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Turbo-compressors: Prototype tests of mechanical vapour re-compression for steam driers

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Abstract

Steam driers show especially for the food industry improved quality compared to the more common air drying. One of the main advantages of steam drying is the possibility to recover the excess product steam through mechanical vapour recompression (MVR) and use this heat as thermal energy source for the process steam. Effective steam driers have a specific energy consumption as low as 0.8 kWh_{thermal} per kg of product steam. By applying MVR it is possible to reduce this number to 0.2 kWh/kg. In order to enable an effective and simple MVR-drier specialized atmospheric steam driers are available, which require a temperature lift of 25 Kelvin from the MVR. Consequently, the MVR system requires a pressure ratio of 2.5 when the process is operated at atmospheric pressure.

A prototype of a modified turbo-charger was tested in an experimental setup, with focus on cost effective and compact design. The electric drive capacity is 59kW and the DC-motor is directly mounted through the planetary gearbox on the impeller.

The main operation condition of the MVR-drier as well as the principal function of the turbo-compressor are explained in the present paper. Further, the accomplished experiments are described and the performance of the compressor is documented. It was possible to proof that the specific energy consumption of the drier can be reduced by 80% with the used technology. A simplified return of investment calculation shows that the system is paid back in one year. The technology has the potential to substitute fossil energy carriers.

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1. Introduction

The agreement on the Paris climate conference in December 2015 concluded to limit global warming to a 2-degree increase. 195 countries have adopted the agreement, and it is due to enter into force in 2020 [1]. In this context it is expected that the thermal process industry has to reduce both the use of fossil fuels and the specific energy consumption.

It is estimated that drying processes account for 15-25% of the total industrial energy demand of the developed countries, while at the same time having a low process efficiency of about 35-45% [2]. Many drying processes are based on convection and fossil fuels are used as energy source. The University of Stuttgart did an analysis of the potential of industrial heat pumps [3]. They showed that the use of high-temperature heat pumps with a condensation temperature of up to 150°C is particularly promising for drying processes, since suitable surplus heat are available at the same time as the high temperature heat is required. The specific energy consumption of drying can thereby be reduced by up to 80% [4], and at the same time, fossil fuels are replaced by renewable energy (surplus heat and electricity).

Pressurized steam is often used as energy carrier in thermal processes above 100°C, due to the good thermal properties as well as the fact that water is cheap, easy available and non-toxic. Steam based thermal processes are especially interesting when applying industrial heat pumps since water/steam also can be used as working fluid in the heat pump. Excess steam from the main thermal process can be applied directly in a heat pump called open-loop-heat pump or mechanical vapour recompression (MVR). In such a system, the main thermal process can act as the evaporator for the heat pump, reducing the amount of needed components and the heat transfer losses.

The specific energy consumption for superheated steam drying is in the best cases between 0.72 and 0.83 kWh per kg of product steam. The excess superheated steam has atmospheric pressure and can thus condense at about 100°C. However, since the process requires a significantly higher temperature, the energy of the excess steam is usually lost, and the process steam is heated by means of fossil fuels. By compressing the excess steam, it can condense at a higher temperature level, and the process steam can thus be re-heated to the desired drying temperature. The concept of superheated steam driers has already been investigated in a previous work [5], showing an energy saving potential of up to 75%. A principal sketch of such a system is given in Fig. 1.

