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High-efficiency low temperature lift heat pump with novel oil-free, gas-bearing turbo compressor

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Abstract

The Coefficient of Performance (COP) of a heat pump and the Energy Efficiency Ratio (EER) of a chiller are strongly dependent on the internal temperature lift, which corresponds to the difference between the condensation and evaporation temperatures. In modern and efficient building heating and cooling systems, low temperature lifts can be achieved. Turbo compressors have high efficiencies at low pressure ratios during full and part-load operation, thus suiting low temperature lift building heating/cooling applications well. This paper presents the results of experimental investigations on a prototype low temperature lift heat pump/chiller with a heating capacity of 500 kW. The heat pump/chiller is equipped with four novel oil-free turbo compressors built with patented, self-acting gas bearings. High efficiency values during full and part-load operation were measured. Second-law efficiencies for building heating applications of up to 62% were determined. Further increases in efficiency during part-load operation are possible by implementing inlet guide vanes or by shifting the design point of the compressor to lower capacities.

Keywords: turbo compressor; oil-free; low temperature lift; water/water heat pump

1. Introduction

1.1. Background

The development of highly efficient processes and products to significantly reduce CO2 emissions and reach stringent climate targets is one of the biggest challenges of this century. The heating and cooling of buildings requires large amounts of energy, reaching up to 73% of regional energy demands [1]. Experts agree that there is still substantial potential to increase the efficiency of heat pump and chiller systems. The efficiency of heat pumps or chillers is specified by the Coefficient of Performance (COP) or the Energy Efficiency Ratio (EER) and is strongly dependent on the internal temperature lift, which corresponds to the difference between the condensation and evaporation temperatures. The required internal temperature lift depends on the heat source and sink of the system. For modern and efficient building heating systems low temperature lifts between 15 K and 30 K are often sufficient: For efficient building cooling systems even lower temperature lifts between 10 K and 20 K are possible. However, heat pumps and chillers only achieve high second-law efficiencies at low temperature lifts if they are specially designed for the corresponding operating conditions. With the development of low temperature lift heat pumps and chillers, the potential of low temperature lift applications can be exploited.

Moreover, moderate to poor part-load efficiencies often diminish the annual efficiency of building heating and cooling systems, which is a frequent issue as maximum capacity is required only rarely during the year. Turbo compressors have high efficiencies at low pressure ratios during both full and part-load operation, thus suiting low temperature lift building heating and cooling applications well. Resultantly, annual efficiencies of heat pumps and chillers that employ turbo compressors can be significantly higher as compared to systems that employ other compressor types.

1.2. Temperature lifts of building heating and cooling

A short summary of temperature lifts in building heating and cooling applications is presented below. For further details, a comprehensive overview can be found in Gasser et al. [2].

Heating applications

The internal temperature lift of a heat pump used for building heating depends substantially on the heat source and the heat delivery system. In most applications, the temperature lift is typically around 15 K to 75 K (Fig. 1, left). For an efficiently heated low-energy building, desirable temperature lifts range from 15 K to 30 K. This would entail, for example, geothermal heat probes or ground water as the heat source and a modern low-temperature heat distribution and delivery system such as an underfloor heating system. The COP_{ideal} of an ideal heat pump (i.e. Carnot heat pump between the evaporation and condensation temperatures) as a function of the temperature lift for a constant evaporation temperature of 10°C is shown (Fig. 1, right). For a temperature lift of 20 K, a COP_{ideal} of 15.2 is possible. If a system with a second-law efficiency of 60% were implemented, a COP of 9.1 would result.

Cooling applications

The temperature lift of a chiller used for building cooling depends substantially on the space cooling and re-cooling system. In most applications, the temperature lift is typically around 10 K to 55 K (Fig. 2, left). For an efficiently cooled low-energy building located in central Europe, desirable temperature lifts range from 10 K to 30 K. This would entail, for example, space cooling using chilled ceilings, concrete core activation or highly efficient fan coil units and an efficient hybrid/adiabatic re-cooling system. The EER_{ideal} of an ideal chiller (i.e. Carnot chiller between the evaporation and condensation temperatures) as a function of the temperature lift for a constant evaporation temperature of 14°C is shown (Fig. 2, right). For a temperature lift of 12 K, an EER_{ideal} of 24 is possible. If a system with a second-law efficiency of 60% were implemented, an EER of 14.4 would result.

