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Cordin Arpagaus
NTB University of Applied Sciences of Technology Buchs, Institute for Energy Systems, Buchs, Switzerland,
cordin.arpagaus@ntb.ch

Frédéric Bless
NTB University of Applied Sciences of Technology Buchs, Institute for Energy Systems, Buchs, Switzerland,
frederic.moix@ntb.ch

Jürg Schiffmann
Ecole Polytechnique Fédérale de Lausanne, Laboratory for Applied Mechanical Design, Neuchâtel, Switzerland,
jurg.schiffmann@epfl.ch

Stefan Bertsch
NTB University of Applied Sciences of Technology Buchs, Institute for Energy Systems, Buchs, Switzerland,
stefan.bertsch@ntb.ch

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Cordin ARPAGAUS[1]*, Frédéric BLESS[1],
Jürg SCHIFFMANN[2], Stefan S. BERTSCH[1]

[1]NTB University of Applied Sciences of Technology Buchs
Institute for Energy Systems
Werdenbergstrasse 4, 9471 Buchs, Switzerland

[2] Ecole Polytechnique Fédérale de Lausanne
Laboratory for Applied Mechanical Design
Rue de la Maladière 71b, 2002 Neuchâtel, Switzerland

* Corresponding Author, E-mail: cordin.arpagaus@ntb.ch, Phone: +41 81 755 34 94

ABSTRACT

Reducing primary energy consumption by utilizing heat recovery systems has become increasingly important in industry. In many sectors, heating and cooling is required at different temperature levels at the same time. For this purpose, heat pumps are highly attractive energy conversion devices. Heat pumps are widely used for refrigeration, air-conditioning, space heating, hot water production, heat upgrading, or waste heat recovery.

The aim of this paper is to review the literature for mechanical driven heat pumps and refrigeration systems with focus on multi-temperature applications. Different design strategies are presented, including cycles with multi-stage compressors, (multiple) ejectors, expansion valves, cascades (with secondary loops), and separated gas coolers.

This review highlights the major advantages, challenges, and industrial applications of each multi-temperature heat pump cycle family. Schematics and pressure-enthalpy diagrams illustrate the most promising cycles. The performance of the cycles is compared in terms of First Law efficiency (COP) and Second Law efficiency (exergy) using simplified thermodynamic simulations.

The literature reveals that the major part (approximately 70%) of multi-temperature heat pump applications are found in refrigeration, i.e. supermarket food cooling, household fridges/freezers, and cooling/air-conditioning/storage during transportation. In contrast, studies on multi-temperature heating applications are rather rare with the exception of space floor heating and hot water production.

Most multi-temperature cycle designs use two heat sources or two heat sinks. Heat pumps with more than three stages are not common, except for natural gas liquefaction. In supermarket applications, multiple compressors with transcritical CO2 are an established key technology. Cascades with secondary loops are another frequently applied system, mostly in the USA. Cycles with multiple ejectors are ready to market and seem to be a promising modification for system performance improvement. Ejector cycles in refrigeration and air-conditioning systems are still under development. Expansion valve cycles are an established technology in household refrigeration. Separated gas coolers for space and hot water heating have recently attracted attention due to the possible combination with supercritical CO2 cycles.

Overall, this review paper serves to select the most appropriate multi-temperature heat pump cycle for a specific application.

1. INTRODUCTION

Heat pumps are highly attractive energy conversion devices, since they offer an efficient means to reduce primary energy consumption by utilizing heat recovery (Jung *et al.* 2000). In residential buildings, heat pumps are already widely used for space heating and hot water production, particularly in Europe and Japan (Watanabe, 2013). More and more, they are also spreading into the industrial sector, especially for waste heat recovery, heat upgrading, cooling and refrigeration in processes or for heating and cooling industrial buildings (IEA, 2012).

