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Performance analysis of a high-temperature heat pump with ejector based on butane as the refrigerant

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Abstract

The performance of an innovative concept for highly efficient heat pump suitable for industrial applications is presented in this paper. The heat pump is based on butane as the refrigerant, therefore a set of safety and design considerations were undertaken given the flammability of the refrigerant. For improved energy efficiency, the heat pump configuration with ejector was designed. The quantification of the energy performance was realized by a measurement campaign on the heat pump prototype, which was aiming at the temperature levels relevant for industrial applications: from 50–80°C range on the low temperature side, to 100–130°C range on the high temperature side. The final outcome of this analysis is the evaluation of the efficiency increase for the heat pump design with ejector, as compared to a standard configuration.

Keywords: Industrial heat pumps; Ejector; Butane;

1. Introduction

Various industrial processes cause waste heat that is being emitted to the surroundings, because it cannot be used in other internal processes due to its low temperature levels. An industrial heat pump capable of creating a temperature lift of the waste heat relevant for industrial applications could be a viable solution for using this currently idle waste heat potential. However, despite their economic and ecological advantages, the heat pumps are rarely deployed in the industrial practice, due to lack of mature technologies in the high temperature range.

In an attempt to open up this highly attractive heat pump market, the present paper analyzes the performance of an innovative concept of a heat pump for industrial applications with increased energy efficiency. The refrigerant deployed in this heat pump is butane, since it allows for high efficiencies, low discharge temperatures and low global warming potential (GWP). Given the flammability of butane, in order to deploy this refrigerant in an industrial heat pump, special safety and design considerations have to be performed. Aiming at the improvement of the energy efficiency, the design of this heat pump included the configuration with ejector.

In the process of designing this heat pump, the system simulation tools were used for the analysis of the dynamic behavior of the system, whereas the detailed numerical simulations were used to investigate the performance of

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individual components. For the purpose of quantitative analysis, a measurement campaign on the heat pump prototype was scheduled, with the results intended to be shown in this paper including the temperature measurements on the heat source and sink sides, at the temperature levels relevant for industrial applications. Namely, the intended application is lifting the temperature of a waste heat fluid from the range of 50–80°C, to industrially usable range of 100–130°C. Due to very challenging design and operating conditions, however, the measurement campaign was still under way at the time of the paper submission. Nevertheless, even these preliminary measurement results are confirming the increase of the coefficient of performance (COP) when the ejector heat pump configuration is deployed. Although the designed operating conditions have not been fully reached yet, this main outcome of the present work will be quantified and briefly discussed.

2. Heat pump configuration

The aim of the work presented here is to demonstrate improved efficiency of an industrial heat pump based on the butane as refrigerant, and to that purpose the ejector configuration was implemented. To allow for the experimental validation of increased efficiency of the heat pump configuration with and without ejector, the refrigerant cycle was constructed such that the heat pump prototype can be operated in both configurations. Clearly, the result comparison between these two operation modes makes it possible to quantify the effect of the ejector.

2.1 Refrigerant cycles

The configuration of a simple compression heat pump with single refrigeration cycle consists of the evaporator, condenser, compressor and expansion valve (Fig. 1 left). The rationale for implementing ejector in a heat pump comes from the fact that a limiting factor for the efficiency of single compression refrigeration cycle is the irreversible process of expansion valve throttling of saturated steam or undercooled liquid at high pressure to wet steam at low pressure. In order to reduce these process losses, and hence to increase the system efficiency, the use of a two-phase ejector is seen as a promising method (Fig. 1 right).

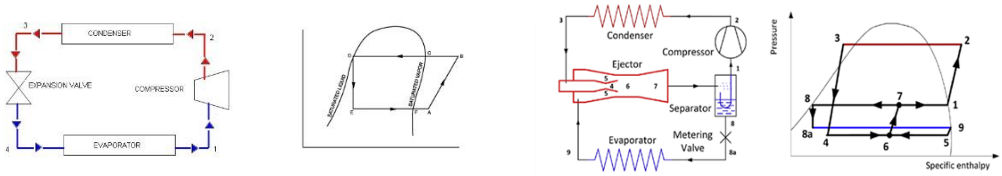

Figure 1: Scheme and p-h diagram of a simple refrigerant cycle (left) and standard refrigerant cycle with ejector (right).

