due to all the findings, experiences, advantages while designing the compressor mentioned above, it is considered as a suitable base for the two-stage application in EEETHOS. All those points mentioned, which can be optimized in terms of operation and construction, will be considered while construction the two-stage compression system.

3.2. Compressor preliminary design and optimization

CFturbo GmbH’s [20] design software is applied to perform the fundamental sizing and initial aerodynamic design of both compressor stages. Although the software is based on the relevant literature [21], [22], [23] etc., assumptions and decisions have to be made prior to the design. For example, the materials, the operating speed or the number of blades is user defined. Table 2 lists the inlet boundary conditions and the material parameters used for the design of both stages. On the one hand, the design is based on theoretical considerations and findings as described in [24], [25] und [26]. On the other hand, as described in section 3.1, the experience gained from past projects are incorporated.

Table 2 – Inlet and boundary conditions for the design of both stages

	Unit	Stage 1	Stage 2
Inlet pressure	[bar(a)]	2	4.2
Inlet temperature	[°C]	125	155
Rotational Speed	[rpm]	90.000	
Working medium	[-]	Steam	
Number of Blades	[Main / Splitter]	9 / 9	
Impeller material	[-]	Ti6AL4V	
Max. stress	[MPa]	550	
Shaft material	[-]	16MnCr5	
Housing material	[-]	Stainless steel	

The respective inlet pressures and temperatures are determined by cycle simulations, in which the potential of waste heat utilization was taken into account and used to determine the cycle layout. In addition, a preliminary design study was carried out in order to define the optimal speed for each stage. Due to the different inlet volume flow rates, the optimum speed varies between the stages. 90.000 rpm is defined as a compromise for both stages. The reason for this is that the setup, especially for the drive, of the two stages should be identical, i.e. modular. Each compressor is driven by a standard three-phase AC motor. The target speed is achieved via a gearbox. Compressor, gearbox and motor are mounted on a frame, similar as depicted in Figure 3, that allows different dimensions (in terms of power).

Table 3 lists both the results of the preliminary design (stage 1 and 2) as well as of the optimization (stage 1). Latter was carried out based on the setup described in [27], including aerodynamic and mechanical constraints.

Table 3 – Design parameters of both centrifugal compressor stages

	Unit	Stage 1	Stage 1 optimized	Stage 2
Flow Coefficient	[-]	0.053	0.0416	0.031
Work Coefficient	[-]	0.615	0.721	0.719
Pressure ratio	[-]	2.4*	2.744	2.3*
Efficiency	[%]	77.7	82.11	74.6
Mass flow rate	[kg/s]	0.37*	0.377	0.38*
Shaft power	[kW]	85.62	78.64	89.32
Torque	[Nm]	9.08	8.344	9.48
D_2	[mm]	120	120	115
b_2	[mm]	4.81	3.68	3.51
D_{1s} / D_2	[-]	0.479	0.2535	0.4357
D_{1h} / D_{1s}	[-]	0.348	0.4	0.399
b_2 / D_2	[-]	0.04	0.0306	0.031
u_2	[m/s]	565.5	565.49	541.9

*Specification based on the initial cycle design

The preliminary design results indicate that the flow coefficient is outside the recommended range due to the inlet boundary conditions and the drive setup [21]. As a result, the achievable stage efficiency drops significantly compared to state-of-the-art air compressors. Same applies for the pressure ratio. Nevertheless, comparing the results of the preliminary design to those of the optimization, a significant increase in isentropic efficiency (+4.5%) as well as in pressure ratio (+14%) is obtained. On the one hand, this enables operating both stages at part-load speed, which reduces the mechanical load per unit and therefore the operational risk. On the other hand, the difference to be expected between the CFD and the measurements can be compensated. As a result of the optimization, the ratios of D_{1s} / D_2, D_{1h} / D_{1s} und b_2 / D_2 decreased to their respective recommended lower design limits, see [26]. However, the values are in the range of reference centrifugal compressors for steam [12]. Based on the optimized geometry, the performance map was determined applying CFD once more. The impeller optimization of the second stage is currently carried out, which is why an adapted version of the first stage performance map has been used for the following simulations of the system performance and start-up procedure.