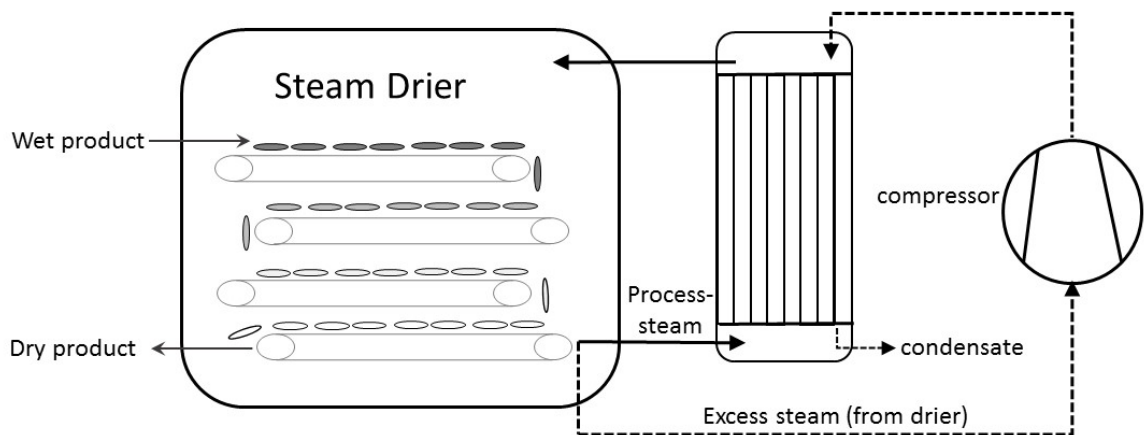


Fig. 1. Schematic layout for superheated steam drier with energy recovery through mechanical vapour recompression

The investment cost of the compressors for steam compression has been found as a barrier for commercial use of the technology [6], and it is emphasized that without development of low-cost compressors an industrial application is not to be expected. This is reinforced by country-specific energy prices at various locations. The turbo-compressors is considered as a potentially cost-effective and effective alternative to conventional compressors [7, 8]. Turbo-compressors are used in nearly every diesel vehicle for air compression, but they usually achieve a pressure ratio of a maximum of 2.5 when operated with steam. This means that a multi-stage compression system must be used when higher pressure ratios are required. For superheated steam driers it is also possible to reduce the required temperature of the superheated steam by increasing the residence time of the product, increasing the mass flow of superheated steam, optimize the design of the drying chamber and reduce system losses. With this measures, it is possible to construct superheated steam driers which operate between 125°C and 103°C at atmospheric pressure.

In this study, a single-stage turbo-compressor MVR system has been tested with the aim to operate between the specified operation conditions of such a superheated steam drier. The basic structure of the prototype is described in this article, and the experimental setup and the results of the turbo-compressor tests are included.

2. Construction of the turbo-compressor

The turbo-compressor used in this experiment is a further development of a conventional radial turbocharger from the automotive industry. The patented design has been adapted by the manufacturer (Rotrex A/S, Copenhagen, Denmark) for use in steam compression (see Figure 2). For this purpose, the impeller was designed in titanium; the rest of the casting is done in alumina. The carbon sealing between the compression chamber and the planetary traction drive was reinforced to ensure a better sealing. The aim of these modifications was to increase the achievable pressure ratio at a high efficiency and to improve the durability for long-term, continuous operation.

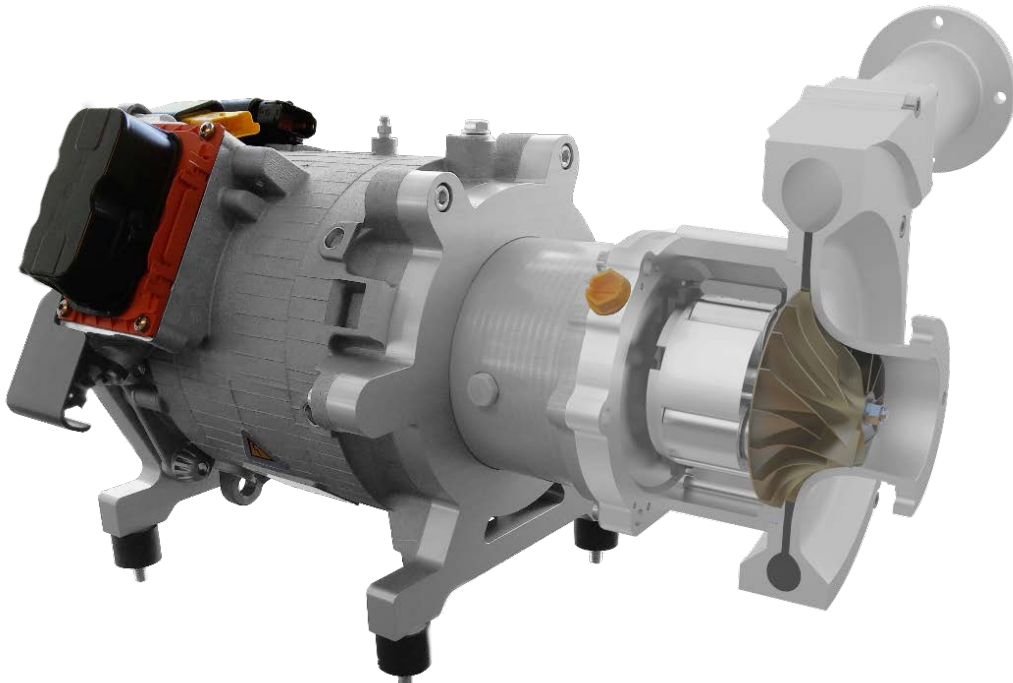


Fig. 2. Section view through the prototype of the turbo-compressor including motor and gearbox.