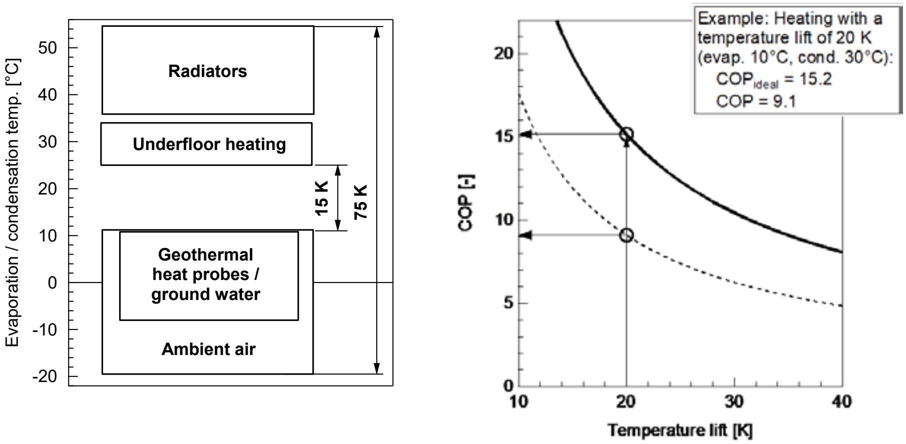

Fig. 1. Left: Typical evaporation and condensation temperatures and temperature lifts of different heat pump systems
Right: COP of an ideal heat pump and a real heat pump with a second-law efficiency of 60% as a function of the temperature lift for a constant evaporation temperature of 10°C [2]

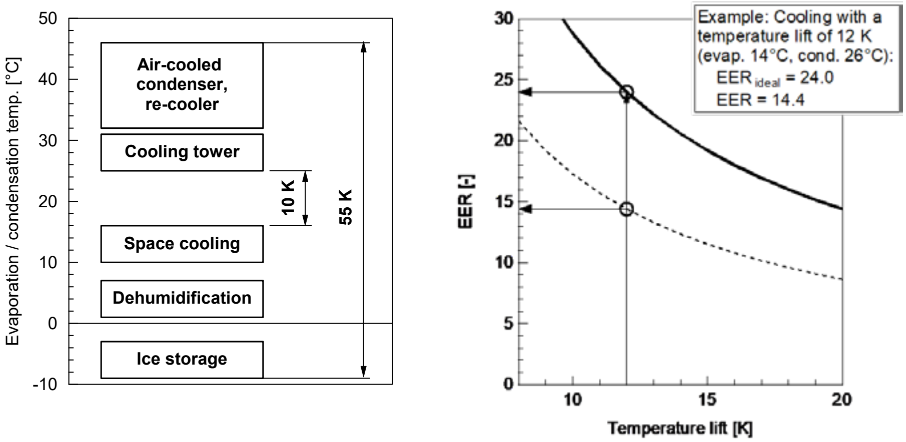

Fig. 2. Left: Typical evaporation and condensation temperatures and temperature lifts of different chiller systems
Right: EER of an ideal chiller and a real chiller with a second-law efficiency of 60% as a function of the temperature lift for a constant evaporation temperature of 14°C [2]

1.3. State-of-the-Art and recent research

The use of turbo compressors in compression heat pumps and chillers originates in 1922 when the first centrifugal refrigeration compressor was presented [3]. Today, turbo compressors are mainly widespread in larger heat pumps and chillers with up to several megawatts of heating/cooling capacity. Prestigious manufacturers include Carrier, Danfoss Turbocor, Friotherm and Mitsubishi Heavy Industries.