4. Simulation results

At first, the performance of the compressors is evaluated using a steady-state simulation model. Therefore, the P&I diagram shown in Figure 2 was modelled in EBSILON Professional [28]. Herein, the droplet separator and the flow separator are represented by a defined pressure loss of 25 mbar. The throttles are represented by a through flow coefficient characteristic, which were know from a different project with the same pipe size. In addition to the depicted scheme in Figure 2, the pipes are integrated as well in order to determine the pressure and heat losses at design and off-design operating conditions. The pressure losses are determined by the pipe geometry. The heat losses are determined by heat convection and conduction from the fluid to the environment. Therefore, the material properties of the insulation, the ambient temperature and the convective heat transfer coefficient must be determined. Latter mentioned is calculated based on a first order polynomial depended on the respective inlet mass flow rate. This level of detailed modelling is required in order to determine the inlet condition upstream of the respective compressor most accurately due to their impact on the respective compressor operating point. The two compressors are each characterized by their performance map and, for inlet conditions deviating from the design condition, the resulting pressure ratio and the compressor efficiency are determined based on Mach similarity [23]. The amount of water injected between the compressors is determined by a controller in order to realize a constant superheat of 10 K [29] at the inlet of the subsequent stage. At first, the performance of the two-stage system is evaluated and secondly a quasi-static start-up process is simulated to validate the matching of the stages.

4.1. System performance at design and off-design conditions

The goal of the development is a safe and stable operation while providing 6.1 bar(a) process steam using a cascade heat pump, on the one hand. On the other hand, the compressor must be designed / optimized in a way that they are able to provide the required process steam for all paper types, see Table 1. Latter depends mainly on the discharge pressure and temperature of the two-stage compression system. In addition to the pressures and temperatures, the respective pressure ratios and efficiencies over the rotational speed are shown in Figure 4.

