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Performance evaluation of two stage mechanical vapour recompression with turbo-compressors

Michael Bantle[(a)], Christian Schlemminger[(a)], Ignat Tolstorebrov [(b)],Marcel Ahrens[(a)], Kjetil Evenmo[(c)]

[(a)] SINTEF Energi AS, Department of Thermal Energy

7465 Trondheim, Norway, Christian.Schlemminger@sintef.no & Michael.Bantle@sintef.no

[(b)] Norwegian University of Science and Technology, Department of Energy and Process Technology

[(c)] Epcon Evaporation Technology, 7079 Flatåsen, Norway, ke@epcon.org

ABSTRACT

Mechanical Vapour Recompression (MVR) is an open loop heat pump system using water (R718) as working fluid, one of the most abundant and safest refrigerant on the planet. The concept can significantly reduce the energy consumption for steam based processes like drying, pasteurization, evaporation or distillation but also for steam production itself. However, the compression technology is commonly not cost efficient especially for small scale productions in the capacity range from 500 kW to 4 MW.

A two stage turbo-compression system was developed and tested based on mass produced automotive turbocharger technology. The turbo-compressor of the first stage reached a pressure ratio of 1.69 and is designed for a mass flow of 400-600 kg/h superheated steam. The second stage turbo-compressor had an identical design and achieved the same pressure ratio. Between compression stage one and two de-superheating is applied by water injection. With the developed system it is possible to compress superheated steam from atmospheric pressure to above 2.8 bar, where it can be condensed at a temperature of 131°C. The COP of the performed investigation was 7.8, when the achievable condensation energy is compared to the total amount of energy supplied to the system. The compressor efficiency is around 70% of the Carnot efficiency.

Keywords: R718, Energy Efficiency, Superheated Steam Drying, Turbomachinery, COP

1. INTRODUCTION

The limitation of global warming and the related reduction of greenhouse gas emission forces the thermal process industry to consider alternative, preferable renewable, energy sources in order to substitute fossil energy carriers. This development is, among others, driven by the outcome of the Paris climate conference in 2015 and the expected impact on legislative and national regulation. With this background the market potential of high temperature heat pumps is increasing, however industrial phase in of this technology is currently limited by the low technological readiness level of heat pumps operating with a heat sink of higher than 100°C (Elmengaard et al, 2017, IEA, 2014) and the relative high investments costs (when compared with fossil driven boiler/burner technology). The potential for implementing high temperature heat pumps at process temperatures between 100 °C and 150 °C was estimated to be 175 TWh for just Europe (Wolf, 2014).

Mechanical Vapour Recompression (MVR) is a special sub-application of heat pumps in which the evaporator is replaced by a thermal process which supplies low-pressure excess steam. This excess steam is then compressed to a higher pressure so that its condensation energy can be utilized at a higher temperature level so that it can be used re-heat a process fluid. Distillation columns, Evaporators, pasteurization processes and steam-driers are ideal processes for the MVR implementation. The concept is illustrated on the example of a superheated steam drier in Figure 1.

The application of MVR for these processes can reduce the specific energy consumption by over 75%. However, MVR technology is normally only applied for thermal capacities higher than 10 MW (Bantle et al, 2017), because the specific investment costs (per kW installed capacity) for smaller systems can be as high as 1000€/kW. It is outlined in Elmegaard (2017) that heat pump systems require investment costs between 100 – 200 €/kW in order to be an economic feasible alternative for the industry. The steam compressor technology

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is currently the main cost component in MVR and for thermal process with an capacity between 500 kW – 4 MW a cost efficient compressor technology is not available at the market (Elmegaard, 2017).

