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ORIGINAL RESEARCH

Thermoelectric building temperature control: a potential assessment

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Abstract

This study focuses on thermoelectric elements (TEE) as an alternative for room temperature control. TEE are semi-conductor devices that can provide heating and cooling via a heat pump effect without direct noise emissions and no refrigerant use. An efficiency evaluation of the optimal operating mode is carried out for different numbers of TEE, ambient temperatures, and heating loads. The influence of an additional heat recovery unit on system efficiency and an unevenly distributed heating demand are examined. The results show that TEE can provide heat at a coefficient of performance (COP) greater than one especially for small heating demands and high ambient temperatures. The efficiency increases with the number of elements in the system and is subject to economies of scale. The best COP exceeds six at optimal operating conditions. An additional heat recovery unit proves beneficial for low ambient temperatures and systems with few TEE. It makes COPs above one possible at ambient temperatures below 0 °C. The effect increases efficiency by maximal 0.81 (from 1.90 to 2.71) at ambient temperature 5 K below room temperature and heating demand $\dot{Q}_h = 100W$ but is subject to diseconomies of scale. Thermoelectric technology is a valuable option for electricity-based heat supply and can provide cooling and ventilation functions. A careful system design as well as an additional heat recovery unit significantly benefits the performance. This makes TEE superior to direct current heating systems and competitive to heat pumps for small scale applications with focus on avoiding noise and harmful refrigerants.

Keywords Thermoelectric · TEC · Heat pump · Optimization · Heating · Heat recovery

Introduction

Electrically operated heat pumps belong to the clear winners within the market of heat generators (market share with newly established buildings in Germany at 46%) [1]. Nevertheless, there are challenges to make this development even more advantageous in the future: The subsequent installation of heat pumps in existing buildings is associated with significant expenditure, which building owners often do not want to or cannot accept. High supply temperatures are challenging but often required since the specific heating loads are comparatively high and the subsequent installation of a panel heating system is costly. On the other hand, noise emissions and the harmonious integration of the free-standing outdoor unit into the architectural design can be an obstacle with air heat pumps [2]. Furthermore, later installation for individual rental or owner-occupied apartments is hardly feasible since heat pumps available on the market are oversized for these purposes and the necessary changes in the pipe routing are costly. This option would be of particular interest for many apartment buildings in the building stock currently heated with decentralized heat generators (gas floor heating). To open up further market segments and energy saving potentials in the field of decentralized construction, a concept study for a thermoelectric-based micro heat pump is being conducted at the FH Aachen.

Several TEEs are connected in series from the flow point of view to form an air heating system. The TEEs extract heat from the exhaust air and transfer it to the supply air flow. In this study, a counterflow heat exchanger type design is chosen to achieve a good thermal performance. An optimization problem is defined to determine the optimal heating operation for a combination of TEEs in a ventilation system. The objective of the study is to generate data for the dimensioning of a TEE-based air conditioning system and to derive its energy savings potential compared to conventional heating/cooling systems.

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Previous studies have examined the integration of a TEE system for air conditioning purposes in various ways. Early studies for the thermoelectric technology focus on niche environments like the air conditioning of a submarine [3], using thermoelectric generators as a power source in spaceships [4] or controlling temperature in electronic instrumentation [5].

For the past three decades, the efficiency figure ZT for TEE has been seen to be virtually limited to $ZT \leq 1$, but recently ZT values of 1.6 and above have been reached with expectation of further improvement ($ZT \approx 3$). ZT is defined as square of Seebeck coefficient times absolute temperature divided by thermal conductivity and electrical resistance. This improvement is partly due to the introduction of new nanostructuring in materials which reduces thermal conductivity and to systematic exploration of new materials on a nanoscale [6–8].

Serious efforts have been made in order to optimize a single TEE in laboratory conditions [9–11] and to simplify and validate models of thermoelectric heat transfer [12]. Matuška was able to measure a COP for heating above 3, while the highest reported cooling COP that could be found in literature is 2.4 [7].