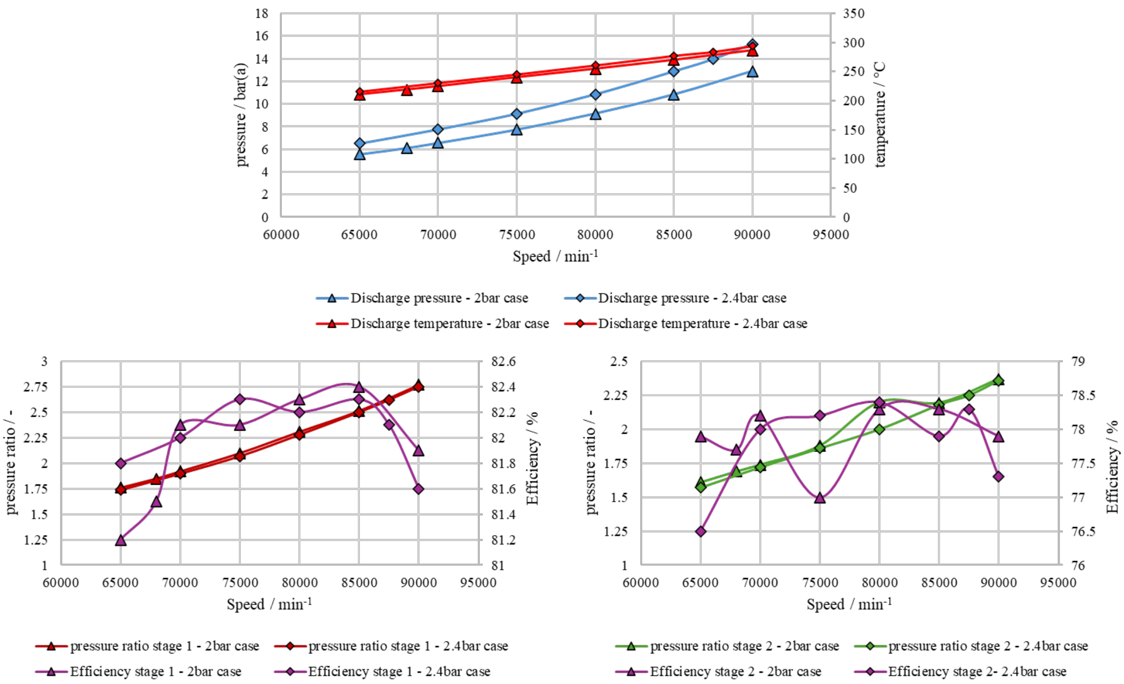

Figure 4 - System performance with increasing compressor speed

As described in section 2.1, two main operating points are considered: a steam pressure of 2 bar(a) and 2.4 bar(a) upstream of the first compressor. The design goal is to deliver steam at 6.1 (case 1) respectively 11.1 bar(a) (case 2). Those two operating points are the reason why in Figure 4 the minimum rotational speed was set to 65.000 rpm (for evaluation purposes).

In fact, for case 1, the first depicted operating point is the last one (in terms of rotational speed), at which 6.1 bar(a) cannot be achieved. It's vice versa for the case 2. Subsequently, at all other operating points a discharge pressure of 6.1 bar(a) can be achieved for both cases. The inlet mass flow rates feed upstream of the first stage were defined in order to ensure a safe operation with respect to the surge and choke limit. The comparability between the different operating point (with increased rotational speed) is in particular evident due to the small fluctuations in efficiency of less than 2% per compressor stage. In the end, at nominal speed of 90.000 rpm in case 1 a discharge pressure of 13.01 bar(a) is achieved and in case 2 a discharge pressure of 15.29 bar(a) is achieved. Consequently, the provision of the required pressurized process can theoretically be ensured for all paper types, regardless of the inlet pressure.

With respect to the demonstration case, the resulting discharge pressures aren't the only decisive factor, because as additional requirement the cascaded heat pump should achieve an overall thermal capacity of 1 MW. Therefore, a steam mass flow rate of 0.31 kg/s needs to be supplied by the hydro-carbon heat pump in case 1. Consequently, an discharge pressure of 9.91 bar(a) is achieved, due to the corresponding operating points in the performance maps considering an certain safety margin. These values have been obtained at an rotational speed of 82.500 rpm for both compressor stages. The same calculation has been performed for case 2. Due to the respective operating point at 77.500krpm for both stages, a discharge pressure of 9.73 bar(a) was achieved. Under these conditions, a steam mass flow rate of 0.35 kg/s needs to be supplied by the hydro-carbon heat pump. However, steam is still consumed from the central steam line due to surge protection requirements. Consequently, at this particular operating point, the steam mass flow rate needs to be uplifted even more.

On the one hand, one can argue that this demonstrates a certain flexibility at different inlet conditions, but one the other hand it also shows some lacking in load flexibility in comparison to the displacement compressor types, see section 2.2.

Of course, the discharge temperature increases constantly with the discharge pressure, varying form 210.7 °C to 286.9 °C for case 1 and from 215.3 °C to 293.6 °C for case 2. However, the highly superheated steam has no impact on the compressors expected lifetime, as the materials used (titanium alloy and stainless steel) have been selected accordingly.

In general, one can argue that the case 2 does not necessarily have to be investigated to ensure the provision of process steam for the different paper types. However, on the one hand the performance of the hydro-carbon heat pump in a wide operating range should be investigated. On the other hand, at a rotational speed of 87.500 rpm a discharge pressure of 6.22 bar(a) after the first stage is achieved, so that considerably the second one can be neglected. Of course, those are just very first calculation, but nevertheless it shows the potential of operational flexibility of the system.

4.2. Two-stage compressor start-up procedure

Even at an early design stage, matching of the compressor stages is highly important to ensure safe operation. In addition to the part-load operation, described above, start-up is particularly relevant, because while starting up the prevention of surge and choke is crucial. Kriese et al. have shown in [12] how this can be done for a three-stage turbo compression system with a limited operating range. A similar investigation has been performed for the two-stage compression system as depicted below. The start-up procedure is modeled as a time series of steady-state simulations.