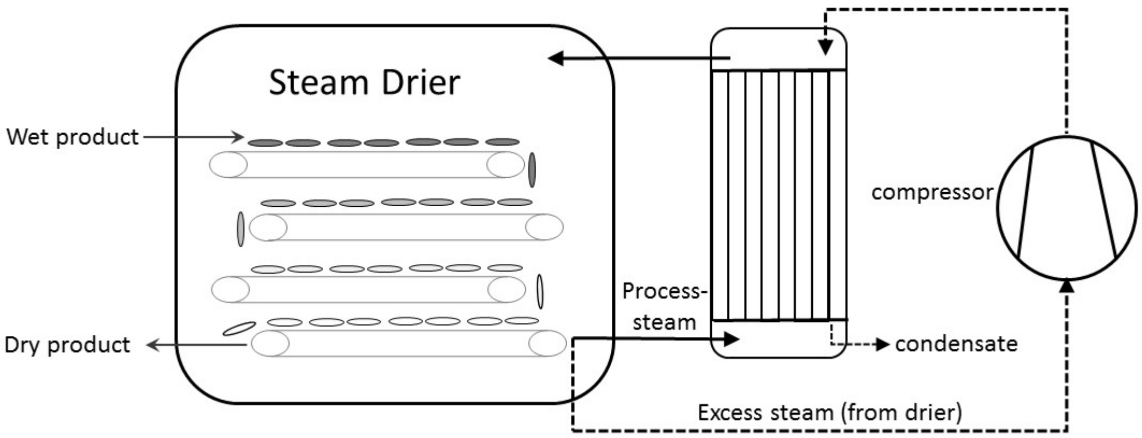

Figure 1: Simplified schematic layout for superheated steam drier with energy recovery through mechanical vapour recompression.

The turbo-compressors technology is considered as a potentially cost-effective and efficient alternative to conventional compressors (Bantle et al., 2014; Weel et al., 2013). Turbo-compressors are used in nearly every diesel vehicle for air compression. Bantle et al. (2017) demonstrated that the operating conditions were stable for steam compression. At a rotational speed of 90 000 rpm a standard automotive unit of a turbo-compressor achieved a pressure ratio of 2.4 with an isentropic efficiency of 72% and a mass flow of 450 kg/h. The measured efficiency was close to) the expected performance (3%-points lower) at the design point of the unit. The COP of MVR equipped with this prototype would be 11.5, however the temperature increase (between thermal process and the condenser) would be limited to around 23 Kelvin. It must be outlined that such a performance could be also achieved with conventional compressor technology, but the return of investment for the turbo-compressor system will be less than a year even for countries with a high price ratio between electricity and fossil fuel/gas (e.g. Germany).

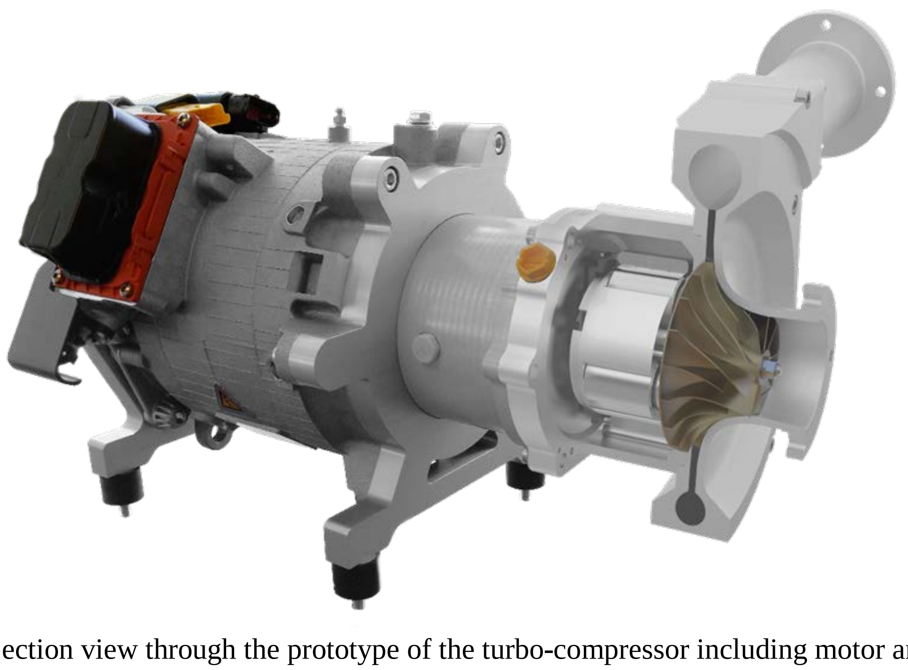

Figure 2: Section view through the prototype of the turbo-compressor including motor and gearbox.