Building technology has emerged as one region of particular interest for thermoelectric applications. Numerous works have examined the possibility to use thermoelectrics for the purpose of air conditioning in theoretical [13–16] and experimental [17–23] setups. The predominant use case is the cooling case, while one study focuses on an IoT-control approach and also uses the heating function in reversed operation mode [23]. The IoT-control results in an increase of COP of 0.46 and 13% increase in cooling power.

One main focus is to utilize TEE to supplement other building technology like photovoltaics, a solar thermal or a domestic hot water system. The combination with photovoltaics holds the advantage of directly using the generated PV power and a good temporal match between demand and generation. Forced or natural convection in between PV and TE system reduces the operating temperature and benefits the efficiency of both systems [13, 15, 24]. In comparison to a conventional wall, solar-driven thermoelectric coolers can reduce the daily heat gain by 70%. In case the TE system is used to generate surplus electricity from the excess heat of a photovoltaic or solar thermal system, the overall efficiency increases by 34% or 11%, respectively [15].

Several heat transfer techniques have been investigated ranging from free convection [13] over conventional heat sinks and heat pipes to liquid cooling based on water or slurries of phase change material (PCM-Slurry) [14, 16]. For free convection, two concepts have been examined: Façade integrated radiant panels [13, 16, 25] or conventional heatsinks [26] are used to condition the room without the need of complex distribution systems. While electrical power requirements decrease with higher number of thermocouples per TEE and more TEE in the system, [26] finds that 20 TEEs are the optimal trade off between achievable cooling power and required elements for their system. Activated ceiling panels are the second concept that is investigated to make use of free convection heat transfer [14, 16, 27]. One advantage is the more homogeneous heat distribution. A similar system based on liquid cooling system by [15] is found able to provide comfort in the room, while they state that 70 TEEs are optimal to cover the cooling demand of 1107W and 6 TEEs are sufficient for optimal heating operation (200W). The integrated system of [15] consists of a TEE assisted PV and solar thermal system as well as TEE supported activated ceiling. While net zero energy demand can not be achieved without the use of TEE, the TEE system makes the test building a plus energy house.

Another focus is on a third way of conditioning a room, that is, via air conditioning. One promising direction is the combination of an dedicated outdoor air system (DOAS) with an TEE heat pump [20]. This system is not yet competitive to conventional DOAS systems but will be once ZT values above 1.35 become commercially available. Irshad implements 24 TEEs with heat sinks in an supply air duct and reaches cooling COPs of 0.679 at a cooling load of 500 W. The excess heat is transferred to outdoor air [22, 23]. The effect of joining several TEEs in series in an air flow is investigated by [17]. They find that the efficiency of 6 joined TEE is twice the effiency of only 2 TEE when operating at the same total electrical power input. In many applications, it is stated that a good thermal connection between TEE and heatsinks is crucial in order to achieve high efficiency [16, 21, 22].

This simulation study, as well uses air conditioning to meet the proposed demand loads, but in contrast, focuses on the benefits of recovering heat from an exhaust air stream via TEE. This heat is then transferred to the supply air stream for heating purposes. Furthermore, a favourable system design is proposed to ensure high thermodynamic efficency of the heat recovery/heat pump process. Using the exhaust air as a heat source has not been done before, but is promising since indoor air is at a higher temperature than outdoor air in the heating case. Furthermore, the study examines to what extend an additional upstream heat exchanger unit can benefit the system by providing a fraction of the required heat at no extra cost. Finally, the result of ideal individual TEE control is compared against equal heat contribution of each TEE in the system.

Methodology

First, we present the theoretical basis of the peltier effect and the considerations underlying the model developed. Then, the assumptions made for the modelling and the setup variations considered are explained. Subsequently, the model is described from the highest to the lowest hierarchical level.

Peltier effect

Thermoelectric elements (TEE) contain semiconductor elements which become cold on one side when current flows. The heat absorbed on this side is released on the other side, resulting in a temperature increase compared to the environment. The basis for this is the Peltier effect. It states that a (small) temperature difference is formed at the contact point of two different metals as soon as a current flows through them. The cause of this effect is that the electrons in the two metals have different energy levels. To bridge the energy difference between the electrons, it is necessary to absorb or release energy from the thermal energy of the environment, depending on the difference in potentials. This results in an increase in temperature on the energy-dissipating side and a decrease in temperature on the energy-absorbing side.