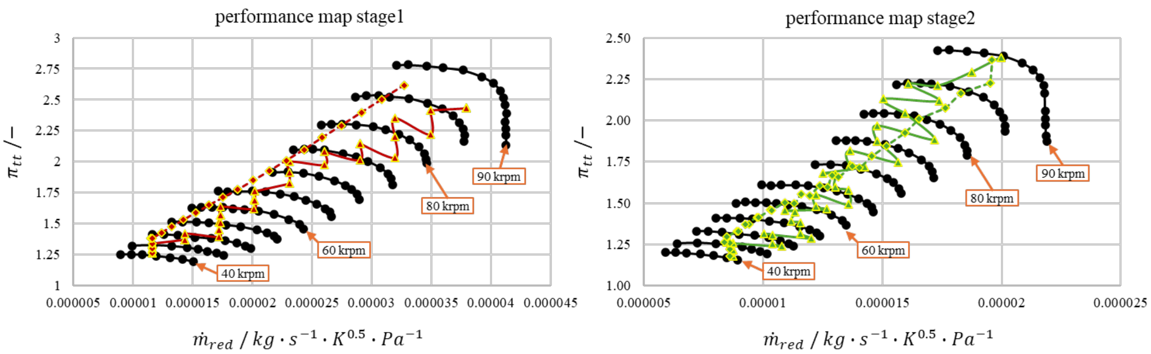

Figure 5 - Comparison of the working lines, baseline and optimization, while starting up both compressor stages

Figure 5 shows the working lines for the base scenario (solid line with triangles) and for the optimized case (dashed lines with rhombuses). As in [12], the first compressor is ramped up first, starting at 40.000 rpm. As soon as the first stage reaches 50.000 rpm, the second compressors speed is increased. The inlet condition of the first compressor met the design condition, which is not the case for the second compressor (inlet pressure is approx. 2 bar(a) lower). As a result, at comparatively low speeds (50.000 and 40.000 rpm), the first compressor operates close to the surge line and the second one close to the choke line (defined according to [21]). At first a base case was defined in order to ensure that a safe start-up is possible, see solid line with triangles for the first stage (red) and the second stage (green) in Figure 5. Initially, the first stage operates at a sufficient surge and choke margin, but with increasing speed (40.000 to 50.000 rpm), the operating point gets closer and closer to surge limit. Unfortunately, this is required to prevent the second stage at idle speed (40.000 rpm) from choking. During the subsequent speed increase, the operating point of the first stage repeatedly alternates between a sufficient surge margin and close to surge. Similar effects occur in the second stage, but in contrast it's the operation between a sufficient choke margin and close to choke. In the end, starting from 60.000 rpm a stable and safe operation can be defined for both compressors. Nevertheless, there is a need for optimization for two reasons. At first, in order to limit the operation of both stages at lower speed to a certain extend to surge and choke and secondly to reduce the overall mass flow rates, see equations 1a-c, especially at higher speed to make the start-up more efficient. Therefore, the following boundary conditions / restrictions were applied in the optimization of the start-up process:

$$\min_{x} \quad \dot{m}_{steam\ line} \tag{1a}$$

$$\dot{m}_{ByPass,1st\ stage} \tag{1b}$$

$$\dot{m}_{ByPass,2nd\ stage} \tag{1c}$$

$$subject\ to \quad \dot{m}_{steam\ line} + \dot{m}_{ByPass,i} \geq 1.15\ \dot{m}_{surge,i} \tag{1d}$$

$$\dot{m}_{choke,i} \geq \dot{m}_{steam\ line} + \dot{m}_{ByPass,i} \tag{1e}$$

$$\dot{m}_{steam\ line}(t) \geq \dot{m}_{steam\ line}(t-1) \tag{1f}$$

$$\dot{m}_{ByPass,i}(t) \geq \dot{m}_{ByPass,i}(t-1) \tag{1g}$$

A considerably safe surge limit is defined according to [23], see equation 1d. In this context, choking is considered less crucial for operation, which is why a mass flow rate smaller as the choke mass flow rate at the respective operating point, see equation 1e, is sufficient. In order to limit the controllability effort, only an increase in mass flow rate from one speed to another is allowed, see equation 1f-g.