For this study a two stage turbo-compressor system was investigate with the aim to demonstrate the performance of two identically constructed prototypes when operated in series. The basic structure of the

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prototype is described in this article, the experimental setup and the results of the test are included and future challenges are discussed.

2. METHODS

2.1 Design of the turbo-compressor prototype

The turbo-compressors used in this experiment are 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. At the design point N_{ref} =68000 rpm, the volume flow of air on the low-pressure side is Vc_{ref} = 0.27m^3/sec at T_{ref} = 288.15 K and p_{ref} = 1.013 bar. The compression ratio achieved is PR_{ref} = 2.72.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 kilowatts. 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.

Two identical turbo-compressors were installed in series in order to achieve a 2-stage compression with the aim to reach a pressure ratio of 3. The theoretical design conditions can be found in Table 1 and the aim to investigation was to evaluate if these conditions can be achieved using identical impeller designs.

Table 1: Operation parameter for first and second stage with identical impeller design.

	Impeller 1: Low stage	Impeller 2: High stage
Inlet conditions:	$\dot{m}_{\mathrm{in}} = 0.081 - 0.116$ kg/s $\dot{V}_{\mathrm{in}} = 499 - 734$ m^3/h $p_{\mathrm{in}} = 1.02$ bar ± 0.01 $T_{\mathrm{in}} = 115 - 120$ °C	$\dot{m}_{\mathrm{in}} = 0.085 - 0.125$ kg/s $\dot{V}_{\mathrm{in}} = 401 - 497$ m^3/h $p_{\mathrm{in}} = 1.36 - 1.65$ bar ± 0.01 $T_{\mathrm{in}} = 120 - 130$ °C
Rotational speed:	$n = 54000 - 72000$ rpm	$n = 54000 - 72000$ rpm
Pressure ratio (max):	PR = 1.7	PR = 1.7
Expected isentropic efficiency:	$\eta_{\mathrm{is}} = 0.80 - 0.75$	$\eta_{\mathrm{is}} = 0.74 - 0.70$

2.2 Test-setup for the experiments

The two turbo-compressors were installed in an MVR heat pump unit in which it was possible to by-pass the condenser (see Figure 4). The inlet of the first stage turbo-compressor is connected to an MVR-condenser which is operated as steam supply unit. The pressure in the condenser is approximately at atmospheric condition since the purge valve is open to the ambient. Temperature, pressure, mass flow and oxygen content is measured at the inlet and outlet of each compressor. Between the first and second stage a controlled amount of water is injected in order to de-superheat the working fluid. Water injection is also used to de-superheat after the expansion valve in order to reduce the thermal stress to the condenser. All piping work was constructed in a way that the thermal expansion is compensated during start-up and operation, so that the mechanical stress to the compressor is reduced to a minimum. The piping is insulated by 50mm mineral wool (k=0.05 W/m·K).

Both compressors are controlled by DC-Inverters, seperate 3-phase power meters were installed upstream each DC-Inverter. Gearbox and motor of the turbo-compressors are cooled by water and an energy-flow meter monitored the required cooling. An internal steam cooler in the MVR-condenser cooled the total system by de-superheating the working fluid before the inlet to the first stage. Also, here the energy flow is recorded. The

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inlet temperature to the first and second turbo-compressor was controlled to 10 K superheat, depending on the pressure. The system is fully automated and controlled by a self-developed software program.