2.2 Ejector

The basic idea is to put in direct contact the fluid streams from high and low pressure refrigerant cycle sides, and thus to use the expansion of a high-pressure fluid for carrying away the entrained low-pressure fluid. After this mixing, the fluid which leaves the ejector is at an intermediate pressure, and therefore the compressor does not have to overcome the entire pressure difference between condenser and evaporator, as is the case in the simple compression refrigeration cycle. The main parts of an ejector include the nozzle, mixing zone and diffuser, as indicated in Fig. 2. The high-pressure fluid is introduced in the nozzle, where it accelerates until the total pressure difference is converted into the kinetic energy. The low-pressure fluid is being introduced in the mixing zone, where it entrains into the high-pressure fluid due to its high velocity. In the diffuser an intermediate pressure level is established, with a simultaneous reduction in speed of fluid mixture.

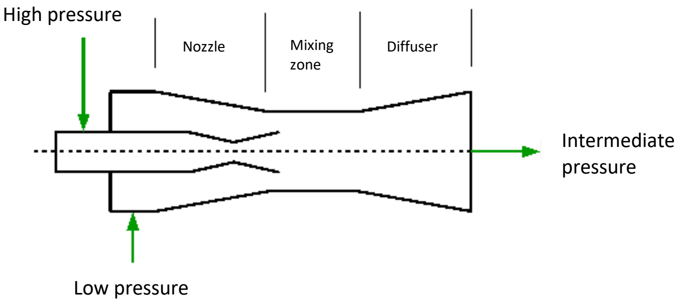

Figure 2: Typical ejector scheme.

The characteristic of an ejector is described by two parameters: the entrainment ratio (ER), and the pressure recovery (PR). The entrainment ratio is indicating how much of the high-pressure mass flow is mixed with the low-pressure fluid, and the pressure recovery is the ratio between the intermediate pressure and low pressure. The efficiency of the refrigeration cycle with standard ejector design depends critically on these characteristics: for poorly selected ejector performance characteristics, there is a danger that the refrigerant will not flow through the evaporator (somewhat more complex ejector design can eliminate this unwanted behavior) [1].

2.3 Refrigerant charge

Hydrocarbons, such as butane, are flammable and therefore when used as refrigerants special safety considerations are needed. Depending on the safety group of the refrigerant, machine type, and its location, the standard EN 378 [2] defines limits of refrigerant charges. According to EN378-1 Table E.1 and Table F.1, butane is classified as refrigerant group A3, and for refrigerants belonging to the groups A2 or A3 it is defined that permanently closed machine unit with a capacity of less than 150 g can be placed without restrictions in a workers-occupied zone which is not a special machine room. At higher refrigerant charges, the location of machines requires special safety measures, depending on the refrigerant toxicity and the location site. The location site is categorized in different groups, depending on whether the refrigeration unit is placed in a workers-occupied area; whether a refrigeration unit has the core refrigerant-containing parts (compressors and condensers) located outdoors or in a separated space or machine room which is not workers-occupied; or whether a refrigeration unit has all refrigerant-containing parts located outdoors or in a separated space or machine room which is not workers-occupied.

In the preset case all the refrigerant-containing parts of the heat pump are located in a vented housing (Fig. 3). The location site is defined to be in a machine room which is not workers-occupied area. Based on the constructed refrigeration system scheme, a risk analysis was conducted using the professional software (PED, from TÜV Süd) following the danger analysis according to pressure equipment directive. The standard DIN EN ISO 12100 [3] was subsequently considered, based upon which the risk assessment was carried regarding defined location site and risk analysis. With all risk reduction measures considered for this refrigeration unit, the system can designed such that no restrictions on the refrigerant charge are required.