The effect can also be observed in semiconductor devices, which are manipulated either by electron mismatches (p-doping) or excess electrons (n-doping) compared to a normal metal (undoped). If a current flows through a contact point between undoped metal and semiconductors, either heat absorption or heat emission occurs, depending on the type of semiconductor (p-doping/n-doping). By favourable design of these contact points, heat absorption on one side and simultaneous heat emission on the other side can be achieved. Such a combination is called a thermocouple. Several thermocouples can be connected in series to form thermoelectric elements, thereby increasing the heat flux transferred. The produced heat on the hot side is not due to the supply of electrical power alone, but also to transfer heat from the cold side [5]. This corresponds to the heat flows in a conventional heat pump, so that both systems appear comparable.

Considerations for system design

The following key issues are taken into account to enhance the performance of a thermoelectric building temperature control system: The highest efficiency of a TEE is at low utilized capacity and the smallest temperature difference possible between hot and cold side. For a given heating demand, this implies the following system design guidelines and challenges:

- *Split heating demand across multiple TEEs* When the entire heating demand can be distributed to more than one TEE, the individual heating demand is lowered, and efficiency increases in most cases. Since efficiency drops again, when heating demand becomes to small, it is not yet clear which number of TEEs is ideal for a certain heating demand.
- *Using exhaust air energy* Since the temperature difference between indoor air and outdoor air is positively proportional to the heating demand, outdoor air is not ideal as heat source. The energy in the exhaust air should be harnessed similarly to a heat recovery unit in order to reduce the temperature difference.
- *Few heat transfers* Every transfer of heat from one material/fluid to another is inevitably associated with a temperature offset. Therefore, it is necessary to limit the number of heat transfers to the lowest possible number. For space heating with air as a heat source, we assume that the only heat transfers are ”exhaust air to cold side heat sink”, ” cold side heat sink to TEE”, ”TEE to hot side heat sink” and ”hot side heat sink to supply air”.
- *Counter flow heat transfer* The mean temperature difference in a heat exchanger is smallest for counter flow heat exchangers. This means that the hottest point in the exhaust air stream should transfer its heat to the hottest point in the supply air stream. The coldest exhaust air point should correspond to the coldest supply air point, respectively. For our problem, this results in matching the last point of the heating process to the first point of the exhaust energy extraction process.
- *Good thermal connection* TEEs can generate very high specific heating and cooling powers of several hundred watts at 16 square centimeters [28]. It is therefore essential that these heat sources and sinks are particularly well thermally connected. Otherwise, the generated heat/cold cannot be sufficiently dissipated, which leads to an increase in temperature difference.
- *Thermal serial vs. thermal parallel operation* In principle, there are two main ways to interconnect several TEEs. First, one could stack them on top of each other to split the overall difference in temperature (operation in thermal series). This benefit is contrasted by the fact that the next TEE towards the hot side of the stack needs to absorb not only the heat absorbed by the previous TEE, but also the electrical energy that has been put in. So, the more TEEs the stack has, the smaller is the proportion of the emitted heat that was initially absorbed from the cold side. For n TEEs stacked on top of each other to be as effective as a single element, the combined power demand must not exceed the power demand of a single element at the same heat output. On average, the individual TEE in the stack needs to be n times as efficient as the single TEE due to the on average $1/n$ times smaller temperature difference. Second, one could place the elements next to each other in direction of the air flow. This has the downside of a relatively high temperature difference across the individual TEE. However, the possibility to establish a counter flow reduces the temperature differ-

ence and partly makes up for this negative effect. On the plus side, each element only has to transfer its electrical power demand into heat. This alternative is considered superior and the only one to be pursued further.