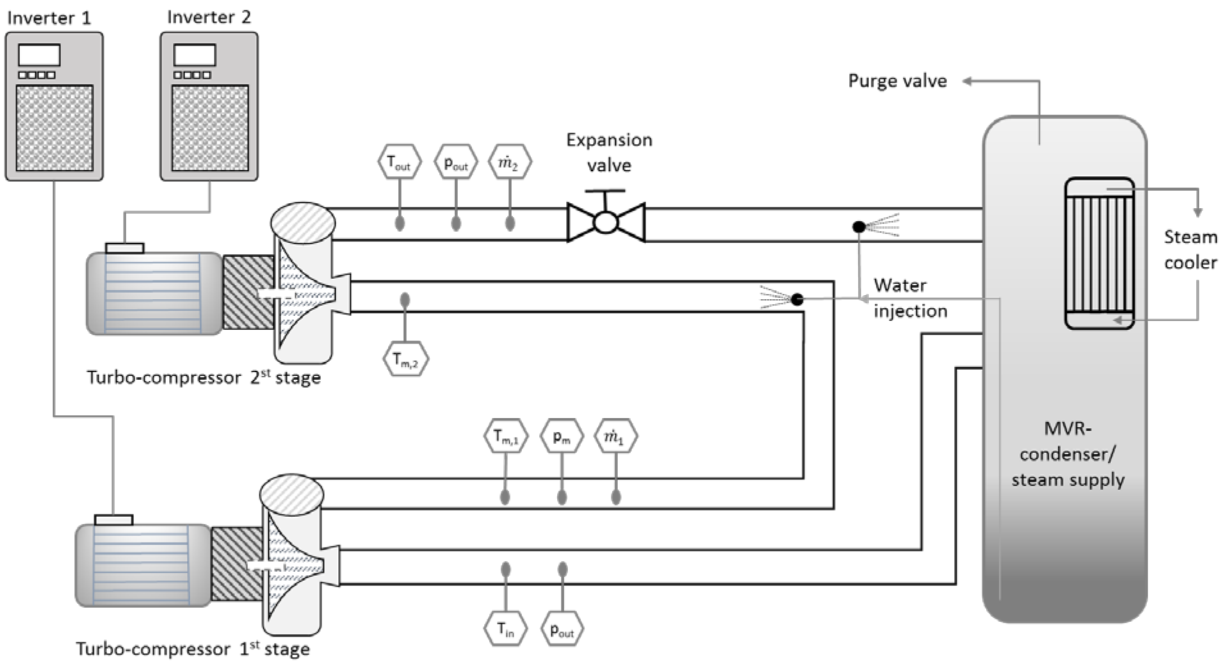

Figure 3: Principle setup of the 2 stage turbo-compression test son the MVR heat pump system.

2.3 Start-up of the system

The impeller of the turbo-compressor is quite sensitive to particle or droplet impact at higher speed and the inlet conditions of the working fluid need therefore to be superheated in order to avoid liquid droplets. Hence the system cannot be started with saturated steam and needs therefore to be heated to above 100°C (at atmospheric pressure) before water/steam is injected. Since the turbo-compressors are originally designed for air the start-up or preheating of the system can therefore be performed with air as working fluid. The turbo-compressors are hereby started when the system is dry and filled with ambient air. The system is then heated up the excess heat of the compressors until the coldest point (inlet to first compressors) reaches a stable temperature above 100°C. Then water is injected through the de-superheaters which is immediately evaporated and filling the system with superheated steam by replacing the air through the purge valve of the MVR-condenser. The start-up procedure takes around 45 minutes and after that the system is completely filled with steam (oxygen content <0.1%).