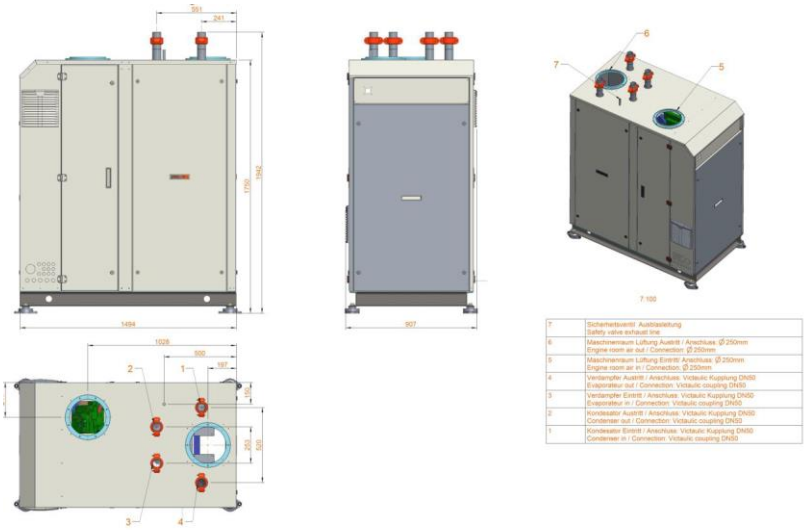

Figure 3: Heat pump vented housing.

2.4 Components selection

For refrigeration systems working with flammable refrigerants, it is sought to use the components that are permanently technically sealed. As described in VDMA 24020-3 [4], refrigeration systems with flammable refrigerants of the security group A3 according to EN 378, technically sealed system components can be fully or semi-hermetic compressors (e.g. separating hood compressors), and open compressors (with double mechanical seals). This imposes a strong limitation regarding the selection of the compressor for the designed heat pump. Namely, following ATEX 94/9/EC test [5], the operation area of open compressors is rated as the danger zone 1 (the area in which an explosive atmosphere can occasionally be formed during its normal operation), which as a design consequence requires a special explosion-proof measures for the electric drive. On the other hand, the operation area of semi-hermetic compressors is rated as the danger zone 2 (seldom and short term risk), which for the design safety requirements implies special protection against exceeding pressure limits, and specific design and arrangement of electrical equipment. Furthermore, the design measures must be taken to ensure a safe ventilation in case of refrigerant leakage: in no case must a flammable gas mixture occur. Given all these design requirements, as prescribed in the standards DE 378, Bitzer Kühlmaschinenbau GmbH was selected as the most suitable compressor manufacturer. They typically offer semi-hermetic reciprocating compressors for hydrocarbon applications, and for the designed heat pump their NESP 20P compressor type was selected.

3. Numerical analysis

The cornerstone of the present design of butane-based industrial heat pump were the steady state numerical simulations, in this work performed using the programming language Modelica under the simulation environment Dymola [6]. As presented in Fig. 4, the heat pump components (heat exchangers, compressors, expansion valve, regulation and control) are constructed using the commercial library TIL (TLK-Thermo GmbH), since it has a special focus on the refrigeration applications. The aim of these steady state system simulations for both refrigeration cycles was twofold: on the one hand it is the dimensioning of the individual components (both for the simple compression refrigeration cycle without ejector, and the ejector refrigeration cycle of the standard type), and on the other to investigate the extent to which the efficiency of the simple refrigeration cycle is improved by introducing an ejector in the heat pump system.