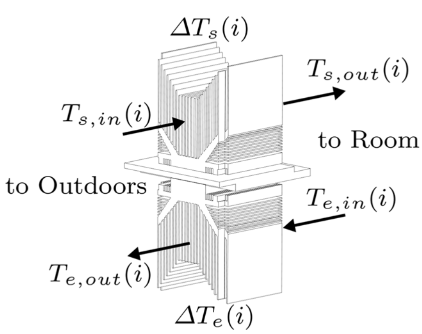
– *Over-temperature vs. heating demand* Regarding the over-temperature over room temperature, the optimal choice is not clear. A small over-temperature is beneficial to reduce the temperature difference over the individual TEE. On the other hand, decreasing over-temperature also implies that the air volume flow to cover the heating demand increases (as long as no recirculating air is taken into account). Since outside air must be heated to at least room temperature to have a positive effect, there is always a temperature offset that leads to increased heating power requirements for higher volume flows . Larger heating loads, again, reduce efficiency. The optimal over-temperature can therefore only be determined by the optimization algorithm.

The combination of these considerations results in the system design shown in Fig. 1.

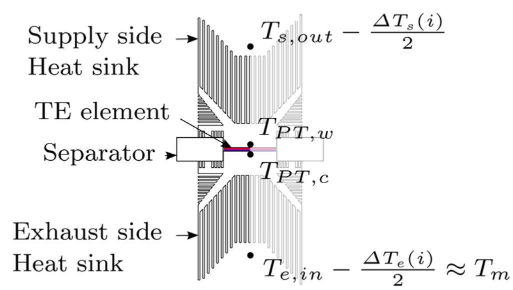
The system consists of a number of n TEEs in series. The relevant air flows are the supply air flow that is heated up (Index "s") and the exhaust air flow, from which energy is extracted (Index "e"). Each TEE is indexed by its number $i \in [1 : n]$ with TEE 1 being the element closest to the room. When looking at the individual TE group (TEE plus heat sinks), two temperatures for each air flow are of interest: The temperature before (Index "in") and after (Index "out") the airflow passes the heat sink (s. Fig. 1).

Modelling assumptions and variations

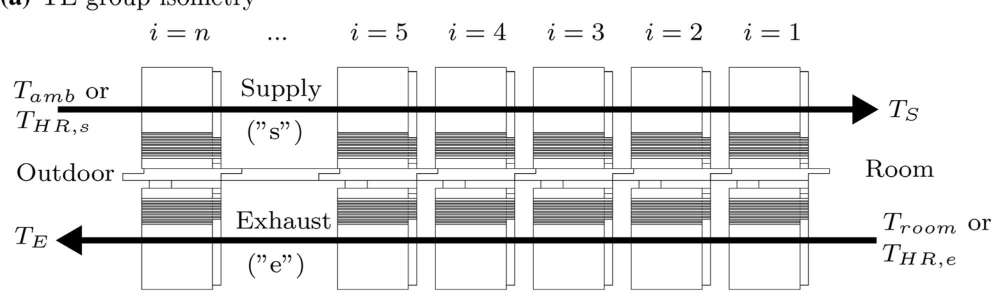
In order to numerically model the proposed system design, the following assumptions are made: Steady state is assumed for operating and outdoor conditions. The aim of this simplification is to keep the computational effort manageable and to derive favourable operating points. The heat sink's thermal capacity is about 910J/K. A Peltier element with 100W of thermal power needs 9,1s/K to heat this heat sink. This is fast compared to a water-filled radiator with a response time of about 42s/K and heat pumps that need approximately 15 minutes to start [29]. Therefore, the short term transient behaviour can been neglected. To estimate the accuracy of the steady state simplification in the long term, the thermal load is allocated according to an annual heating capacity curve to approximate transient operation [30]. The design point in steady state is chosen so that the annual heat output is the same. Since TEE are usually operated far below their nominal operating power, it is assumed that higher loads in transient operation can still be met at higher temperature differences and worse coefficients of performance. Assuming optimum operating conditions for a given heat demand and neglecting short term effects (s. below), the energy demand in transient operation can be derived based on the data sheet