2.4 Performed tests

For this study the turbo-compressor speed in both units was set to 54000 rpm, 63000 rpm and 72000 rpm in order evaluate and achieve an overall pressure ratio of 3. In order to compare the conducted tests with the performance predicted the following correction were performed:

Relative flow correction:

$$V_C = \frac{\dot{V}_{in}\sqrt{ZRT/Z_{ref}R_{ref}T_{ref}}}{\dot{V}c_{ref}\ (p/p_{ref})}$$

Reduced ratio:

$$PR_C = \frac{PR-1}{PR_{ref}-1}$$

Relative speed correction:

$$N_C = \frac{N\sqrt{ZRT/Z_{ref}R_{ref}T_{ref}}}{N_{ref}}$$

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Here the reference conditions as described in paragraph 2.1 were applied. The air gas constant and compressibility was R_{air}=287.05 J/kgK and R_{air}=1, respectively. The relevant superheated steam properties were calculated using REFPROP version 9.

3. RESULTS

The MVR heat pump with 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 to make the necessary fine adjustment. Then the compressor was operated at full load (60-80% of maximum capacity). In Table 2 the operation conditioned are summarized.

Figure 4 shows the achieved operational conditions of the turbo-compressors during the pilot tests. The points illustrated in the figure were held for at least 2 hours and the isentropic efficiency is based on the average compressor performance during this time. The achieved mass flow at the highest capacity was 420 kg/h in the first stage and the isentropic efficiency of the turbo-compressor was constant at 73%. In the second stage the mass flow increased by the water injection from approximately 306 kg/h at 54000 rpm to 450 kg/h at 75000 rpm; here the isentropic efficiency decreased from 75% to 70%.

Table 2: Operating conditions of two stage MVR heat pump

	Impeller 1: Low stage	Impeller 2: High stage
60%		
Inlet conditions:	$\dot{m}_{\mathrm{in}} = 0.081$ kg/s	$\dot{m}_{\mathrm{in}} = 0.085$ kg/s
	$\dot{V}_{\mathrm{in}} = 499\ \mathrm{m^3/h}$	$\dot{V}_{\mathrm{in}} = 401\ \mathrm{m^3/h}$
