Manuscript ID: 973
DOI:10.18462/iir.icr.2019.0973

Turbo-compressors for R-718: Experimental evaluation of a two-stage steam compression cycle

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ABSTRACT

Water (R-718) is a safe and energy-efficient refrigerant. Mechanical vapour recompression (MVR), an open-loop heat pump using R718, can significantly reduce the energy consumption for steam-heated processes like drying, pasteurization, evaporation or distillation. However, the existing compression technology is not cost-efficient, especially in the capacity range from 500 kW to 4 MW. Therefore, a novel two-stage turbo-compressor system, developed for application in industrial superheated steam drying and based on mass-produced automotive turbocharger technology, was developed. Its performance was evaluated in a test facility, showing that it is possible to compress superheated steam from atmospheric pressure up to 3 bar, delivering 300 kW at 133°C, with a COP of 5.9, an isentropic efficiency of 74% and a Carnot efficiency of 48%. With an estimated investment cost of 150 €/kW installed heating capacity, the system clearly has the potential of being a cost-effective solution for heat recovery in steam-heated industrial processes.
Keywords: R718, turbo-compressor, mechanical vapour recompression, heat pump, steam drying

1. INTRODUCTION

Industrial heat pumps available on the market today normally operate with a heat delivery (heat sink) temperature of not more than 70 °C - 100 °C, depending on technology and refrigerant applied. In this temperature range, air or water is often used as a secondary heat carrier. There are, however, numerous industrial processes with heat demands at temperatures above 100 °C, such as drying, evaporation, pasteurization, sterilization or distillation, for which steam is used as a heat transport medium (Elmegaard et al., 2017).
Industrial drying processes require intensive use of energy. Estimates show that 12 % to 25 % of the national industrial energy consumption in developed countries is attributed to industrial drying, with limited utilization of waste heat streams. There is an accordingly large potential for more efficient and environmentally friendly technologies within industrial drying processes. By integrating a heat pump system, the specific energy demand can be reduced with up to 80% (Sannan et al., 2016., Kang and Ryu, 2016).
Heat pumps operating with heat sinks above 100 °C are normally classified as high-temperature heat pumps (HTHP). The choice of refrigerant for a HTHP is limited. Water (R718) is one of the few alternatives that is not only thermodynamically effective but also environmental-friendly, non-flammable, non-toxic and cost-effective (Larminat et al., 2014). Since water vapor (steam) is used as energy source in numerous plants its industrial acceptance is very high. Disadvantages include high specific volumes, as well as a large volume change and superheating during compression.
Mechanical vapour recompression (MVR) is an open-loop heat pump technology using R718. The MVR technology has the potential of significantly reducing the energy consumption for steam-heated processes. However, MVR is today normally applied only for thermal capacities higher than 10 MW because the specific investment costs (cost per kW installed capacity) for smaller systems can be as high as 1000€/kW (Bantle et al., 2018). To be an economically feasible alternative for the industry

an investment cost of 100 – 200 €/kW is required (Elmegaard et al., 2017). Steam compressor technology is today the main cost component in MVR systems and for capacities between 500 kW and 4 MW there is no cost-efficient technology available on the market (Elmegaard et al., 2017). Turbo-compressor technology is however considered as a potentially cost-effective and efficient alternative to conventional compressors in this capacity range. High impeller speeds allow a compact design with low material consumption. Turbo-compressors are normally used for compression of air and about 95% of all diesel cars are equipped with a "turbocharger" (Weel et al., 2013).

There are a several research projects focused on developing cost-effective R718 compressors for use in HTHPs. Larminat et al. (2014), Chamoun et al. (2014) and Madsboell et al. (2015) all present compressor technology development to supply 500 kW - 700 kW heat at a condensing temperature of 130 °C. Meroni et al. (2018) compared different HTHP cycles and compressor designs for steam generation at 150 °C and conclude that direct compression of steam (i.e. MVR) in a two-stage cycle out-performed the closed loop cascade cycles using other refrigerants. Zühlsdorf et al. (2018) suggests design recommendations for two-stage R718 turbo-compressors, to obtain lower investment costs and efficient de-superheating.