As presented by the International Energy Agency IEA (2012), there is a wide demand for heating and cooling at different temperature levels. In the 27 EU countries, a high demand for low exergy heat is found between 60°C to 100°C, mainly in the pulp and paper and food and tobacco industries. Rather than providing this heat through the combustion of fossil fuels, a more efficient way of providing this service is by using heat pump technology. There are

large amounts of recoverable industrial waste heat, which can be used as a heat source to generate process heat through heat pumps, e.g. for space heating (> 35°C), drying processes (> 70°C) or process heat (120 to 150°C) (IEA, 2014).
Typical working temperatures of industrial heat pumps range from 18°C to 82°C with average temperature lifts of about 31°C (Wolf *et al.* 2014). Major heat sources for heat pumps are available from production processes, e.g. air of 20°C to 50°C (i.e. cooling air, waste gas, exhaust air) and liquids of 15°C to 40°C (i.e. cooling water, oil, cooling lubricants) (Wellig *et al.*, 2012).
Heat pump systems with multi-temperature sinks and sources can cover various cooling and heating needs at the same time while upgrading low quality heat. This topic is the focus of the present literature review. While the simplest heat pump cycle consists of a single stage compressor, a condenser, an expansion valve, and an evaporator, cycles that are more complex are required to share multiple heat sinks and sources at different temperature levels and to improve performance. As shown by Uhlmann *et al.* (2014), over 30% higher COPs can be reached compared to single-stage cycles, when using a two-stage system with two evaporators.
Preliminary screening of the open literature reveals that there is no comprehensive comparison of heat pump cycles addressing multiple heat sources and sinks. Hence, the objectives of this review paper are:

1) a thorough review of heat pump and refrigeration systems, focusing on cycles for multi-temperature applications, especially cooling or heating at two temperature levels,
2) a systematic performance comparison of the different selected cycles in terms of energy and exergy efficiency,
3) identification of a set of unified design guidelines, and future research areas in the field of multi-temperature heat pumps.

An extended version of this study by Arpagaus *et al.* (2016) has been accepted for publication in the International Journal of Refrigeration on May, 24 2016.

2. DESIGN STRATEGIES FOR MULTI-TEMPERATURE HEAT PUMPS

Several design strategies for multi-temperature cycles have been identified and analyzed in this study. The split of different temperature levels is primarily achieved by means of special arrangement of the compressors, heat exchangers, expansion valves, ejectors, or cascaded systems. This enables different design strategies for multi-temperature heat pumps and refrigeration systems to be classified in cycle families.
Figure 1 gives an overview of these cycle families, including multi-stage compressors, expansion valves, (multiple) ejectors, cascades (with secondary loops), and separated gas coolers.