	$p_{\mathrm{in}} = 1.02$ bar ± 0.01	$p_{\mathrm{in}} = 1.36$ bar ± 0.01
	$T_{\mathrm{in}} = 115$ °C	$T_{\mathrm{in}} = 120$ °C
Outlet conditions:	$p_{\mathrm{out}} = 1.41$ bar ± 0.02	$p_{\mathrm{out}} = 1.87$ bar ± 0.02
	$T_{\mathrm{out}} = 152$ °C	$T_{\mathrm{out}} = 161$ °C
Rotational speed:	$N = 54000$ rpm	$N = 54000$ rpm
Pressure ratio:	PR = 1.38	PR = 1.38
Isentropic efficiency:	$\eta_{\mathrm{is}} = 0.78$	$\eta_{\mathrm{is}} = 0.75$
Electric power consumption:	$P_{\mathrm{el}} = 9.6\ kW$	$P_{\mathrm{el}} = 9.9\ kW$
70%		
Inlet conditions:	$\dot{m}_{\mathrm{in}} = 0.098$ kg/s	$\dot{m}_{\mathrm{in}} = 0.105$ kg/s
	$\dot{V}_{\mathrm{in}} = 613\ \mathrm{m^3/h}$	$\dot{V}_{\mathrm{in}} = 455\ \mathrm{m^3/h}$
	$p_{\mathrm{in}} = 1.02$ bar ± 0.01	$p_{\mathrm{in}} = 1.48$ bar ± 0.01
	$T_{\mathrm{in}} = 116$ °C	$T_{\mathrm{in}} = 121$ °C
Outlet conditions:	$p_{\mathrm{out}} = 1.55$ bar ± 0.02	$p_{\mathrm{out}} = 2.26$ bar ± 0.02
	$T_{\mathrm{out}} = 168$ °C	$T_{\mathrm{out}} = 180$ °C
Rotational speed:	$N = 63000$ rpm	$N = 63000$ rpm
Pressure ratio:	PR = 1.53	PR = 1.53
Isentropic efficiency:	$\eta_{\mathrm{is}} = 0.76$	$\eta_{\mathrm{is}} = 0.69$
Electric power consumption:	$P_{\mathrm{el}} = 14.3\ kW$	$P_{\mathrm{el}} = 15.1\ kW$
80%		
Inlet conditions:	$\dot{m}_{\mathrm{in}} = 0.116$ kg/s	$\dot{m}_{\mathrm{in}} = 0.125$ kg/s
	$\dot{V}_{\mathrm{in}} = 734\ \mathrm{m^3/h}$	$\dot{V}_{\mathrm{in}} = 497\ \mathrm{m^3/h}$
	$p_{\mathrm{in}} = 1.02$ bar ± 0.01	$p_{\mathrm{in}} = 1.65$ bar ± 0.02
	$T_{\mathrm{in}} = 120$ °C	$T_{\mathrm{in}} = 130$ °C
Outlet conditions:	$p_{\mathrm{out}} = 1.73$ bar ± 0.01	$p_{\mathrm{out}} = 2.79$ bar ± 0.02
	$T_{\mathrm{out}} = 191$ °C	$T_{\mathrm{out}} = 205$ °C
Rotational speed:	$N = 72000$ rpm	$N = 72000$ rpm
Pressure ratio:	PR = 1.69	PR = 1.69
Isentropic efficiency:	$\eta_{\mathrm{is}} = 0.73$	$\eta_{\mathrm{is}} = 0.70$
Electric power consumption:	$P_{\mathrm{el}} = 21.0\ kW$	$P_{\mathrm{el}} = 22.5\ kW$