In the present work, a system with two-stage turbo-compression of steam was conceptualized, constructed and evaluated in a test facility. The objective is to investigate the potential of using an MVR heat pump with a turbo-compressor system based on mass produced components from the vehicle industry, in order to find a cost-efficient solution for waste heat recovery in industrial processes such as steam drying. By performing measurements with varying impeller speeds and mass flow rates the compressor performance map and system efficiency are determined, enabling an assessment of operating area and energy efficiency. In section 2, the test facility and experimental procedure are presented followed by a description of the method used for performance evaluation in section 3. The results are presented, discussed and concluded in section 4 and 5.

2. EXPERIMENTAL SETUP

2.1 Test facility system

In Fig. 1 the essential components and measuring points of the test facility are presented. The two-stage turbo-compressor system is designed for an industrial steam dryer providing up to a ton of excess steam per hour at atmospheric pressure and 10-20 K superheat. Drying conditions are simulated by means of a steam generator and steam cooler.

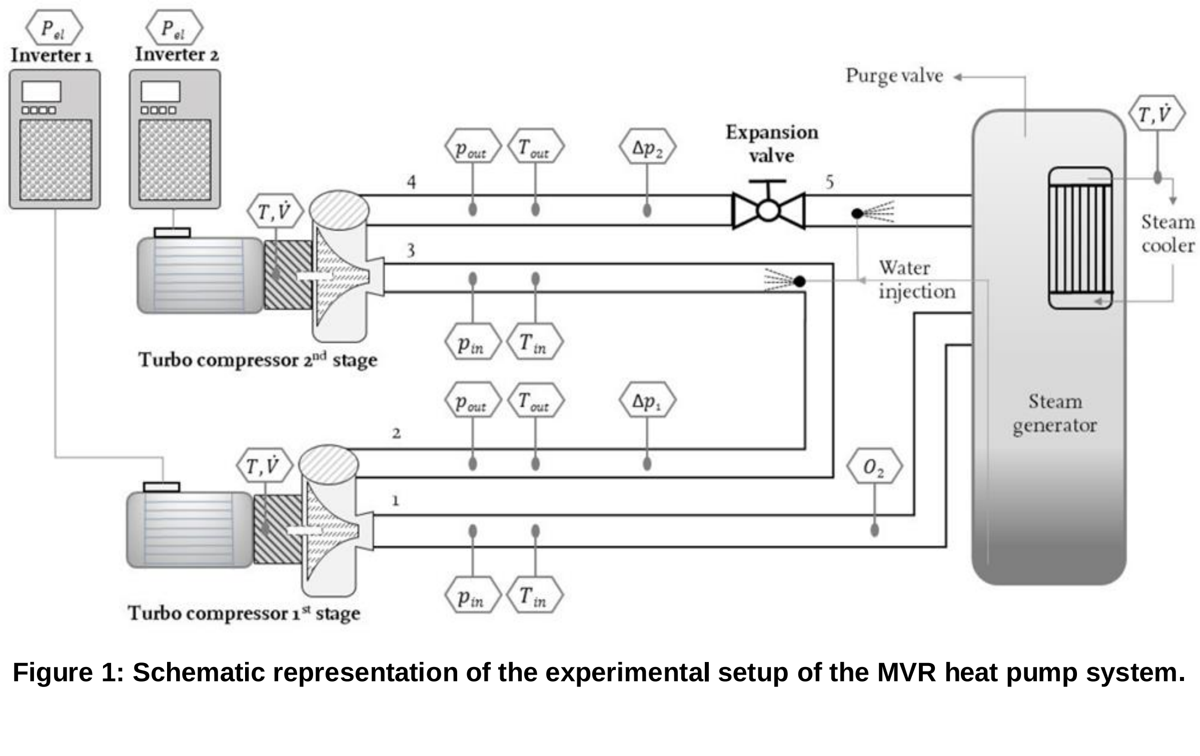

Figure 1: Schematic representation of the experimental setup of the MVR heat pump system.