At the design point, the volume flow on the low-pressure side is $0.22\text{m}^3/\text{sec}$ and the compression ratio is 2.7. The isentropic efficiency of the compressor was calculated to be 74.5%. Depending on the temperature and pressure of the steam on the low-pressure side, it will be possible to achieve a thermal capacity of up to 300 kW while achieving a temperature lift of approximately 25 Kelvin (between high and low-pressure saturation temperatures).

A planetary traction gearbox is mounted on the drive shaft of the turbo-compressor that enables a high transmission ratio. The transmission ratio is 7.5 at a mechanical efficiency of 98% under full load. The planetary gearbox is lubricated by means of an internal oil pump and at the same time cooled by an external oil cooler.

A 650 Volt DC-motor which can deliver up to 12 000 rpm is placed directly on the drive shaft of the gearbox, thus a rotational speed of up to 90 000 rpm can be achieved at the impeller. The motor is driven by an inverter that delivers 59 kilowatt. Figure 3 is showing a principle layout of the primary components needed and used to run the turbo-compressor.

The total weight of the complete unit (turbo-compressor, gearbox, DC-motor) is about 40 kg and the dimensions are 50 cm in length, 40 cm in width and 35 cm in height, which illustrates the compact and light weighted design.

Figure 4 is showing the expected compressor map, and it shows that the efficiency of the compressor strongly depends on the flow rate and the revolution speed. The compressor must not be operated in an operating state outside the characteristic field shown in the figure, as otherwise surge or choke effect will occur.

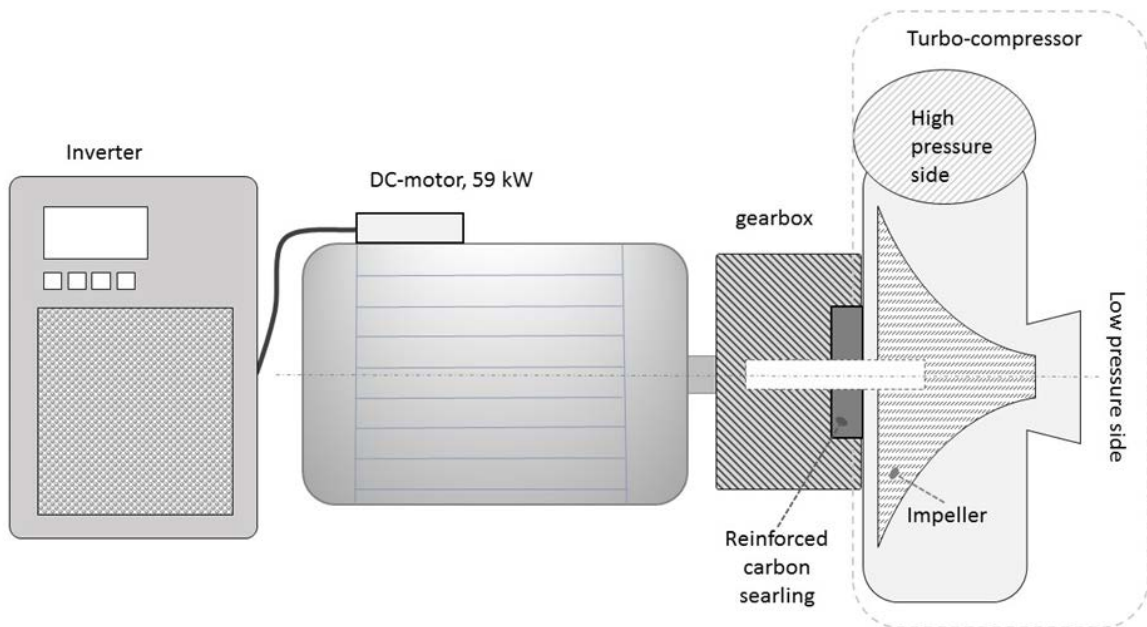


Fig. 3. Primary components of the turbo-compressor setup.