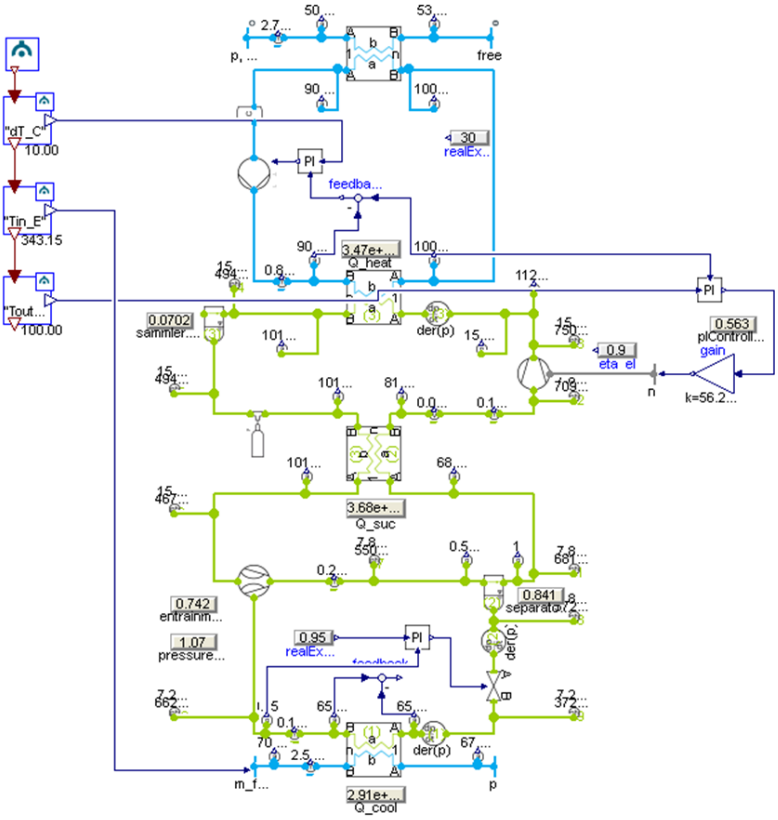

Figure 4: The scheme of the heat pump model in Dymola.

Looking at the characteristic saturation line of butane, it is clear that the refrigerant has to be superheated above a certain level, in order to ensure that the compressor does not operate in the harmful region of wet steam conditions. A large overheating without internal heat exchanger (suction gas heat exchanger) would lower the evaporation temperature and consequently reduce the efficiency of the refrigeration circuit. For both analyzed refrigeration cycles a semi-hermetic reciprocating compressor of Bitzer is used for a single-stage heat pump with 50 kW heating capacity, the designed evaporation temperature of 60 °C, the condensation temperature of 125 °C, and a compressor displacement of 56.25 m^3/h at 50 Hz frequency. For the desired temperature of the heat source system of around 60 °C and the heat consumption system of 100 to 130 °C, slightly higher temperature lift is obtained, and thus the obtained COP is somewhat worse. The design of the ejector refrigerant cycle is obtained from the compressor performance data, and the prescribed system values. The selected heat pump application (in the present case from the paper industry) determines the temperatures of the heat source and heat consumption, while the secondary mass flows are determined partly by the heat pump testing infrastructure, and partly by the temperature differences necessary for the accurate energy balance.

The resulting log p-h diagrams for the two analyzed refrigerant cycles are shown in Figure 5. For both refrigeration cycles the heat exchanger elements (condenser, evaporator and exhaust gas heat exchanger) are identical, while the thermal losses and pressure losses in the heat exchangers or pipes are not considered. The obtained log p-h diagram for the simple refrigeration cycles are generally in good agreement with the compressor design data, although the calculated temperature differences are slightly larger. In the ejector configuration the calculated overheating is slightly lower, because the point 1 in the log p-h diagram is always on the saturation line and thus the entire overheating occurs exclusively through the suction gas heat exchanger. According to the calculations for the design conditions (60 °C for the heat source, and 130 °C for the heat consumption), the heating COP for the simple cycle is about 2.6, while the implementation of an ejector increases the heating COP to about 3. In other words, the simulations predict the efficiency improvement of 15% through the ejector deployment. Hereby the ejector efficiency of 35% has been adopted in the simulation, according to the recommendations of SINTEF for the present application [7]. For the present conditions this efficiency corresponds to the entrainment ratio of about 0.5, and the pressure recovery of 1.25.