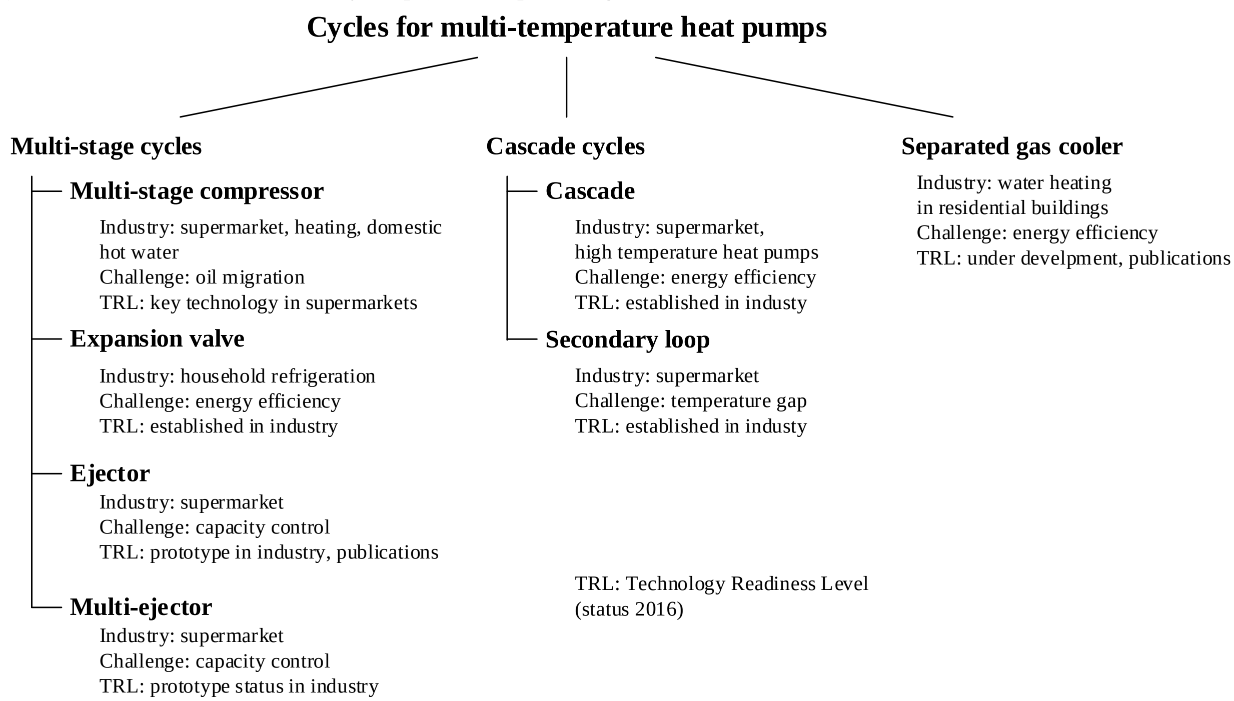


Figure 1: Overview of cycles applied for multi-temperature heat pumps

Table 1 summarizes the major advantages and challenges for the investigated multi-temperature cycles. Factors affecting the choice of cycle are, amongst others, capacity control, oil management, costs, variability and amounts of refrigerant, energy efficiency, retrofit potential, and the technology readiness level.

Table 1: Advantages and challenges of multi-temperature heat pump cycles separated by categories

Cycle families	Advantages	Challenges
Multi-stage compressor	+ industrialized and key technology in supermarkets + simple and independent capacity control for each heat sink / source + high temperature lifts possible + intercooling of discharge vapor to improve efficiency + cost savings (only one compressor instead of two possible)	– oil management required in serial compressor configuration – high amounts of refrigerant – capacity control regulation – commercially available but further development needed to increase efficiency
Ejector	+ recovery of expansion work + simple structure + no moving parts (compressor-free) + low investment and maintenance costs	– capacity control (passive element) – adjustable ejector development – efficiency only over a narrow operating range
Multiple ejectors	+ no problems with oil migration + retrofit potential in supermarkets + efficiency over a wide operating range	– capacity control (on/off, non-continuously controllable) – adjustable ejector development
Expansion valves	+ established in industry (household refrigerators/freezers, heat pumps) + simple and competitive (low costs)	– lower energy efficiency compared to ejector or multiple compressors
Cascade	+ established technology in industry + high temperature lifts + variability of refrigerants extends operation range	– temperature gap in cascade heat exchangers (reduces energy efficiency) – higher cost for maintenance
Secondary loop	+ established technology in supermarkets (especially in the USA) + smaller amounts of refrigerants in the system	– temperature gap in the chiller (reduces energy efficiency) – pump losses of viscous secondary fluid
Separated gas cooler	+ combined heating of space and hot water possible with higher efficiency + future perspective for monovalent heat pumps	– high efficiency only for specific flow rates of heat source and sink

Beside the common multi-stage and cascaded cycles, thermally driven absorption cycles are also an option to achieve multi-temperatures. However, this review does not focus on these particular cycles. For further information on absorption cycles, reference is made to a review by Srikhirin *et al.* (2000) or Dincer and Kanoğlu (2010).

2.1 Multi-stage compressor cycles

Multi-stage compressor cycles are systems with two or more compression stages in the same heat pump cycle (Dincer and Kanoglu, 2010). The compressors are connected either in series or in parallel using single, multiple or multi-stage compressors with injection ports between the compression chambers. Multi-stage compression is recognized to provide high COPs (IEA, 2014). Intercooling of the discharge vapor between compressor stages is an important additional feature to improve the efficiency and performance of the cycle (Baek *et al.*, 2002; Yari, 2009).
Advantages of multiple compressor systems in series are the simple compression power control for each compression stage. Challenges are oil management regulation, oil migration, and potentially high amounts of refrigerant in the system. Multi-stage compressor cycles are a common configuration in supermarket refrigeration (e.g. cooler, freezer) or water heaters. In the food refrigeration industry, facilities maintain space at typically -23°C for storing frozen food and +2°C for unfrozen fruit and vegetables (Stoecker, 1998). The technology readiness level for multi-stage compressors and secondary loops is already very high.

2.2 Expansion valve cycles

Expansion valve cycles are established technology in household refrigerators and freezers for food conservation, as well as in heat pump applications, as they are simple and cost competitive. Compared to multi-stage compressor cycles the investment costs are lower (Yoon *et al.*, 2010, 2011; Visek *et al.*, 2014).