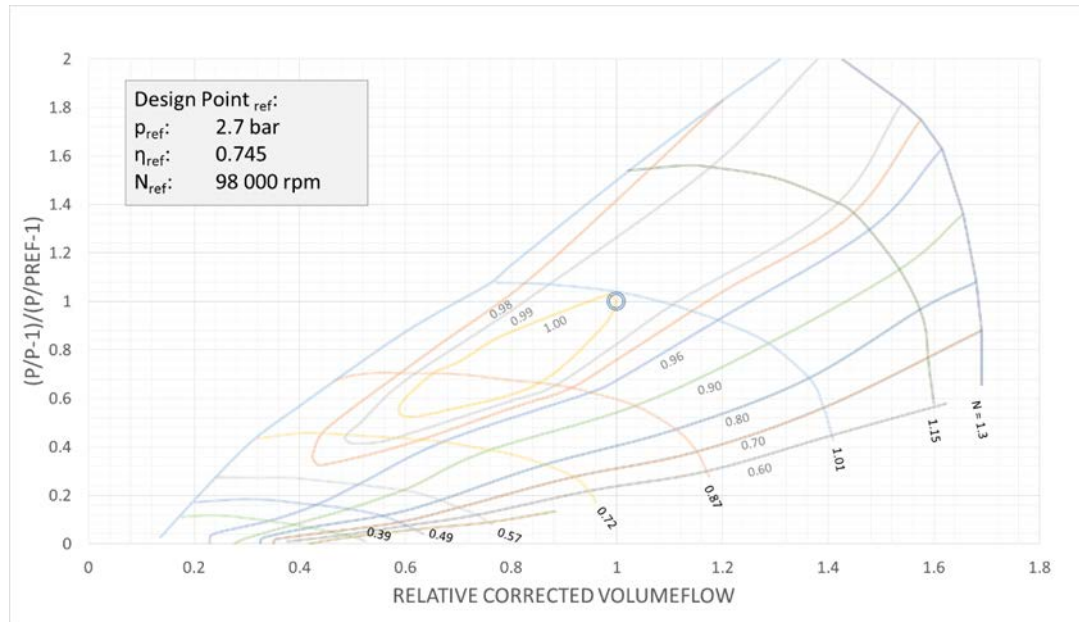


Fig. 4. Estimated compressor map for prototype compressor.

3. Pilot tests

3.1. Design and implementation

For the pre-industrial tests a superheated steam drier with a capacity of approximately 600kW was used to deliver up to 450 kg of steam per hour to the turbo-compressor. The temperature of the superheated steam could be varied between 150°C and 105°C, and the oxygen content was as low as 1% (± 0.2). During the tests, the drier was not filled with any drying products, ensuring that the steam was clean without any particles that could have damaged the prototype turbo-compressor. As the primary aim of the investigation was to document the performance of the turbo-compressor, the compressed steam was not fed back into the drier. Instead, it was condensed by an external MVR-condenser. A principle overview of the test-rig is shown in Figure 5.

The temperature and the pressure of the steam was measured and recorded both before and after the turbo-compressor, while the mass flow of the steam was measured after the turbo-compressor. These data was used to determine the pressure ratio and isentropic efficiency of the turbo-compressor. The operating state was at the same time used in the compressor map in order to illustrate the process and to determine if the compressor was getting close to the choke or surge area.

During the tests, the gearbox and the electric engine was water-cooled, and the required cooling capacity was documented by an energy flow meter. All pipe connections were based on DN100 in order to compensate for the high volume flow of steam, especially at lower pressures, and the connections of the test-rig were flexible couplings accepted for food technology, which were approved for up to 6 bar.