The inlet of the first turbo compressor stage (1) is connected to the steam generator. After the first compression stage (1-2) a controlled amount of water is injected to reduce the steam superheat to around 20 K before the second stage inlet (2-3). After the 2^{nd} compression stage (3-4) the pressure of the superheated steam is reduced to atmospheric conditions in the expansion valve (4-5). To minimise thermal stress on the steam generator, water injection is applied after the expansion valve. A steam cooler connected to the steam generator cools the system (i.e. simulates the dryer) by de-superheating the steam before it enters the 1^{st} compression stage (5-1) at atmospheric pressure and 20 K superheat. To minimise mechanical stress of the compressor, all piping work was constructed to ensure compensation for thermal expansion during start-up and operation. The piping is insulated by 50 mm mineral wool (k=0.05 W/m·K). DN100 piping was used to minimize system pressure drop.

2.2 Turbo-compressor setup

Two identical turbo-compressors are installed in series to achieve a total pressure ratio of 3. A planetary gearbox, enabling a high transmission ratio of 7.5 with a mechanical efficiency of 98%, is mounted on the drive shaft of each turbo-compressor. The planetary gearbox is equipped with an internal oil pump and an external water-cooled oil cooler. A 650 V water cooled DC-motor is placed directly on the drive shaft of the gearbox, enabling a rotational speed of up to 90 000 rpm at the impeller. The motor is controlled by an inverter which can deliver 59 kW at an efficiency of 97%.
The turbo-compressor prototype is a further development of a conventional radial turbocharger from the automotive industry (i.e. designed for air), adapted by the manufacturer for use in steam compression systems. The modifications aimed at achieving a larger pressure ratio at a high isentropic efficiency and improving the durability for long-term continuous operation. The modified prototype includes an impeller designed in titanium with a diameter of 100 mm (the rest of the casting is in alumina) and a reinforced sealing between compression chamber and gear box.
It represents a compact and light-weighted design, with a total weight of the complete unit (turbo-compressor, gearbox, DC-motor) of about 40 kg and dimensions of 50 cm in length, 40 cm in width and 35 cm in height. More detailed design conditions are presented in Bantle et al. (2018). Based on costs for this turbo-compressor set-up the MVR heat pump investment cost is estimated to around 150 €/kW installed condenser capacity.

2.3 Data aquisation and measurement uncertainity

Steam temperature, pressure, mass/volume flow rate and oxygen content are measured at different points in the experimental facility, as shown in Fig. 1. Electric power of each inverter is recorded, as well as cooling capacity of the steam cooler and the respective water coolers, and the amount of injected cooling water. The accuracy of the sensors is between ±0.5 to ±0.75%, and the sampling rate of the data acquisition system is 1-5 seconds.
Measurement uncertainties are determined in accordance with ISO-5167-1:2003 and ISO-5167-2:2003, including primarily instrumental inaccuracy. For a pressure ratio of above 1.5 the uncertainty is less than 1.5 %. However, by error propagation, the calculated isentropic efficiency may vary by up to 5%, mainly due to the measuring inaccuracy of the pressure sensors. At higher pressures the uncertainity is significantly reduced.

2.4 Experimental procedure and operating conditions

Since the impeller is sensitive to droplets the system cannot be started with saturated steam. Instead it is started in an air atmosphere and heated up to a temperature of 110 °C, with excess heat from the compressor. Water is then injected through the de-superheaters and immediately evaporated, filling the system with superheated steam by replacing the air through the condenser purge valve. The system is considered completely filled with steam when the oxygen content is less than 0.1%.
Variation in operating points is achieved by changing the impeller speed of the two stages and changing the expansion valve opening posistion. An operating point is defined as stable when measurement variance over a period of 5 minutes is less than determined measurement uncertainty. The maximum investigated impeller speed was 81 000 rpm for the 1^{st} stage and 72 000 rpm for the 2^{nd} stage. Full speed of 90 000 rpm was not reachable due to a too high temperature of the gear box cooling oil. Impeller speeds below 54 000 rpm were not analysed since the corresponding operating conditions are industrially irrelevant due to the small pressure ratio achieved.