2.3 Ejector cycles

The use of ejector cycles has become a promising cycle modification in refrigeration and air conditioning research, thanks to the absence of moving parts, low cost, simple structure, and low maintenance requirements (Elakhdar *et al.*, 2007; Elbel & Hrnjak, 2008; Sarkar, 2012; Elbel, 2011). The main advantage of an ejector is the recovery of the expansion work (COP improvement), and flash gas bypass (evaporator size reduction). The integration of ejectors allows the compressor work to be reduced by increasing the suction pressure or by reducing the exhaust pressure. In contrast to multi-stage compressor systems, the technology readiness level for ejector cycles is lower. These concepts still require major development work in capacity control of multiple stages. Standard ejectors for refrigeration systems are not yet commercially available.

2.4 Multiple ejector cycles

Applications with multiple ejectors are found especially in CO2 supermarket refrigeration (Hafner *et al.*, 2014; Schönenberger, 2013) or transport refrigeration (Kairouani *et al.*, 2009). Capacity control in multiple stages is a challenge, as ejectors are passive elements that are non-continuously controllable; therefore, off-design operation or poor ejector design could make the ejectors fail to entrain suction flow properly (Kornhauser, 1990; Lawrence & Elbel, 2013, 2014; Li & Groll, 2005). Nevertheless, integration of ejectors in existing transcritical CO2 systems is a promising approach in industrial refrigeration to improve energy efficiency.

2.5 Cascades

A cascade cycle is a combination of two or more heat pumps (or refrigeration cycles) where the intermediate heat exchangers connect the cycles. The cascade control improves the range of operation in the next higher cycle with the possibility of having a different working fluid for each cycle. Each fluid can be selected for optimum performance in the specified temperature range (Dincer & Kanoglu, 2010). In addition, a cascade cycle is easier to control and operate than a multi-stage compressor cycle and shows no oil management issues.

A frequent example of refrigerant combination is the use of CO2 in the low temperature cascade and NH3 or hydrofluorocarbon refrigerants in the high-temperature cascade (Bitzer, 2014). The cascade circuits may also be built with a rack of parallel compressors for capacity modulation. This arrangement enables an extended operation range, e.g. high-temperature lifts between heat source and sink ranging from -70 to +100°C without any oil migration issues. On the other hand, the temperature difference in the cascade heat exchangers degrades the system performance, which is a major challenge in terms of energy efficiency.

From an application point of view, cascades are commonly employed in supermarket refrigeration (Bitzer, 2014; Hill Phoenix, 2011; Sharma *et al.*, 2014), high temperature heat pumps for heat recovery (Kondou & Koyama, 2014), or for gas liquefaction (Dincer & Kanoglu, 2010).

2.6 Cascades with secondary loop

Secondary loop systems coupled with conventional cascade refrigeration systems have recently seen increased popularity in retail food application, i.e. supermarkets and storage warehouses, due to significant reduction in working fluid charge compared to traditional direct expansion refrigeration systems (Dincer & Kanoglu 2010). As shown by DelVentura *et al.* (2007), Hill Phoenix (2011), Sharma *et al.* (2014), or Bitzer (2014), secondary loop circles are preferably connected to medium temperature refrigeration systems (e.g. meat, prepared foods, dairy or refrigerated drinks), typically maintained at -7 °C (SCEFM, 2004).

2.7 Separated gas cooler

Heat pump cycles with separated gas cooler sections present a less commonly encountered system for multi-temperature purposes. However, this offers an elegant approach for applications, such as water heating, in particular in combination with supercritical CO2 cycles (Neksa, 2002; Stene, 2005).

3. NUMERICAL COMPARISON OF CYCLES

3.1 Selected cycles for simulation

The most promising cycles of the various cycle families have been selected to compare performance in terms of the First Law efficiency (COP) and Second Law efficiency (η_{2nd}) for the same operating conditions. Table 2 presents the schematics and the p-h diagrams of the selected cycles analyzed in this simulation study. The cycles have two heat sources at low and medium temperature (LT, MT) and one heat sink at high temperature (HT). As a reference (Ref), a cycle with two parallel single-stage heat pumps is used. The MC 1, 2, 3, and 4 cycles represent the multi-stage compressor cycle family. Each system incorporates a specific cycle characteristic, such as a subcooler (MC 1), an open economizer (MC 2), a closed economizer (MC 3), or a booster system (MC 4). The cascade cycle (CAS) selected represents a two-stage cascade with one intermediate heat exchanger using the same working fluid (R134a) in both stages. The selected expansion valve cycle (EXV) is a common household refrigeration cycle with both evaporators working in continuous operation. The analyzed ejector cycle (EJ) uses saturated vapor leaving the separator as the motive fluid to compress partially the saturated vapor leaving the LT evaporator.