The optimization reveals that the operating range has shifted from close to surge and choke towards a region with sufficient safety margin. The working line of both compressors can be described as smoothed. For the first compressor in particular, the working line starting from a speed of 50.000 rpm can now be described by a first-order polynomial with a fixed offset of 15 % to the surge line. In the low-speed ranges, there are hardly any differences to the base case, but the operation, as defined above, can be described as safe. As a result, a performance gain is achieved, as less volume flow is compressed during the entire start-up process.

The second stage's working line has been adjusted accordingly that it now achieves near-optimal efficiency while ramping up from idle to nominal speed. Nevertheless, more volume is compressed as required, so that the power consumption is higher as needed. This is due to the requirement for an equal or higher mass flow rate compared to the previous iteration step, which means that the total volume to be compressed is slowly but continuously increased, see also Figure 6. Since in general a safe operation is possible, for further optimization two options can be concluded. At first, a simple smoothing of the second stage's working line relative to the base case, or second, an adaptation of the optimization constraints. The objective is to ensure safe operation while minimizing control effort. The latter approach would additionally reduce the number of required components by potentially eliminating one or both bypass valves.

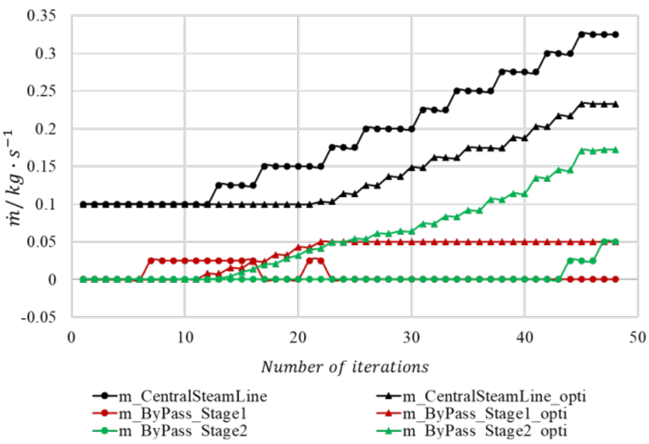

Figure 6 - Comparison of the base line operation to the optimized one with respect to the mass flow rates

Figure 6 shows the mass flow rates at each simulation point of the time series. The lines with the circle marks represent the base case and the lines with the triangle marks represent the optimization. Generally speaking, the controllability effort isn't that high, because the compressors (as mentioned above) are specifically designed for this application, nevertheless some aspects stand out.

In the first 10 iterations there aren't much changes, delta below 2 %. Beginning at iteration step 12 multiple changes occur (speed for stage 1 = 52.500 rpm, stage 2 = 45.000 rpm). First, the reduction of the mass flow rate supplied from the central steam line upstream of the first compression stage is particularly noticeable. The minimum amount of 0.1 kg/s is kept constant until iteration step 20 (speed for stage 1 = 65.000 rpm, stage = 55.000 rpm). Afterwards a nearly constant offset compared to the base case can be seen. In the end (last iteration step at nominal speed of 90.000 rpm) the mass flow rate used from the central steam line is 28.3 % lower compared to the base case. Therefore, from as early as iteration point 12, the mass flow rate is partially circled via the ByPass of the first stage. The amount increases constantly up to 0.05 kg/s and maintaining a plateau afterwards. Secondly, the required mass flow rate consumed from the central steam line in order to prevent the second stage from surging, raises significantly. In the end, even with an optimized performance in terms of aerodynamic operation, the optimization constraints need to be adopted in order to account for the controllability effort and system performance. This implies, it will be considered beneficial to raise the amount of steam consumed upstream of the first stage, so that the ByPass of the first stage will not be used for the start-up procedure. Additionally, a ByPass for the second stage should be implemented in order to get rid of the steam consumption of the central steam line between the stages to improve the overall system performance.