3. PERFORMANCE EVALUATION METHOD

3.1 Compressor performance map

A compressor performance map is a commonly used graph to represent, compare and analyze the performance at various operating conditions. Fig. 2 shows a typical performance map for turbo-compressors illustrating the relationship between mass/volume flow rate, pressure ratio, isentropic efficiency and impeller speed. Unstable operating points are represented by the "surge line" and "choke line". Surge behaviour occurs when the fluid flows back from the high-pressure to the low-pressure side, while choke behaviour is characterised by a rapid drop in pressure ratio.

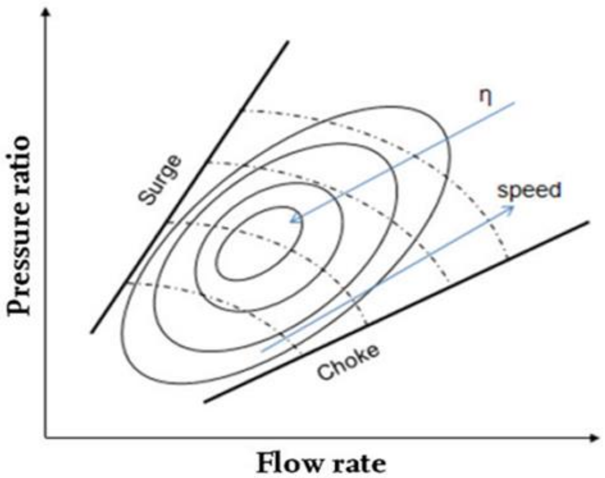

Figure 2: Schematic representation of a typical performance map for a turbo compressor

The determination of the performance map for the investigated turbo-compressor system was made as follows. All measured operating points are calculated back to a standardised reference point (i.e. a defined pressure and temperature). This is especially important for steam (R718) whose density is strongly reduced with reduced pressure. It is also recommended to adjust the impeller speed to correct for temperature influence (Hafaifa et al., 2014). These corrections enable a comparison of different operating points in the same performance map. Otherwise, the respective input parameters, e.g. temperature, pressure and flow rate, must be indicated for each point in the map.
Isentropic efficiency (η_{is}) is calculated from the ratio of enthalpy difference over the compressor for isentropic and real operating conditions (given from temperature and pressure measurement at inlet and outlet);

$$\eta_{is} = \frac{h_{out,is} - h_{in}}{h_{out} - h_{in}} \qquad \text{Eq. (1)}$$

Pressure ratio (Π_t) is given from the ratio of discharge and suction pressure of each compressor stage;

$$\Pi_t = \frac{p_{t\,out}}{p_{t\,in}} \qquad \text{Eq. (2)}$$

The corrected mass flow rate ($\dot{m}_{red}$) in Eq. 3 is achieved by calculating the mass flow rate back to the suction side reference point (van Essen, 1998), which was set to 1.013 bar and 120°C (corresponding to typical dryer conditions). In addition, the temperature measurements are corrected for heat losses (T_{hl}) between the measuring point and the actual compressor inlet/outlet.

$$\dot{m}_{red} = \dot{m}_t \cdot \frac{\sqrt{\left(T_{hl\,in}/T_{ref}\right)}}{\left(p_{t\,in}/p_{ref}\right)} \qquad \text{Eq. (3)}$$

The corrected volume flow rate is then given by using the density of R718 at the reference point (ρ_{ref} = 0,565 kg/m^3);

$$\dot{V}_{red} = \frac{\dot{m}_{red}}{\rho_{ref}} \quad \text{Eq. (4)}$$

The corrected impeller speed (N_{red}) is given from Eq. 5 where N_{set} is the set speed at the inverter.