A water heating application using R134a as the working fluid is used as case study with selected temperature condition $T_{HT} = 60°C$ (hot water), $T_{MT} = 30°C$ (waste water), and $T_{LT} = 0°C$ (brine), respectively. The total heating capacity is kept constant at $\dot{Q}_{HT} = 10\ kW$. The factor $\beta = \dot{Q}_{MT}/\left(\dot{Q}_{MT} + \dot{Q}_{LT}\right)$ is introduced, representing the ratio between the two heat sources at LT and MT. By varying β from 0 to 1 it is possible to simulate different heat source supply conditions, e.g. $\beta = 0$ corresponds to the situation where the complete heat supply occurs in the LT evaporator.

3.3 Thermodynamic models

Thermodynamic models have been developed in Engineering Equation Solver (EES) software (Klein, 2012) to compare the system performance of the different cycles. Further details can be found in Arpagaus *et al.* (2016). The calculations are based on energy and mass conservation with the following assumptions:

- Pressure drops and heat losses in piping and heat exchangers are neglected.
- Heat exchangers are ideal with a zero approach temperature difference (pinch = 0°C).
- The evaporation (LT, MT) and condensation temperatures (HT) of the refrigerant are the same as the temperatures of the heat sources and sink (no superheating and subcooling).
- The approach temperature in the cascade heat exchanger is 5°C.
- All expansion valves operate adiabatically and provide isenthalpic expansion processes.
- All compressors operate adiabatically with a specific isentropic efficiency of $\eta_c = 0.7$.
- The separators have perfect separation efficiencies and the fluid outlets are saturated vapors and liquids.
- The pressure in the liquid-vapor separator of the MC 4 cycle is the geometric mean of the HT and MT pressure, which is considered to be optimum according to Stoecker (1998).
- The ejector cycle (EJ) is modeled using the approach of Kornhauser (1990), and assumes:
  - steady state and one-dimensional flow (e.g. constant fluid properties and velocities across the ejector cross-section),
  - constant mixing pressure inside the ejector, 0.5°C (Lawrence and Elbel, 2013) lower than the suction nozzle inlet pressure, and
  - constant ejector efficiencies: motive nozzle $\eta_{mn} = 0.8$, suction nozzle, $\eta_{sn} = 0.8$, and diffuser $\eta_{diff} = 0.8$. The kinetic energies outside the ejectors are negligible (velocities are zero).

The Carnot COP is described as a function of the two individual single stage Carnot COP's at LT and MT and the weighted average factor β of the two heat sources. The factor α is used as an auxiliary variable. The Second Law efficiency (η_{2nd}) is defined for each cycle as the ratio of the predicted cycle COP over the Carnot COP. COP_{cycle}, COP_{Carnot} and η_{2nd} are calculated based on equations (1) to (3). As indicated in Figure 2, the Carnot COP increases with β from 5.55 up to 11.11.