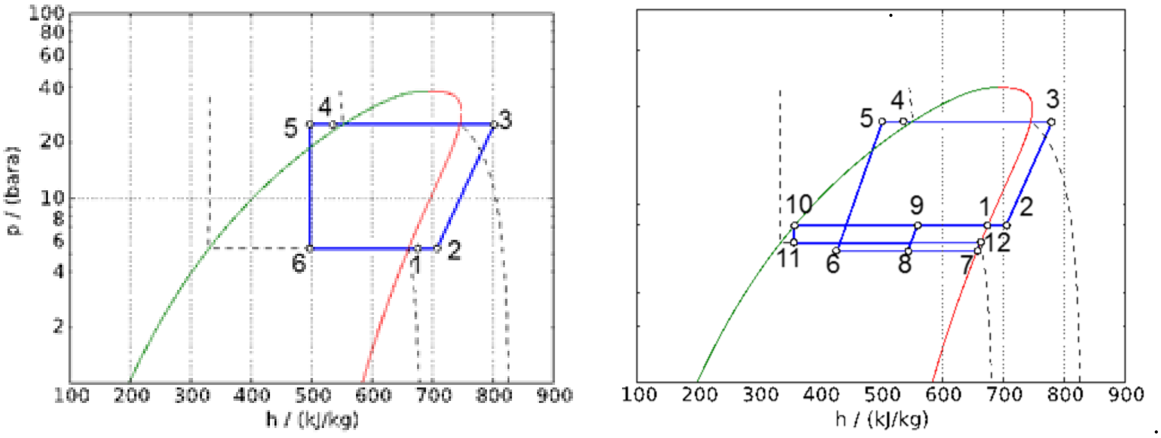

Figure 5: Simple refrigerant cycle (left) and standard refrigerant cycle with ejector (right) in log p-h diagram.

4. Measurements

From the hydraulic scheme of the heat pump prototype, shown in Fig. 6, one can note the design of the prototype which allows for a relatively easy switch between the simple compression refrigeration cycle without ejector, and the ejector refrigeration cycle of the standard type. From the provided measuring points one can compare the operating refrigeration cycle measurements with the numerical results on the one hand, and on the other one can check the pressure conditions in the ejector and throughout the refrigerant circle. Unfortunately, the measurement campaign was not entirely completed before the finalization of this paper, since it took longer than planned due to numerous challenges related to the heat pump design and the operating conditions.

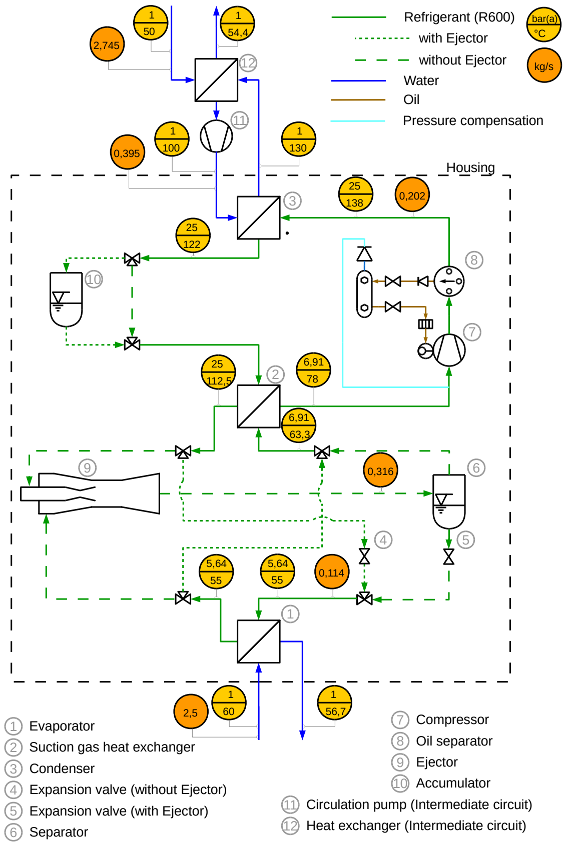

Figure 6: Hydraulic scheme of the heat pump prototype, together with the design operating conditions.

The measurement recordings of the temperatures from both the sink and source side of the heat pump are showing that in this early stage of the experimental investigation the reached temperature levels are not in the expected range. As the values of the averaged performance characteristics from Table 1 show, this holds both for the heat pump design with and without ejector. Nevertheless, the heat pump was driven in both modes (with/without ejector) was driven towards the same temperature ranges, in order to be able to compare the performance of the two heat pump modes (Table 1).