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The compressor map in Figure 4 shows the corrected pressure ratio as a function of the relative corrected volume flow. For both stages an almost linear trendline of the performance from 52000 rpm to 72000 rpm is achieved. The trendline in the second stage has a larger gradient compared to the first stage. This is caused by the high-density change of water-vapor with increased pressure.

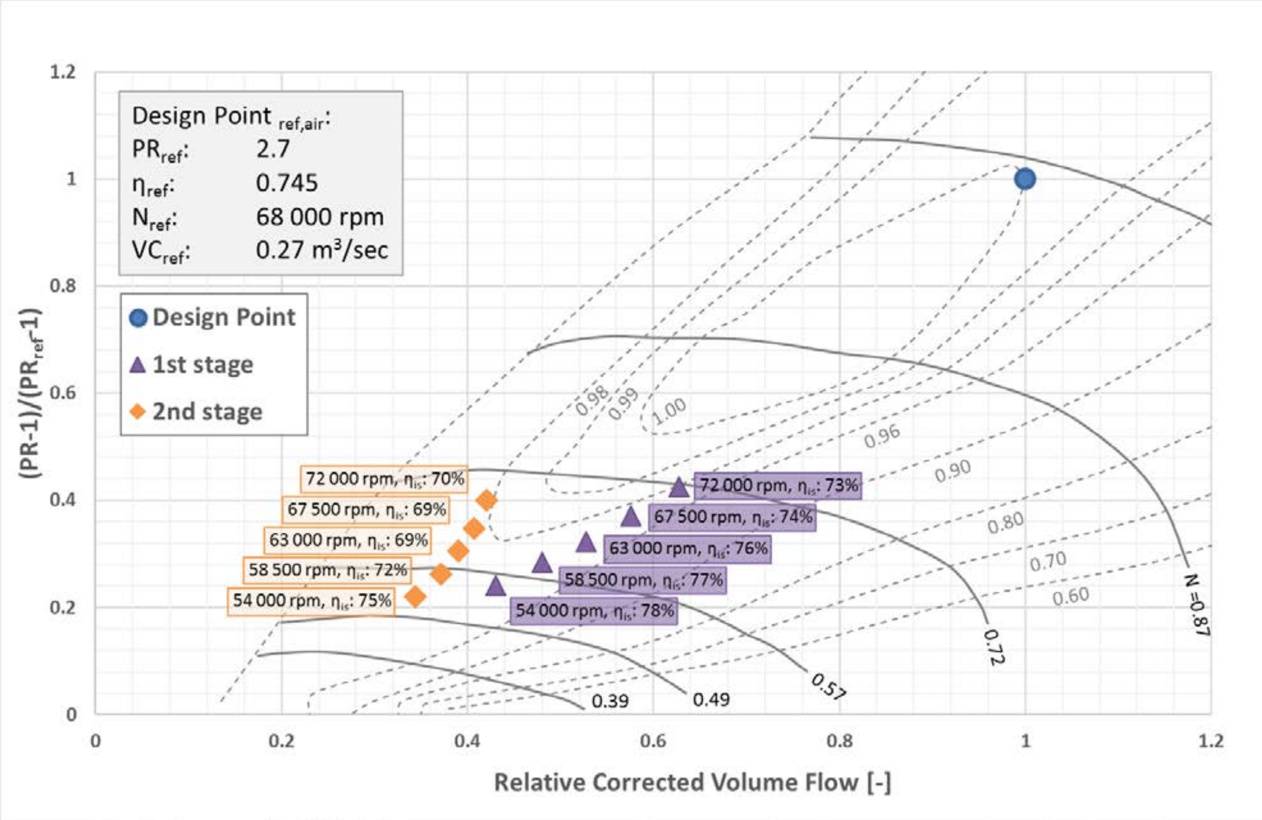

Figure 4: Achieved, stable operational conditions of the turbo-compressors in the first and second stage during the pilot-tests.

4. DISCUSSION

The performed tests demonstrated that it is possible to achieve stable operation conditions in the investigated setup and with the used of identical turbo-compressors. The impeller speed was the same for both turbo-compressors and was never higher than 80% of the maximum design speed. The achieved overall pressured ratio was around 2.8 and resulted from pressure ration of 1.69 and 1.69 in the first and second stage respectively. It must be noted that the volume flow in the second stage was significantly lower compare to the first stage even when the mass flow was increased by water injection after the first stage. This is a result of the specific volume of steam which is reduced with increased pressure. As a consequence, the second turbo-compressor operated closer to the surge line of the unit. Ideally the design of the impeller should be adjusted to the targeted pressure and volume flow in each stage. However, that would require a more modular construction of the turbo-compressors.

The performance of each compressor from 54000 rpm to 72000 rpm seems to have a clear trendline in the compressor map. However, it is difficult to predict a trend based on five points only. On the other hand, the identified trend helps to predict if the system would be able to operate stable at high speed. It seems reasonable to assume that the system could be also operated at a speed of 81000 rpm or even 90000rpm. At this point a pressure ratio of 4 (at 81000 rpm) and a pressure ratio 5.5 (at 90000 rpm) could be achieved. That would also mean that the second stage turbo-compressor would operate close to the surge condition of the system and it is currently difficult to estimate if the system can be operated stable at such a high-pressure ratio.

The achieved pressure ratios were almost identical in stage one and stage two, since no pressure control device is installed between the two units. The energy input in the second stage was on the other hand slightly higher (up to 10 % increased) when compared to the energy consumption of the first stage. This might be caused by the fact that the two turbo-compressors were operation at different areas in the compressor map and that the second compressor had an increase mass flow due to the water injection.