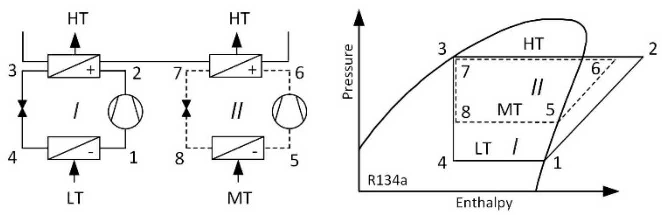
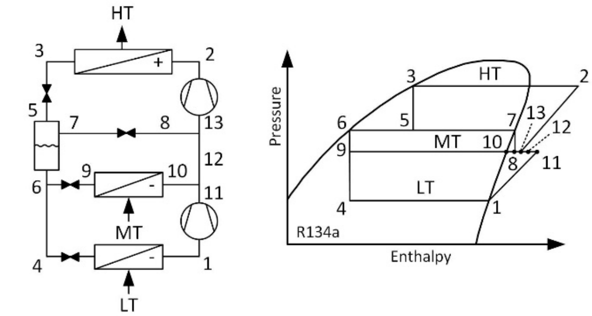
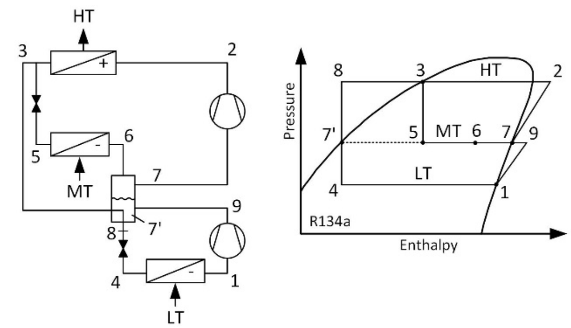
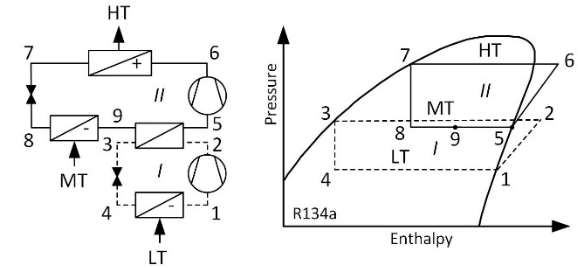
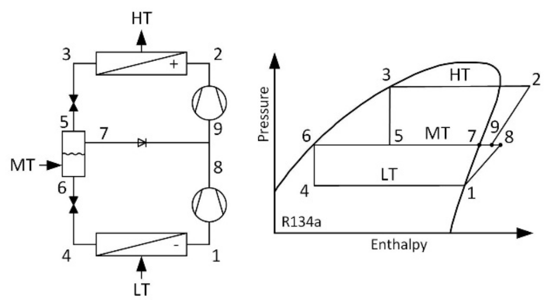
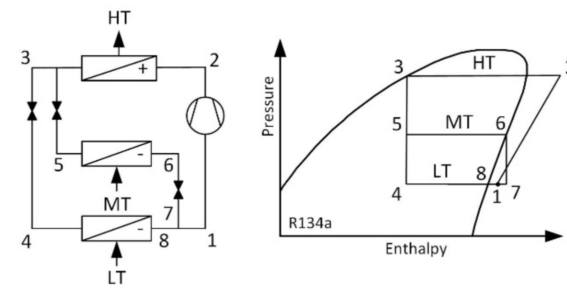
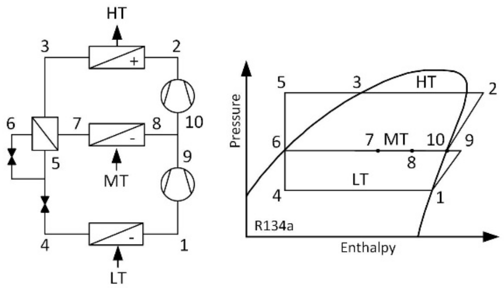
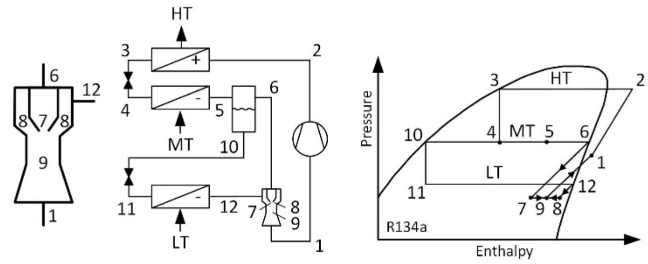
$$COP_{Carnot} = \frac{1}{1+\alpha}\,COP_{Carnot,LT} + \frac{\alpha}{1+\alpha}\,COP_{Carnot,MT} \quad (1)$$

with $\alpha = \frac{\beta}{1-\beta}\;\frac{COP_{Carnot,LT}-1}{COP_{Carnot,MT}-1}$

$$COP_{Carnot,LT} = T_{HT}/(T_{HT} - T_{LT}) = 5.55 \quad (2)$$
$$COP_{Carnot,MT} = T_{HT}/(T_{HT} - T_{MT}) = 11.11$$

$$\eta_{2nd} = COP_{cycle}/COP_{Carnot} \text{ with } COP_{cycle} = \dot{Q}_{HT}/W \quad (3)$$

Table 2: Schematics and p-h diagrams of the selected multi-temperature cycles for simulation