The smallest available compressors from these manufacturers are about 200 kW nominal cooling capacity. Most compressors above 200 kW nominal cooling capacity were found not to utilize gas bearings, requiring oil lubrication. As an exception, Danfoss Turbocor uses magnetic bearings and is thus the only manufacturer providing oil-free compressors within this capacity range. For heat pumps/chillers with lower capacities, the use of turbo compressors is rather rare. The manufacturers Celeroton and Fischer offer oil-free refrigerant turbo compressors in the range of 20 kW nominal cooling/heating capacity and utilize gas bearings. Accordingly, there is a market gap in capacity of available turbo compressors.

In recent years, various turbo compressors for heat pump and chiller systems have been introduced across several IEA Heat Pump Conferences. Generally, the presented turbo compressors can be split amongst two groups, those with nominal heating capacities in the range of 20 kW, and those with nominal heating capacities greater than 500 kW, with exception to that presented by Kim et al. (ca. 130 kW heating capacity) [4].

For nominal heating capacities in the range of 20 kW, it is noticeable that the rotational speeds are relatively high with 160-280 krpm [2], [5], [6]. For this application and these rotational speeds, Arpagaus et al. already presented self-acting gas bearings which remove the need for oil lubrication [6]. At heating capacities above 130 kW the rotational speeds are often significantly lower with 11-24 krpm [4], [7]. Accordingly, ball and rolling bearings [4], [8], and seldomly, magnetic bearings [9], are used.

Only Gasser et al. indicates the resulting second-law efficiency (determined with the operating condensation and evaporation temperatures) when operating the turbo compressor in a heat pump, which was found to be at maximum 64% [2].

The presented turbo compressors for small heating capacities are intended for use in heat pumps for building heating [2], [6]. For higher heating capacities, the turbo compressors are often used to generate hot water in the range of 75-95°C [9], [10], [11].

From a research perspective, much of the work conducted regarding turbo compressors follows the same trends as the market, tending not to focus on the 20 to 200 kW capacity range.

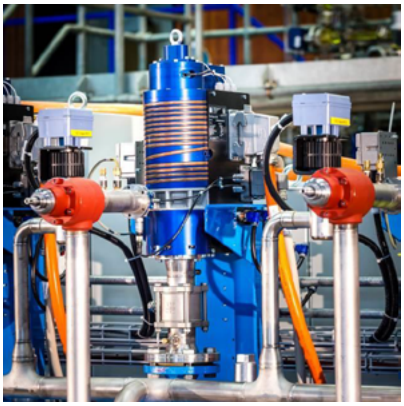

Fig. 3. Novel oil-free, gas-bearing turbo compressor by Teqtoniq GmbH installed in a heat pump/chiller

1.4. Objectives

The goal of this paper is to demonstrate the efficiency of a heat pump/chiller equipped with a novel oil-free, gas-bearing centrifugal turbo compressor. The overall project aim is the development of a marketable and highly efficient centrifugal turbo compressor for heat pumps and chillers with a cooling capacity of 20-100 kW per compressor.

2. Novel oil-free, gas-bearing turbo compressor

The novel, oil-free and gas-bearing centrifugal turbo compressor presented is a development of the Swiss company Teqtoniq GmbH in collaboration with the Lucerne University of Applied Sciences and Arts. The compressor has been developed for use in heat pump and chiller systems with temperature lifts up to 40 K. The cooling capacity of a single compressor can be increased from about 20 kW to 100 kW. Patented self-acting gas bearings are used to support the shaft and impellers at rotational speeds of up to 60 krpm. The gas bearings eliminate the need for oil lubrication, allowing the compressor to operate completely oil-free. Due to the similar gas properties, the compressor can be used with the refrigerants R1234ze, R513a and R134a. In this paper, only the refrigerant R1234ze is considered. The technical data of the compressor are summarized in Table 1.