5. Conclusion and outlook

The present work reports on the successful integration and operational performance of a two-stage centrifugal compression system. This is based on an aerodynamic preliminary design, including know how and experience from previous projects, an optimization of the flow path and in the end a steady-state and time series simulation in EBSILON Professional. The CFD-based performance maps have been implemented in the simulation model, ensuring that the achieved pressure ratios and efficiencies are considered to be realistic. It has been shown that the serial operation of a the two-stage system is capable of delivery steam between 6.1 and 11.1 bar(a). Therefore, the two-stage design is considered modular, in a way that steam for drying purposes can be provided for a broad range of paper types. While feeding steam with a pressure of 2 bar(a) to the two-stage system a maximum outlet pressure of 13.01 bar(a) was calculated. Subsequently, the additional temperature increase provided by the hydrocarbon heat pump is no longer required. Nevertheless, if this is done, it is possible to increase the discharge pressure to 15.29 bar(a). Additionally, that led to a discharge pressure of 6.22 bar(a) after the first stage. Considering the application at Felix Schoeller, the second stage could therefore be neglected. As a result, there are various possibilities for increasing the overall performance of the cascade heat pump system. All that was achieved based on the experience while designing and operating other steam compression system, which were also presented in this paper.

However, these results should be considered preliminary for the time being, as the optimization of the second stage and integration of the performance map is still pending. It is already becoming apparent that the integration of a ByPass is necessary for the second stage, so that steam from the central steam line does not have to be used for the start-up procedure. The design effort and additional costs are negligible, as the second connection for extraction is not required.

In general, the effort of performing a preliminary design of the stage, followed by a CFD-based impeller optimization and doing an investigation if the stages match quite well, is considered pretty high. Therefore, in the next step a optimization framework will be developed to investigate the part-load behavior and start-up procedure of multistage compression systems already in the preliminary design phase.

Acknowledgements

At first, the author would like to thank the EU for funding the EEETHOS project. Secondly, the author would like to thank Felix Schoeller and HEATEN for their contribution, there valuable feedback and input. Thridly, the author would like to thank all members of the department of high-temperature heat pumps of the Institute of Low-Carbon Industrial Processes for their support, their knowledge and help while carrying out the demonstration case as well as this study.

References

[1] C. Arpagaus, F. Bless, M. Uhlmann, J. Schiffmann and S. Bertsch, *High temperature heat pumps: Market overview, state of the art, research status, refrigerants, and application potentials,* Energy, 2018.

[2] T. Fleiter, M. Rehfeldt and B. Pfluger, "A transition pathway for Germany's industry: which role for energy efficiency?," European Council for an Energy-Efficient Economy (ECEEE Industrial Summer Study) 2016, 2016.

[3] D. Furszyfer Del Rio, B. K. Sovacool, S. Griffith, M. Bazilian, J. Kim, A. M. Foley and D. Rooney, "Decarbonizing the pulp and paper industry: A critical and systematic review of sociotechnical developments and policy options," *Journal of Renewable and Sustainable Energy Reviews,* no. Volume 167, 2022.

[4] "Website of the European association representing the paper industry (CEPI)," [Online]. Available: https://www.cepi.org/wp-content/uploads/2025/07/Cepi-2024-Key-Statistics.pdf. [Accessed 04 11 2025].

[5] "Website IEA Annex 58," [Online]. Available: https://heatpumpingtechnologies.org/annex58/task1/. [Accessed 21 12 2022].

[6] K. Rademaker and M. Marsidi, "Website of Netherlands Environmental Assessment Agency," [Online]. Available: https://www.pbl.nl/uploads/default/downloads/pbl-2019-decarbonisation-options-for-the-dutch-paper-and-board-industry_3480.pdf. [Accessed 04 11 2025].

[7] "Website of the SPIRIT project," [Online]. Available: https://spirit-heat.eu/demo-sites/. [Accessed 23 04 2024].