Ref Reference cycle with two single-stage heat pumps in parallel 	**MC 4** Multi-stage compressor cycle with booster system (Advansor, 2015; Bitzer, 2014; Sharma *et al.*, 2014) 
MC 1 Multi-stage compressor cycle with subcooler (Stoecker, 1998) 	**CAS** Cascade cycle (Dincer & Kanoglu, 2010; Kanoğlu, 2002) 
MC 2 Multi-stage compressor cycle with open economizer (Granwehr and Bertsch, 2012; Uhlmann *et al.*, 2014) 	**EXV** Expansion valve cycle with continuous operation (Visek et al., 2014; Yoon et al., 2010, 2011) 
MC 3 Multi-stage compressor cycle with closed economizer (Granwehr and Bertsch, 2012) 	**EJ** Ejector cycle with two heat sources (Elakhdar *et al.*, 2007) 

4. SIMULATION RESULTS

Figure 2 (left) shows the heating COPs of the different cycles as a function of the heat supply ratio β. As can be seen, the COP increases for all cycles with β. The EXV cycle has the lowest COP, followed by the EJ cycle. The COPs of the MC cycles are roughly identical over the whole β-range, independent of the internal cycle characteristics. Compared to the Ref cycle the curves are slightly higher, which results from the internal heat recovery ability of the economizers. On the other hand, this benefit decreases with higher β-values and disappears completely at $\beta = 1$, where the cycles behave like a single stage cycle. At $\beta = 1$, the COPs of the MC 1 to 4, CAS, and Ref cycles converge to 6.62, which means that the full heat supply occurs on the MT evaporator. The CAS cycle behaves similarly to the MC cycles in terms of performance. Its COP is slightly lower over the whole β-range due to exergy losses in the cascade heat exchanger. The smaller the assumed temperature difference, the closer the COP is to the MC cycles.
The COP of the EXV cycle is substantially lower than all the other cycles and almost constant. This is because the evaporation temperature and the pressure lift remain the same in all situations. There is no benefit from the second heat source at MT, except a slight increase in the superheat of the compressor inlet. In practice, the performance of the EXV cycle is sometimes improved by running the cycle in alternating operation by switching between the MT and LT subcycle as a function of time.

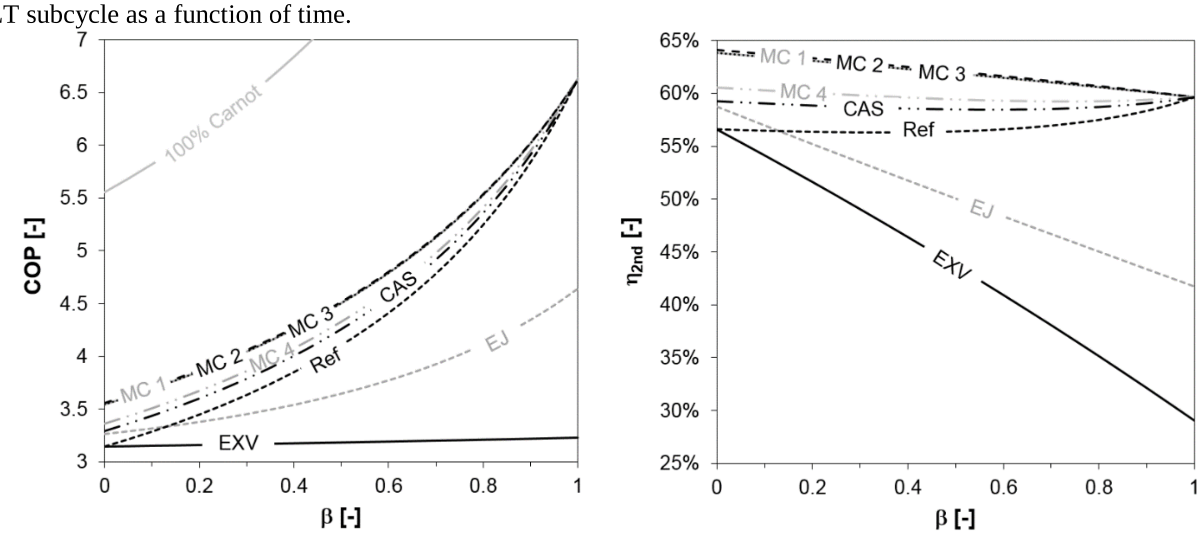


Figure 2: COP and Second Law efficiency (η_{2nd}) of the selected cycles as a function of the heat supply ratio β