$$N_{red} = \frac{N_{set}}{\sqrt{(T_{hl\,in}/T_{ref})}} \quad \text{Eq. (5)}$$

The speed corrected pressure ratio (Π_{rel}), mass and volume flow rate ($\dot{m}_{rel}$, $\dot{V}_{rel}$) are then calculated as:

$$\Pi_{rel} = \Pi_t \cdot \frac{N_{set}}{N_{red}} \quad \text{Eq. (6)}$$

$$\dot{m}_{rel} = \dot{m}_{red} \cdot \frac{N_{set}}{N_{red}} \quad \text{Eq. (7)}$$

$$\dot{V}_{rel} = \dot{V}_{red} \cdot \frac{\mathrm{N_{set}}}{N_{red}} \quad \text{Eq. (8)}$$

3.2 Energy efficiency

The coefficient of performance (COP) of the turbo-compressor system is defined as the ratio between condenser cooling capacity and total electricity consumption of the system;

$$COP = \frac{\dot{Q}_{cond}}{\sum \dot{P}_{el}} \quad \text{Eq. (9)}$$

The COP determined from Eq. 9 considers all losses in the system; such as pressure drop (piping, measuring instruments, water nozzles, heat exchangers), losses of inverter, motor and gear unit, heat losses of the complete plant and compression losses of both turbo compressors.
The maximum theoretical COP, COP_{Carnot}, is determined according to Eq. 10. $T_{cond.,HP}$ and $T_{cond.,LP}$ is the condensing temperature [K] on the high pressure and low pressure side, respectively.

$$COP_{Carnot} = \frac{T_{cond.,HP}}{T_{cond.,HP} - T_{cond.,LP}} \quad \text{Eq. (10)}$$

Finally, the maximum system efficiency, the Carnot efficiency, is given as the ratio between the measured COP and the maximum COP;

$$\eta_{Carnot} = \frac{COP}{COP_{Carnot}} \quad \text{Eq. (11)}$$

4. RESULTS

4.1 Experimental compressor performance map

A performance map of the two-stage turbo compressor system was determined based on 87 measured stable operating points and is shown in Fig. 3. It can be observed that the second stage (blue dots) operates more closely to the surge line and is thus limiting the operating area of the system. An identical impeller design was used for both stages, which is not an optimal solution due to the change in steam volume flow between the two stages. A smaller design of the 2nd stage impeller is needed to obtain stable operation with a good safety margin to the surge line. Fig. 3 also shows the maximum achieved pressure ratio for the two stages; 1.95 at 81 000 rpm for the 1st stage and 1.7 at 72 000 rpm for the 2nd stage. Due to the limitation of the gear box cooling oil temperature this is currently the highest possible pressure ratios that can be achieved in the test facility.

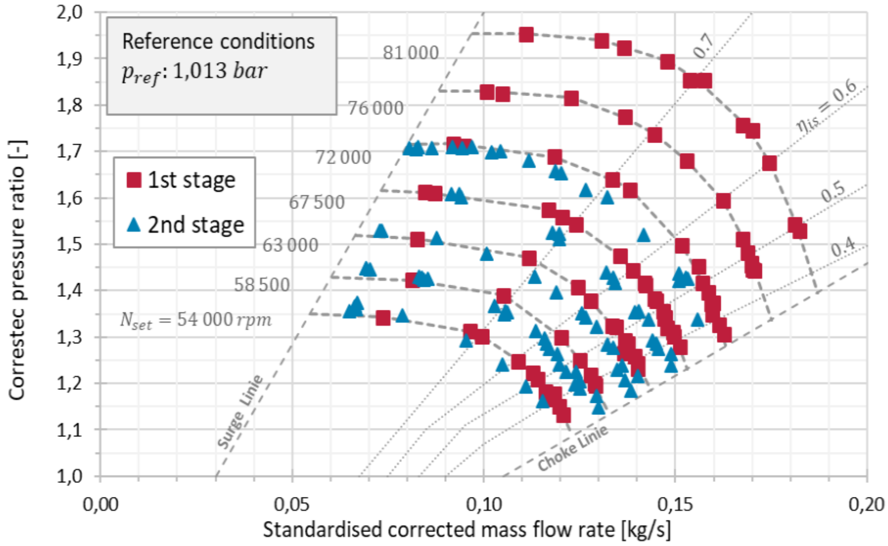

Figure 3: Experimentally determined performance map for the 1st and 2nd compressor stage

To analyze the system behavior when operating the two stages at either equal or unequal impeller speed, six operating points were selected for further evaluation. In Fig. 4, the three solid arrows (1-4) represent the connections between operating points for the 1st and 2nd stage at equal impeller speed, while the dashed arrows (5-6) represent the connections at unequal impeller speed.