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In the compressor map it can be seen that the constant speed lines of the turbo-charger are flat and almost horizontal in some parts of the map. This results in a quite sensitive surge-behaviour of the turbo-compressors even when they are operated in a stable point "in the middle" of the map. Slight pressure change can cause the system to turn fast into surge-operation in which the turbo-compressors no longer operate stable. It is therefore good advice to operate the system closer to the choke area of the compressor map.
The applied de-superheating concept was based on the injection of water and required a precise control of the added amount of water in order to prevent sub-cooling and droplet formation. The disadvantageous of such water spraying systems is that the evaporation time of the water droplets require a relative long pipe length. Hence the system has special space requirements in industrial applications and while the turbo-compressors are compact and small the required pipe work could limit the industrial implementation. For an industrialized solution it might therefore be necessary to investigate more compact but still cost effective de-superheating concepts.
The used cooling oil (Rotrex, SX 150) in the gearbox has a maximum allowed temperature of 140°C. In the investigation operation the oil temperature never exceeded 130°C in the second compressor. However, with increased impeller speed it will be possible to increase the performance of the system even more, but currently it seems that the limiting factor of the turbo-compressor system is the oil temperature. Improved oil cooling could be achieved by a larger oil pump.
During the operation of the turbo-compressors a very small amount of steam was penetrating into the gearbox of the turbo-compressor. This created an emulsion of the cooling oil and could potentially result in system failure. However, since the temperature of the cooling oil reaches more than 100 °C the water was evaporated out of the emulsion in the oil container. It was also observed that the cooling was improved once a small amount of water was present in the oil. The pure oil was analysed after the test and did not show degradation or other negative impact. Fur future applications it must therefore be evaluated if it is necessary to improve the sealing between gearbox and impeller room even more or if a certain water penetration can be accepted. The performed investigation indicate that a certain steam penetration is acceptable.
The COP of the performed investigation is 7.8, when the achievable condensation energy is compared to the total amount of energy supplied to the inverters. This COP value includes the losses in the inverter, motor and gearbox. Without these losses the COP is 9.4, which is the COP when only the isentropic losses of compression are considered. The ideal Carnot COP for the investigated temperature range is 13.4, which means that the isentropic efficiency is around 70% of the Carnot efficiency and the system efficiency (including losses of inverter, motor and gearbox) is still 58% of the Carnot efficiency This outlines of course also the general advantage of R718 as working fluid.
Cost estimations for industrial systems based on the presented technology are at the current technological readiness level only rough estimates and need to be verified further. However, the investments costs in the turbo-compressor systems are dominated by the costs for inverter and motor. Based on the unit costs for this investigation the compressor costs can be estimated to around 50€/kW installed condenser capacity. The costs for the complete MVR heat pump can be estimated to be approximately 200 €/kW; this estimation is based on the costs for the current investigation. Further developments will focus on increasing the size of the turbo-compressor and it is expected that investment costs for a complete heat pump system can be reduced further, when more standardized technology (e.g. for de-superheating) is implemented. However, even with the current cost situation the technology results in acceptable return of investment periods also for countries with high electricity to gas prize ratios.

5. CONCLUSIONS

Two identical turbo-compressor prototypes were used to test two stage vapor compression for the application in an MVR heat pump. The development has the potential to be a very cost-efficient compression technology for industrial high temperature heat pumps. The turbo-compressor of the first stage reached a pressure ratio of 1.69 and is designed for a mass flow of 400-600 kg/h superheated steam. The second stage turbo-compressor had an identical design and achieved a pressure ration of 1.69. Between compression stage one and two de-superheating is applied by water injection. With the developed system it is possible to compress superheated steam from atmospheric pressure to 2.8 bar, where it can be condensed at a temperature of 131°C. The COP of the performed investigation was 7.8, when the achievable condensation energy is compared to the total amount of energy supplied to the system. The compressor efficiency is around 70% of the Carnot efficiency.

The investigation showed that the two turbo-compressors were operated at different areas in the compressor map and each compressor showed a clear trendline. Based on this it seems reasonable to assume that a higher performance can be achieved with the current prototype system. However, the limiting factor is the danger of unstable operation especially of the second compression stage close to the surge condition. Another limitation might be the temperature limitations and stability of the cooling oil in the gearbox.
The costs for the complete MVR heat pump can be estimated to be approximately 200 €/kW; this estimation is based on the costs for the current installation. Further developments will focus on increasing the size of the turbo-compressor and it is expected that investment costs for a complete heat pump system can be reduced further, when more standardized technology (e.g. for de-superheating) is implemented. However, even with the current cost situation the technology results in acceptable return of investment periods also for countries with high electricity to gas prize ratios.

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