The COP of the EJ cycle lies between the EXV and the MC cycles. Compared to the EXV cycle, the EJ cycle recovers some of the expansion work that is otherwise lost by the throttling process. Theoretical COP improvements of 4% (at $\beta = 0$) to 44% (at $\beta = 1$) are reached for the EJ cycle over the EXV cycle under the specified conditions. Several studies show that ejector efficiencies vary widely when operating conditions change (Chen *et al.*, 2013; Li & Groll, 2005; Liu, 2014). In this respect, variable geometry ejectors show an advantage over fixed ejectors (Lino *et al.*, 2013).
Figure 2 (right) shows the Second Law efficiency (η_{2nd}) of the selected cycles as a function of the heat supply ratio β. The η_{2nd} ranges between 56% to 64% at $\beta = 0$, and 30% to 60% at $\beta = 1$. The MC cycles have the highest η_{2nd} and can provide higher thermal efficiency than an EJ and EXV cycles. The ejector in the EJ cycle improves the performance of the EXV refrigeration system by reducing the throttling losses associated with the use of an expansion valve. In the EXV cycle, all of the fluid is throttled from medium pressure to the lowest pressure, resulting in the highest exergy losses.

5. CONCLUSIONS

The literature review reveals that it is possible to classify different design strategies for multi-temperature heat pumps and refrigeration systems into various cycle families; multi-stage compressors, expansion valves, (multiple) ejectors, cascades (with secondary loops), and separated gas coolers.
Factors affecting the design of multi-temperature cycles are, among others, capacity control of the system, achievable temperature lift, oil management regulation between compressor stages, constraints in the compressor exhaust gas temperature, costs, variability and amount of refrigerant in the cycle, off-design operation, energy efficiency, retrofit potential, and the technology readiness level.

The majority (about 70%) of multi-temperature heat pump applications are found in supermarket refrigeration, household refrigeration, and air-conditioning. In contrast, multi-temperature heating applications are rather scarce, with the exception of domestic space heating and hot water production. In supermarkets, multi-stage compressors working in parallel and in series with transcritical CO2 are an established technology. Furthermore, cascades with secondary distribution loops are also widely applied, mostly in the USA. Expansion valve cycles are an established technology in household refrigeration for food conservation. The application of (multiple) ejector cycles in refrigeration and air-conditioning systems are still under development but seems to be a promising modification for system performance improvement. Non-continuous capacity control, low off-design performance, and poor ejector design are challenging factors that hinder a wider successful commercialization. Separated gas coolers for space heating and hot water production have recently attracted greater attention, particularly in combination with supercritical CO2 cycles.

Thermodynamic simulation results reveal that the multi-stage compressor cycles exhibit the highest COP and Second Law efficiency, followed by the cascade, ejector, and expansion valve cycles. The multi-stage compressor cycles behave approximately equally in terms of efficiency, regardless of their internal characteristics, i.e. utilization of an open or closed economizer, which offer internal heat recovery ability. Multi-stage compressor cycles are therefore the energetically preferable circuits for applications comprising multi-temperature heat pumps and refrigeration systems. The COP of the ejector cycle lies between the expansion valve and the multi-stage compressor cycles, as it recovers some of the expansion work, which otherwise would be lost in the throttling process.

Current R&D efforts on multi-temperature heat pumps are found especially in industrial refrigeration with a focus on the implementing multiple ejector cycles in existing transcritical CO2 booster systems for efficiency improvement. Another research focus is on multi-stage compressors with better capacity control regulation and intercooling functions. Furthermore, oil-free compressors, e.g. small-scale turbo compressors, are expected to become more widespread and mitigate the challenges of oil migration and improve compressor efficiency (Arpagaus *et al.*, 2016; Demierre *et al.*, 2015; Javed *et al.*, 2016; Schiffmann, 2014, 2015). Finally, there is a clear trend towards using natural refrigerants, such as CO2, thanks to their low ozone depletion and global warming potential (Advansor, 2015).

NOMENCLATURE

COP	coefficient of performance	(–)
$\dot{Q}$	heat capacity	(kW)
T	temperature	(K)
W	compressor power	(kW)
β	heat source ratio	(–)
α	auxiliary variable	(–)
η	efficiency	(–)
CAS	cascade cycle	
C	compressor	
Carnot	Carnot cycle (theoretical maximum)	
diff	diffusor in ejector	
EJ	ejector cycle	
EXV	expansion valve cycle	
LT	low temperature	
MT	middle temperature	
HT	high temperature	
MC	multi-stage compressor cycle	
mn	motive nozzle in ejector	
TRL	technology readiness level	
sn	suction nozzle in ejector	
2nd	Second Law	

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