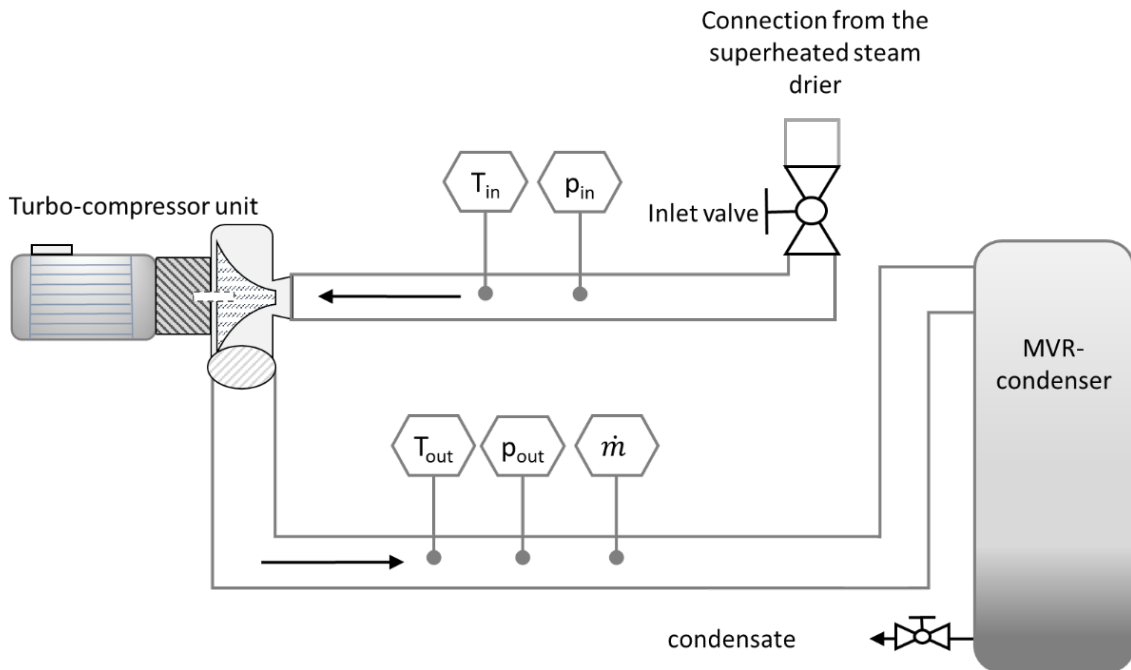


Fig. 5. Principal setup the turbo-compressor test-rig, which was retrofitted to the steam drier.

The superheated steam from the drier was cleaned with a particle/odor filter before it was fed into the test-rig, even if the tests were performed with clean steam.

Since the evaporating temperature of the working fluid (100°C) was higher than the ambient, a special start-up procedure was used to pre-heat the heat pump to avoid that too many liquid droplets came into the compressor. When the superheated steam drier had reached the desired operating temperatures and the oxygen content was reduced to approximately 1%, the inlet and expansion valve of the test-rig was opened and the turbo-compressor was operated at reduced capacity (20%). The temperature and pressure at the inlet and outlet of the turbo-compressor was monitored, and when the inlet temperature reached 105°C , it is ensured that no condensate was present in the system. The capacity of the turbo-compressor was then increased stepwise and for each step, the expansion valve after the MVR condenser was closed so that the pressure was increased.

3.2. Results of pilot tests

The test-rig for the turbo-compressor was operated for several days under various operating conditions. The initial tests were carried out at low capacity (about 50% of the maximum capacity) in order to make sure that the system was stable and make the necessary fine adjustment. Then the compressor was operated at full load (80–100% of maximum capacity). Figure 6 shows the achieved operational conditions of the turbo-compressor during the pilot tests. The points illustrated in the figure were held for at least 1/2 hour and the isentropic efficiency is based on the average performance during this time.