Table 1. Averaged performance characteristics of the heat pump design with and without ejector.

Parameter	Unit	Without ejector	With ejector
High temperature level	[°C]	72,8	80,4
Low temperature level	[°C]	46,8	53,2
Thermal power	[kW]	23,3	20,9
Electric power	[kW]	10,1	6,8
COP	[-]	2,2	2,9

Despite the instabilities in the operation, which impeded the full range measurements (both in terms of temperature and thermal power), one can see that the heat pump design with ejector has about 30% higher COP at moderate temperature levels, as compared to a standard heat pump configuration. Such an outcome is further confirmed by looking at the pressure distribution within the ejector (Fig. 7). Even though for reaching the desired temperature levels much higher pressures are expected, the ejector performance can be observed through the pressure recovery and entrainment ratio. Obtained from the ejector pressure measurements at observed levels, Fig. 7 shows that the pressure recovery is comparable to the value calculated in the design phase. The same holds also for the entrainment ratio, calculated from the energy balance on the evaporator and condenser. This evidence that the ejector effectively establishes the intermediate pressure (at which compressor is operating) explains the COP improvement obtained by ejector deployment (the reduction of the process losses of the expansion valve throttling.

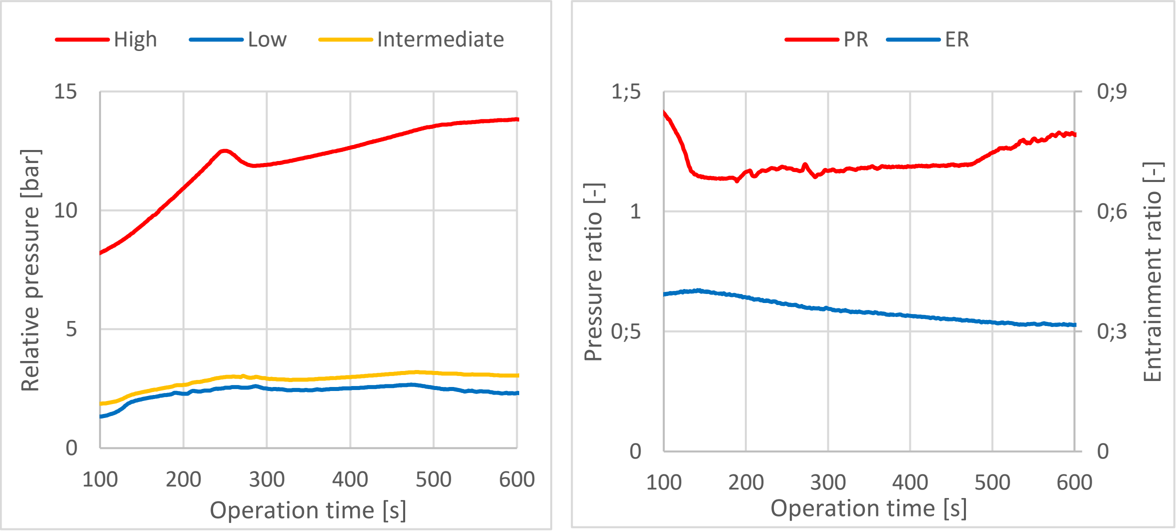

Figure 7: Pressure levels in the ejector heat pump configuration (left), and the ejector characteristics (right).

5. Conclusions

The main outcome of the work presented in this paper is the experimental quantification of the improvement with respect to the heat pump performance factor in the case of ejector heat pump design. The preliminary measurement results at moderate temperature levels show, that the ejector heat pump design brings about 25% increase of COP in this early operating phase, as compared to a standard design heat pump operating at similar temperature ranges. The ejector can create a proper pressure recovery of 1,3 with a corresponding mass entrainment ratio of 0,3 under the present operating conditions. Still, the Carnot efficiency of 15% during the initial operating phase leaves enough place for further improvements.

Acknowledgements

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