Table 1 Technical Data of the novel oil-free, gas-bearing turbo compressor by Teqtoniq GmbH

Type	**centrifugal, two-stage**
Electrical power	max. 28 kW
Rotational speed	variable, max. 60 krpm
Refrigerant	R1234ze
Nominal cooling capacity	100 kW
Pressure ratio	max. 4.0
Mass flow rate	max. 0.8 kg/s

3. Experimental setup

3.1. Test rig

The test rig located at the Lucerne University of Applied Sciences and Arts allows the experimental investigation of various thermal systems. Such systems include chillers and heat pumps (up to a cooling capacity of 600 kW) as well as heat exchangers or thermal energy storages.

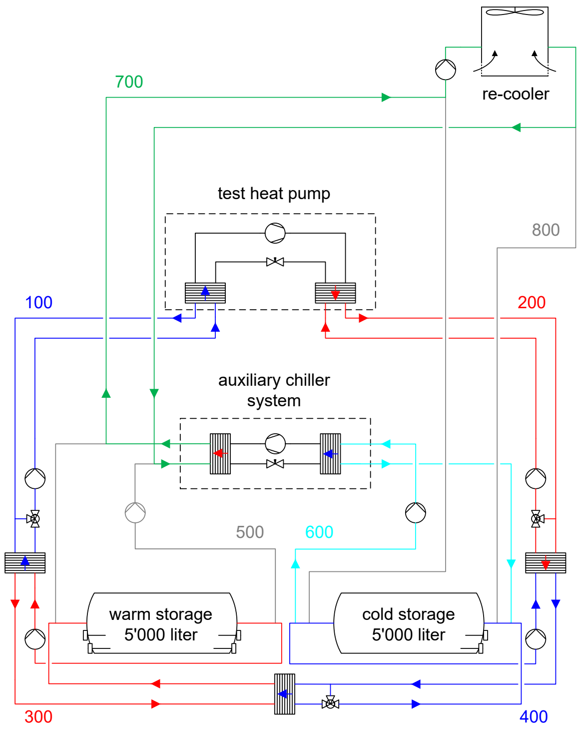

Fig. 4. Simplified schematic view of the test rig at the Lucerne University of Applied Sciences and Arts; application “testing a water/water heat pump/chiller”; cooling of the infrastructure via auxiliary chiller system

With this infrastructure, it is possible to provide heating and cooling within a temperature range of -20°C to +80°C. The core are two separate warm and cold storage tanks, each with capacities of 5,000 litres and filled with an aqueous glycol mixture. Depending on the application, the storage tanks can be conditioned using an auxiliary chiller system, a re-cooler, or electric heating rods. The heat is distributed in various circuits, which are equipped with variable-speed pumps and flowmeters for flow control. The temperatures of some individual circuits are controlled via 3-way mixing valves. With this test rig it is possible to experimentally investigate heat pumps and chillers according to the test standard EN14511. The requirements for control and measurement accuracy can be easily met. Fig. 4 shows a simplified schematic view of the test rig for the application "testing a water/water heat pump/chiller".

When testing a water/water chiller, in circuit 100 the water temperature and mass flow into the evaporator of the test heat pump are adjusted to the specified nominal values. This allows the simulation of a real heat source. In circuit 200, the water temperature and mass flow into the condenser of the test heat pump are adjusted, thereby simulating the heat sink. The two circuits 100 and 200 are connected with the two “inner” circuits 300 and 400 via three heat exchangers. This allows the heat generated at the condenser of the test heat pump to be partially fed to the evaporator of the test heat pump, resulting in maximum heat recovery. The excess heat generated by the larger condenser heat flow in the test heat pump is stored in the cold storage tank, which heats up as a result. When a maximum target temperature is reached, the cold storage tank is cooled to a minimum target temperature via circuit 600 with the auxiliary chiller system. The condenser waste heat from the auxiliary chiller system is released to the ambient air via circuit 700 with the aid of a re-cooler. The task of the warm storage tank is to compensate for temperature fluctuations that occur in circuits 200 and 400. This ensures a constant temperature at the inlet to the heat exchanger, which separates circuits 100 and 300.