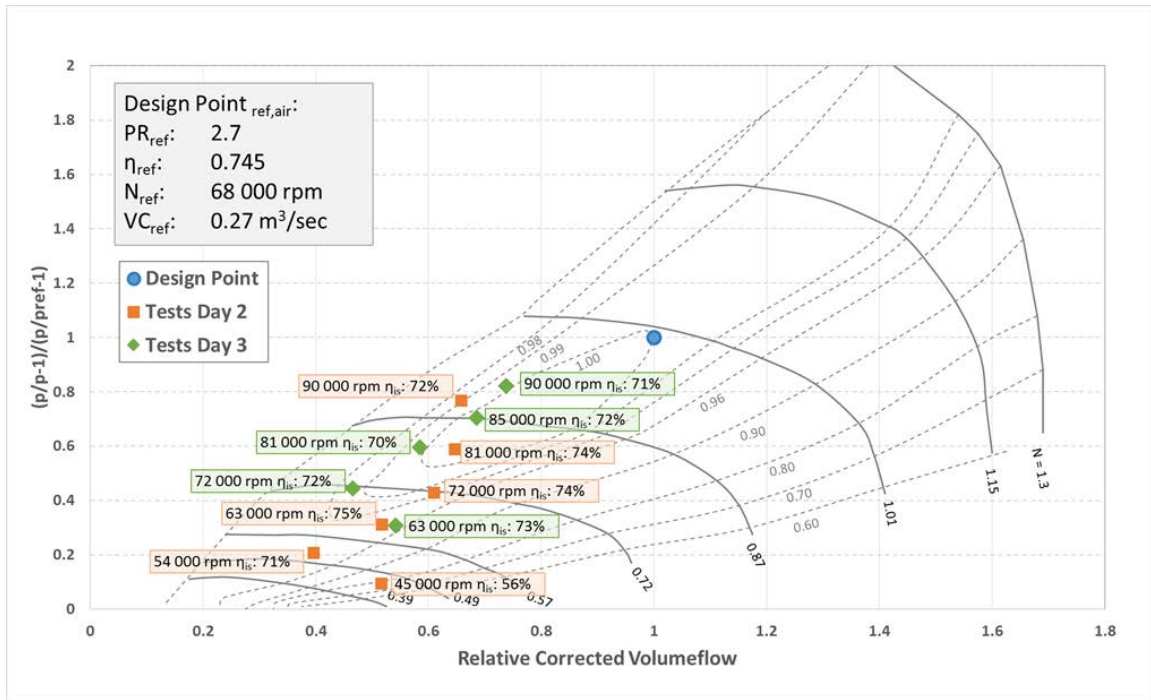


Fig. 6. Achieved, stable operational conditions of the turbo-compressor during the pilot-tests.

In general, the turbo-compressor was operated relatively close to the choke line in the compressor map. This has two reasons:

1. The pressure loss of the system (from piping, filter, condensers, etc.) is a system-specific parameter, which has to be compensated for by the compressor. The turbo-compressor is quite sensitive to pressure changes and in general does not tolerate high system pressure losses.
2. The achievable mass flow of steam from the drier was limited from the system to 450 kg/h. With higher available mass flows, it would have been possible to investigate a broader section of the compressor map.

4. Discussion and Conclusion

The operating conditions achieved (Figure 6) were stable and not influenced by the above-mentioned considerations. At a rotational speed of 90 000 rpm of the turbo-compressor (100% capacity), a pressure ratio of 2.4 was reached with an isentropic efficiency of 72% and a mass flow of 450 kg/h. The measured efficiency was close to (3%-points lower) the expected performance at the design point of the turbo-compressor (isentropic efficiency of 74.5%, pressure ratio 2.7 at 100 000 rpm and mass flow of 550 kg/h).

The outlet temperature of the steam from the turbo-compressor at this operating point was up to 225°C, which is a challenging condition for the lubricant cooling of the gearbox. A high degree of superheat was expected due to the thermal properties of water/steam at the designed operational point. Consequently, multi-stage systems will require intercooling of the superheated steam in order to give a higher compressor efficiency and reduce the thermal stress of the components.

The COP of the MVR system was approximately 11.5 in the performed tests. This seems quite high compared to other heat pump systems, but it is important to take into account that the temperature lift was only 24 Kelvin between the evaporating and condensing temperatures. Even if most superheated steam driers will need a higher temperature lift in order to operate, some manufacturers have already developed superheated steam driers where the process steam only need a temperature lift of around 25 Kelvin. The tested single-stage prototype turbo-compressor can be used in such a drying system.

A superheated steam drier which operates under these conditions will normally have a specific energy consumption of 0.8 to 1 kWh per kg of evaporated water. Such driers will typically be supplied by steam generators, which use fossil fuels as energy carrier. With the investigated MVR system it will be possible to reduce the specific energy consumption to <0.2 kWh/kg. This is an energy saving potential of more than 80%. It has to be said at this point that MVR driven driers will use electricity as main energy source. The energy price gap between fossil fuels and electricity will reduce the economic saving according to the cost price at specific production sites. A high COP of the MVR system is therefore required in order to compensate for the energy price different and give low return of investment for the MVR installation.

For a selected case and location, it was possible to estimate the return of investment to a few month for a turbo-compressor based MVR system. Hereby, it was considered that the electric energy is more than 4 times as expensive fossil fuel based energy (per kWh) and that the MVR system will achieve a temperature lift of 25 Kelvin. For a two stage system, which will achieve a temperature lift of 50 Kelvin the return of investment is estimated to be around one year. These numbers are based on preliminary determined investment costs.

The MVR system needs to be placed close to the drier in order to reduce pressure and heat losses between the two installations. This should be considered during the design of the production site and could be a challenge when retrofitting to existing drying system.

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