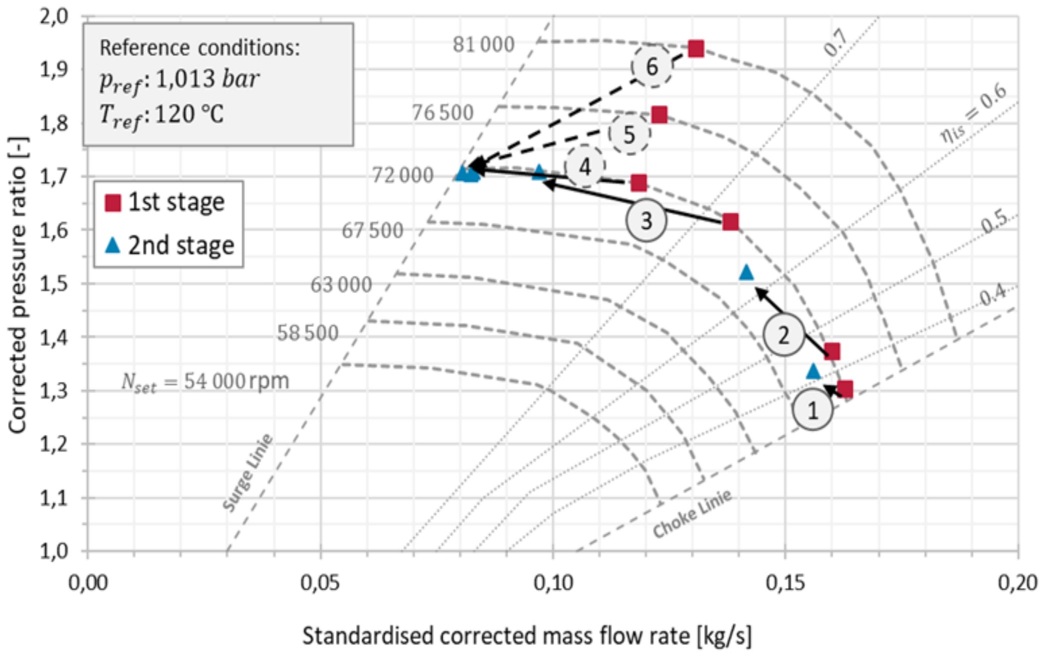

Figure 4: Dependencies (shown as arrows) between the 1st and 2nd stage for six operating points.

In point 1-4 both compressor stages are operated at 72 000 rpm. When then the expansion valve is successively more closed (moving left in the diagram), the pressure ratio and isentropic efficiency is improved for both stages, while the mass flow, and thus the condensing capacity, is reduced. In point 5 and 6 the impeller speed of the 1st stage is increased in relation to the 2nd stage. This leads to a larger pressure ratio for the 1st stage, while the mass flow remains almost unchanged. The operating point of the 2nd stage remains in a nearly constant position, close to the surge line, as when operating at equal impeller speeds (point 4). Both stages remain in the range of an isentropic efficiency of 0.72 to 0.74. With different impeller speeds it is therefore possible to operate the system with higher pressure ratio and flow rate (condensing capacity) as well as a higher isentropic efficiency.