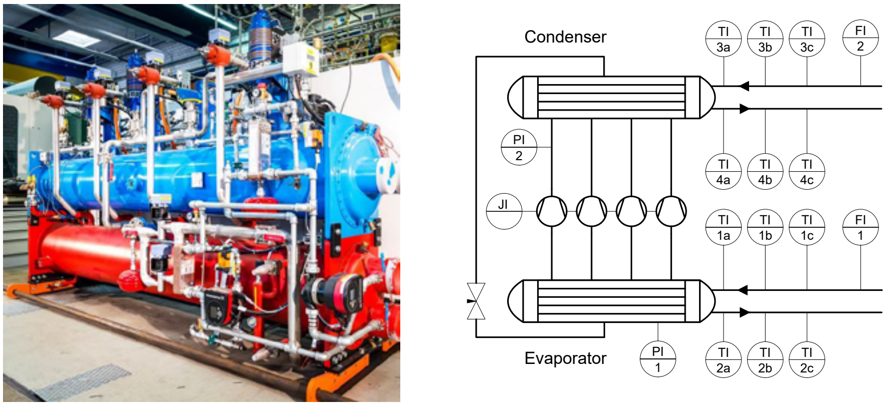

Fig. 5. Left: Water/Water heat pump/chiller with four novel oil-free, gas-bearing turbo compressors by Teqtoniq GmbH
Right: Measurement scheme for the heat pump/chiller; PI: pressure indicator, TI: temperature indicator, FI: mass flow rate indicator, JI: electrical power indicator

3.2. Heat pump/chiller prototype

The tested heat pump/chiller prototype (Fig. 5, left) is equipped with four novel oil-free turbo compressors with self-acting gas bearings. Each turbo compressor has a nominal cooling capacity of approximately 100 kW, resulting in an overall nominal cooling capacity of 400 kW or 500 kW heating capacity, respectively. The capacity of the heat pump can be regulated by the rotational speed of the turbo compressors and by switching compressors on and off. The measurement scheme for the heat pump is also given (Fig. 5, right).

3.3. Test conditions

Heat pump application

The seasonal performance of a heat pump for building heating is determined in compliance with the EN14825 standard [12]. Depending on the heat source and sink, the standard specifies operating points with different part-load ratios for the heat pump. The experimentally determined COP values at the respective operating points can then be used to calculate the expected seasonal performance factor associated with climatic conditions of a location. This paper presents a water/water heat pump designed for low temperature lifts. Hence, only low-lift operating conditions of the EN14825 standard are tested. The operating conditions for a heat pump that works with a heat source such as ground water (10°C) and provides heating water with low temperatures (max 35°C) for an efficient new building therefore apply (Table 2). The water flow rates of heat source and sink in evaporator and condenser are determined by adjusting them such that the water inlet and outlet temperatures match the specified temperatures (10/7°C, 30/35°C) for full load operation (see Table 2). For every other operating point, the determined flow rates are kept constant, resulting in a decreasing temperature differential between inlet and outlet as part-load is reduced.

Chiller application

The seasonal performance of a chiller for building cooling is stated frequently using an index called the European Seasonal Energy Efficiency Ratio (ESEER), which is based on the EN14511 standard. This index represents an average annual efficiency of chillers for building cooling purposes in Europe. To achieve this, EER values at full load and in different part-load operating points are combined using different weighting coefficients. The operating points differ depending on the heat source and heat sink used. The water flow rates of heat source and sink in evaporator and condenser are determined as before (see Table 3). For every other operating point, the determined flow rates are kept constant.

Table 2 Test conditions for the test heat pump in heating applications in compliance with the EN14825 [12] standard

Part-load	Water temperatures evaporator in/out [°C]	Water temperatures condenser in/out [°C]
100%	10 / 7	30 / 35
88%	10 / -	- / 34
54%	10 / -	- / 30
35%	10 / -	- / 27
15%	10 / -	- / 24

Table 3 Test conditions for the test chiller in cooling application in compliance with EN14511 [13] and Eurovent [14] standards

Part-load	Water temperatures evaporator in/out [°C]	Water temperatures condenser in/out [°C]
100%	12 / 7	30 / 35
75%	- / 7	26 / -
50%	- / 7	22 / -
25%	- / 7	18 / -