[8] "Website of the Push2HEAT project," [Online]. Available: https://push2heat.eu/.

[9] "Website of the SOLINDARITY project," [Online]. Available: https://solindarity.eu/.

[10] G. Weber, Strömungs- und Kolbenmaschinen im Anlagenbau, Springer-Vieweg, 2019.

[11] J. Schiffmann and D. Favrat, "Experimental investigation of a direct driven radial compressor for domestic heat pumps," *International Journal of Refrigeration,* no. Volume 32, pp. 1918-1928, 2009.

[12] M. Kriese, M. Lockan, R. Schaffrath and P. Stathopolous, *Quasi-Static Simulation of a Start-Up Procedure for a Multistage Centrifugal Compression System,* Memphis: Proceedings of the ASME Turbo Expo 2025: Turbomachinery Technical Conference and Exposition, 2025.

[13] "Website of WSE Turbo," [Online]. Available: https://www.wseturbo.com/. [Accessed 10 11 2025].

[14] "Website of Pillar," [Online]. Available: https://www.piller.de/de/blower-design/. [Accessed 10 11 2025].

[15] "Website of Epcon," [Online]. Available: https://www.epcon.org/technologies/mvr-heat-pump. [Accessed 10 11 2025].

[16] V. Wilk, M. Bantle and S. A. , "DryFiciency - Deliverable D5.4 - Final report on the heat pump technologies developed," 2020.

[17] A. Marina, S. Spoelstra, H. Zondag and A. Wemmers, *An estimation of the European industrial heat pump market potential,* Renewable and Sustainable Energy Reviews , 2021.

[18] F. Yücel, J. Oehler, E. Jende, M. Kriese, N. Setzepfand and P. Stathopolous, *Design, Commissioning and Closed-Loop Operation of a High-Temperature Brayton Heat Pump – CoBra,* Applied Thermal Engineering, 2025.

[19] M. Kriese, S. Klöppel, N. Setzepfand, R. Schaffrath and E. Nicke, *Part-Load Behavior and Start Up Procedure of a Reverse Rankine High Temperature Heat Pump With Water As its Working Medium,* Boston: Proceedings of the ASME Turbo Expo 2023: Turbomachinery Technical Conference and Exposition, 2023.

[20] "Website of CFturbo GmbH," [Online]. Available: https://cfturbo.com/de. [Accessed 10 11 2025].

[21] M. Casey and C. Robinson, Radial Flow Turbocompressors: Design, Analysis, and Applications, Cambridge University Press , 2021.

[22] K. H. Lüdke, Process Centrifugal Compressor, Heidelberg: Springer-Vieweg , 2004.

[23] W. J. G. Bräunling, Flugzeugtriebwerke, Berlin Heidelberg: Springer Vieweg, 2015.

[24] M. N. Šarevski and V. N. Šarevski, *Thermal characteristics of high-temperature R718 heat pumps with turbo compressor thermal vapor recompression,* Applied Thermal Engineering, 2017.

[25] M. Šarevski and V. Šarevski, *Characteristics of water vapor turbocompressors applied in refrigeration and heat pump systems,* International Journal of Refrigeration, 2012.

[26] A. Meroni, B. Zühlsdorf, B. Elmegaard and F. Haglind, *Design of centrifugal compressor for heat pump systems,* Applied Energy, 2018.

[27] R. Schaffrath, E. Nicke, C. Voß, N. Forsthofer, Kunc and O., "Gradient-Free Aerodynamic Optimization With Structural Constraints and Surge Line Control for Radial Compressor Stage," American Society of Mechanical Engineers (ASME), Journal of Turbomachinery, 2025.

[28] "Website of Ebsilon Professional," [Online]. Available: https://www.ebsilon.com/de/. [Accessed 18 12 2022].

[29] B. Zühlsdorf, C. Schlemminger, M. Bantle, K. Evenmo and B. Elmegaard, Design Recommendations for R-718 Heat Pumps in High Temperature Applications, Proceedings of the 13th IIR-Gustav Lorentzen Conference on Natural Refrigerants, 2018.