4.2 System efficiency

Table 1 shows the results from the performance analysis of the six operating points (#1 - #6) presented in Fig. 4. For point 1-4 (operation at equal impeller speed) an increased pressure ratio results in both a lower condensing capacity and a lower COP. When operating the 1st stage with a higher impeller speed (points 5-6) the total pressure ratio and condensing capacity are increased at the same time. For the desired pressure ratio of 3.0, a condensing capacity of 300 kW (steam flow of 500 kg/h) was achieved at a COP of 5.9, and a Carnot efficiency of 48 %. Even if the compressors account for the largest losses the isentropic efficiency is above 70%, which in principle is a satisfactory result. More notable is the observed heat losses, contributing to above 20% of the total losses, despite the use of thermal insulation. However, the compressor outlet temperature can be up to 230 °C, i.e. a temperature difference of up to 200 degrees to the environment, while the insulation thickness is only 5 cm.

Table 1: Summary of performance analysis for the six selected operating points (shown in Fig. 4).

#	N_{set} (1/min)	$\dot{m}_t$ (kg/s)	Π_t (−)	T_{sat} (°C)	T_{lift} (K)	$\dot{Q}_{Kond.}$ (kW)	P_{el} (kW)	COP (−)	COP_{carnot} (−)	ηSystem
1.	72.000 – 72.000	0,167	1,4	110,4	10,4	373,1	52,9	7,1	36,8	19,2%
2.	72.000 – 72.000	0,164	1,8	116,6	16,6	362,5	52,4	6,9	23,5	29,3%
3.	72.000 – 72.000	0,141	2,5	127,7	27,7	306,8	47,4	6,5	14,4	45,1%
4.	72.000 – 72.000	0,125	2,6	129,5	29,5	272,3	42,1	6,5	13,6	47,8%
5.	76.500 – 72.000	0,130	2,8	131,8	31,8	282,4	45,8	6,2	12,7	48,7%
6.	81.000 – 72.000	0,138	3,0	133,5	33,5	299,5	50,7	5,9	12,1	48,8%

5. CONCLUSIONS

This paper presents an experimental investigation of a two-stage turbo compressor MVR system, consisting of two identical impellers and developed for application in industrial superheated steam drying. A performance evaluation shows that the novel system has a clear potential of being a cost-effective alternative to conventional vapour compression systems. With the possibility of using series-produced components from vehicle industry, the estimated investment cost for the MVR system is 150 €/kW installed heating capacity. Stable operating conditions were documented for impeller speeds up to 81 000 rpm for the 1st compressor stage and 72 000 for 2nd stage. The desired total pressure ratio of 3.0 was reached, with an isentropic efficiency of 74 % for both stages. For the application of an atmospheric superheated steam dryer a steam flow of 500 kg/h at a condensing temperature of 133 °C was achieved, delivering 300 kW heating capacity with a system COP of 5.9 and a Carnot efficiency of 48%. The 2nd stage operates close to the surge line, implying that the design of the impellers should be adjusted to the targeted pressure and volume flow conditions at each stage, to ensure stable operation of both stages near the optimum operation point.

ACKNOWLEDGEMENTS

This work is a part of the DryFiciency project (www.dry-f.eu). DryFiciency receives funding from the European Commission under the Horizon 2020 programme - grant number 723576.

NOMENCLATURE

Abbreviations

COP	Coefficient of performance	
HP	High pressure	
HTHP	High temperature heat pump	
LP	Low pressure	
MVR	Mechanical vapour recompression	

Roman

h	specific enthalpy	$[J \cdot kg^{-1}]$
$\dot{m}$	mass flow	$[kg \cdot s^{-1}]$
N	impeller speed	$[min^{-1}]$
$\dot{P}$	power	[W]
p	pressure	[Pascal]
$\dot{Q}$	heat flow	[W]
T	temperature	[K]
$\dot{V}$	volume flow	$[m^3 \cdot s^{-1}]$

Greek

η	efficiency	[-]
Π	pressure ratio	[-]
ρ	density	$[kg/m^3]$

Subscripts

cond	condenser
el	electrical
hl	heat losses
in	inlet
is	isentropic
out	outlet
red	reduced or standardised
ref	reference conditions
rel	relative
sat	saturated
t	total

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