4. Results and discussion

Heat pump application

For heat pump application measurements, the heat pump presented in Section 3.2 was operated with only three turbo compressors. This is because one turbo compressor was demounted for inspection at the time of the measurements. The nominal heat output at 100% load ratio is accordingly related to three turbo compressors and corresponds to approximately 370 kW heat output. The operating points listed in Table 2 are measured based on the EN14511 standard. The results are shown in Fig. 6. The characteristic values COP and second-law efficiency are determined according the following equations:

$$\mathrm{COP_{HP}} = \frac{\dot{Q}_{cond}}{P_{el}}\ ;\ \mathrm{COP_{Carnot}} = \frac{T_{cond}}{T_{cond}-T_{evap}} \tag{1}$$

$$\eta_{2nd\ law,HP} = \frac{\mathrm{COP_{HP}}}{\mathrm{COP_{Carnot}}} \tag{2}$$

It can be seen that the COP increases from 6.12 to 8.48 by lowering the internal temperature lift from 31 K to 16 K (Fig. 6). The second-law efficiency (related to the evaporation and condensation temperature, i.e. internal behavior of the heat pump) decreases from 0.62 to 0.46. This behavior can be attributed to the design point of the turbo compressor, which lies at higher refrigerant volume flows and pressure ratios. This effect is further intensified by the inlet guide vanes, which are not in operation with the current prototype.

The efficiency of a turbo compressor at lower volume flows than at the design point can be increased by the correct use of inlet guide vanes. It can be seen in the right-hand plot of Fig. 6 that the second-law efficiency is the lowest at the operating point with the lowest heating output, by a considerable margin. This behavior can be explained again with reference to the design point of the turbo compressor. The required heating capacity is too small for the turbo compressor at the given temperature lift, resulting in a lower efficiency. If the nominal heat output of the heat pump had been determined with four compressors, the required heat output for one compressor in this operating point at 15% part-load ratio would have been significantly higher, and the resultant second-law efficiency would have been similarly elevated.

In summary, given a second-law efficiency within the range of 46-62% the heat pump with three novel turbo compressors shows remarkable efficiency values despite its prototype status. With optimization measures such as the tuned operation of the already installed inlet guide vanes, the efficiency of the system could be further improved. Additional increases in efficiency are possible by shifting the design point of the turbo compressor to smaller refrigerant volume flows and temperature lifts. Especially in view of the fact that the part-load operating points have a higher share of operation during the year, therefore having a larger impact on the annual efficiency.

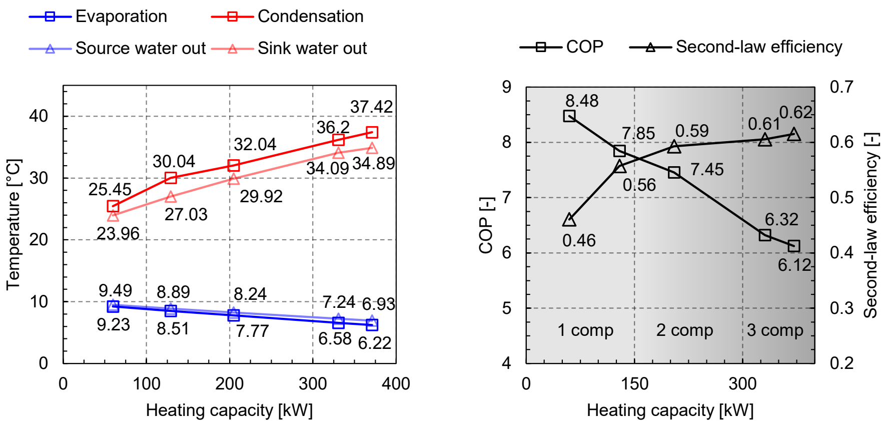

Fig. 6. Left: Water outlet temperatures of evaporator and condenser and corresponding evaporation and condensation temperature depending on the heating capacity at the operating conditions mentioned in Table 2
Right: Corresponding COP and second-law efficiency

Chiller application

The chiller presented in Section 3.2 was operated using all four turbo compressors for the chiller application measurements. The nominal cooling capacity of the system corresponds to approximately 400 kW. The chiller operating points listed in Table 3 are measured based on the EN14511 standard. The measurement scheme of the test heat pump stays the same and can be seen in Fig. 5. The results are shown in Fig. 7. The characteristic values EER and second-law efficiency are determined according to equations (3) and (4):

$$\mathrm{EER_{Ch}} = \frac{\dot{Q}_{evap}}{P_{el}}\ ;\ \mathrm{EER_{Carnot}} = \frac{T_{evap}}{T_{cond}-T_{evap}} \tag{3}$$

$$\eta_{2nd\,law,Ch} = \frac{\mathrm{EER_{Ch}}}{\mathrm{EER_{Carnot}}} \tag{4}$$

In general, similar behavior in the performance of the turbo compressor between the two applications was observed. Noticeably, the second-law efficiency has significantly increased constancy across all operating points. Here, too, the chiller with four novel turbo compressors shows remarkable efficiency values despite its prototype status. Similarly, achieving heightened efficiencies for chiller applications can be accomplished by adjusting the design point of the turbo compressor.

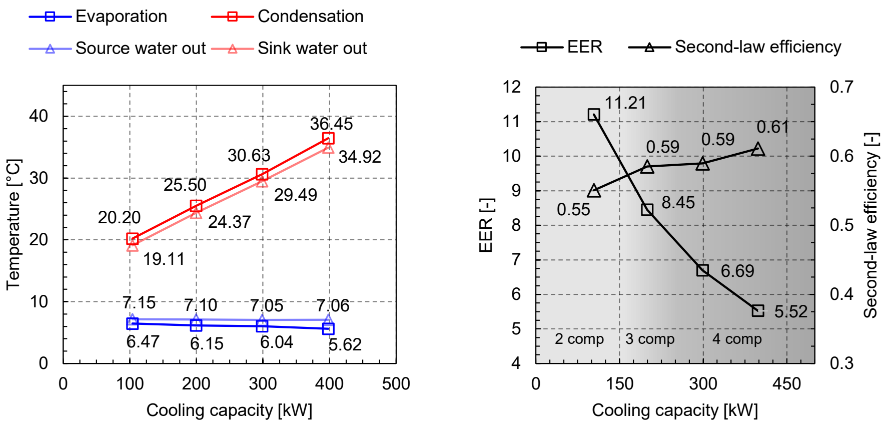

Fig. 7. Left: Water outlet temperatures of evaporator and condenser and corresponding evaporation and condensation temperature depending on the heating capacity at the operating conditions mentioned in Table 3
Right: Corresponding EER and second-law efficiency

5. Conclusions

The consistent use of available low temperature lifts in building heating and cooling can save considerable amounts of electricity. A prerequisite is that the implemented heat pump or chiller be specifically designed for these operating conditions, thus enabling high second-law efficiencies.

The heat pump prototype with four novel, oil-free and gas bearing turbo compressors shows efficiency values which are in the top range for comparable products on the market. For typical temperature lifts encountered in modern building heating and cooling systems, second-law efficiencies around 60% can be achieved for both heating and cooling applications. Overall, the prototype performs better for building cooling purposes than building heating, although is generally well suited to both. Results also indicate potential to increase the efficiency further, by ensuring that compressor part-load performance is optimized for the operating conditions most frequently encountered throughout the year. Increases in efficiency values of several percentage points would be expected, and further work is already being conducted in this area.

Finally, in order to achieve highly efficient building heating and cooling, not only must the heat pump or chiller be efficient, the entire system has to be designed to function efficiently. A non-exhaustive list of design concerns includes the hydraulic circuits, the heat exchange on the source and sink side, building dynamics, and controls systems. Considered implementation of all these measures together with efficient low-lift heat pumps and chillers will lead to a significant reduction in both exergy consumption and on-going operating costs for the heating and cooling of buildings.

Acknowledgements

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