(a) TE group isometry



(b) TE group sectional view



(c) Schematic for combined TE groups

Fig. 1 Schematic system design

Table 1 Variations

Variable	Variation; Range	Unit
Q_h	50; 100; 500; 1000; 2000; 3000	W
Efficiency heat recovery	0; 0.8	
TEEs in system	[1:50]	pc(s).
T_{amb}	[−10 : 19]	$°C$
Heating load distribution	Unevenly; evenly	

Table 2 Corresponding room sizes

Heating load	Insulation standard	
	Passive	Low energy
	$10W/m^2$	$30W/m^2$
100W	$10m^2$ (bathroom)	$3.33m^2$ (small bathroom)
1000W	$100m^2$ (large apartment)	$33.33m^2$ (student apartment)
2000W	$200m^2$ (storey of MFH)	$66.67m^2$ (apartment)

of the TEE. Results and implications Conclusions from this analysis can be found in 3.1.

Further assumptions in the study are:

- Equal distribution of ohmic losses across a TEE
- Homogeneous air temperature perpendicular to flow direction
- Neglect of AC-DC conversion loss and hydraulic losses
- No heat losses via system walls and the distribution system
- The Thomson effect is neglected [31]
- No air leakage in the room

The optimization is carried out with several variations of the boundary conditions, to generate optimal results for different scenarios (s. Table 1). Rooms sizes corresponding to the considered heating loads at insulation standards ”Passive house” (passive) and ”low energy house” (low energy) [32] can be found in Table 2.

Optimization model

The objective function of the optimization problem is to minimize the total electrical effort $P_{el,total}$ for a given heating demand Q_h and number of TEE n. To achieve this, the algorithm is enabled to choose the overtemperature over room temperature of the supply air into the room. The lower limit of this supply temperature T_s is the room temperature $T_{s,min} = 20°c$ and the upper limit is set at $T_{s,max} = 80°c$. Furthermore, the heating demand $\dot{Q}_h$ can be distributed unevenly among the individual TEEs. The share of the entire heating demand for every TEE $\dot{Q}_{h,i}$ is therefore also an input of the optimization problem. The resulting minimization function is:

$$min\left(P_{el,total}(x)|x \in \Omega\right) \tag{1}$$

$$x = \begin{bmatrix} \frac{T_S - T_{S,min}}{T_{S,max} - T_{S,min}} \\ Q_1/Q_h \\ Q_2/Q_h \\ \dots \\ Q_n/Q_h \end{bmatrix} \tag{2}$$

s.t.

$$0 < \Omega < 1 \tag{3}$$

$$\sum_{i=2}^{n} \Omega(i) = 1 \tag{4}$$

The basic outlines of the algorithm are shown in the program flow chart (Fig. 2).

Model of air flow and non TE components

The total systems electrical demand $P_{el,total}$ for one configuration is calculated as follows: First, inlet ($T_{s,in}$) and outlet temperature of the supply stream ($T_{s,out}$) are defined. The necessary mass flow rate $\dot{m}$ to cover the heating demand $\dot{Q}_h$ is defined and is assumed to be the same in the heating and cooling process since leakage in the room is not considered.

$$\dot{m} = \frac{\dot{Q}_h}{c_{p,air} \cdot [T_S - T_{room}]} \tag{5}$$

The process is subject to temperatures at the start of the supply and exhaust air channel. In case there is no heat recovery unit, these temperatures are the ambient temperature T_{amb} and the room temperature $T_{room} = 20°C$. When a heat recovery unit is considered, we assume that it preheats the supply air to $T_{HR,s} = T_{amb} + \eta_{HR} \cdot \left(T_{room} - T_{amb}\right)$. Since mass flow rates are equal for supply and exhaust air flow, the exhaust air flow is pre-cooled to $T_{HR,s} = T_{room} - \eta_{HR} \cdot \left(T_{room} - T_{amb}\right)$ before entering the system. The humidity change of air is not taken into account.

To calculate heat transfer from heat sink to air and vice versa, the area between the heat sink fins and the share of air flowing through the area needs to be known. It is assumed that 80% of the mass flow rate is directed directly through the area in between the fins. We model a heat sink of width w_{HS},

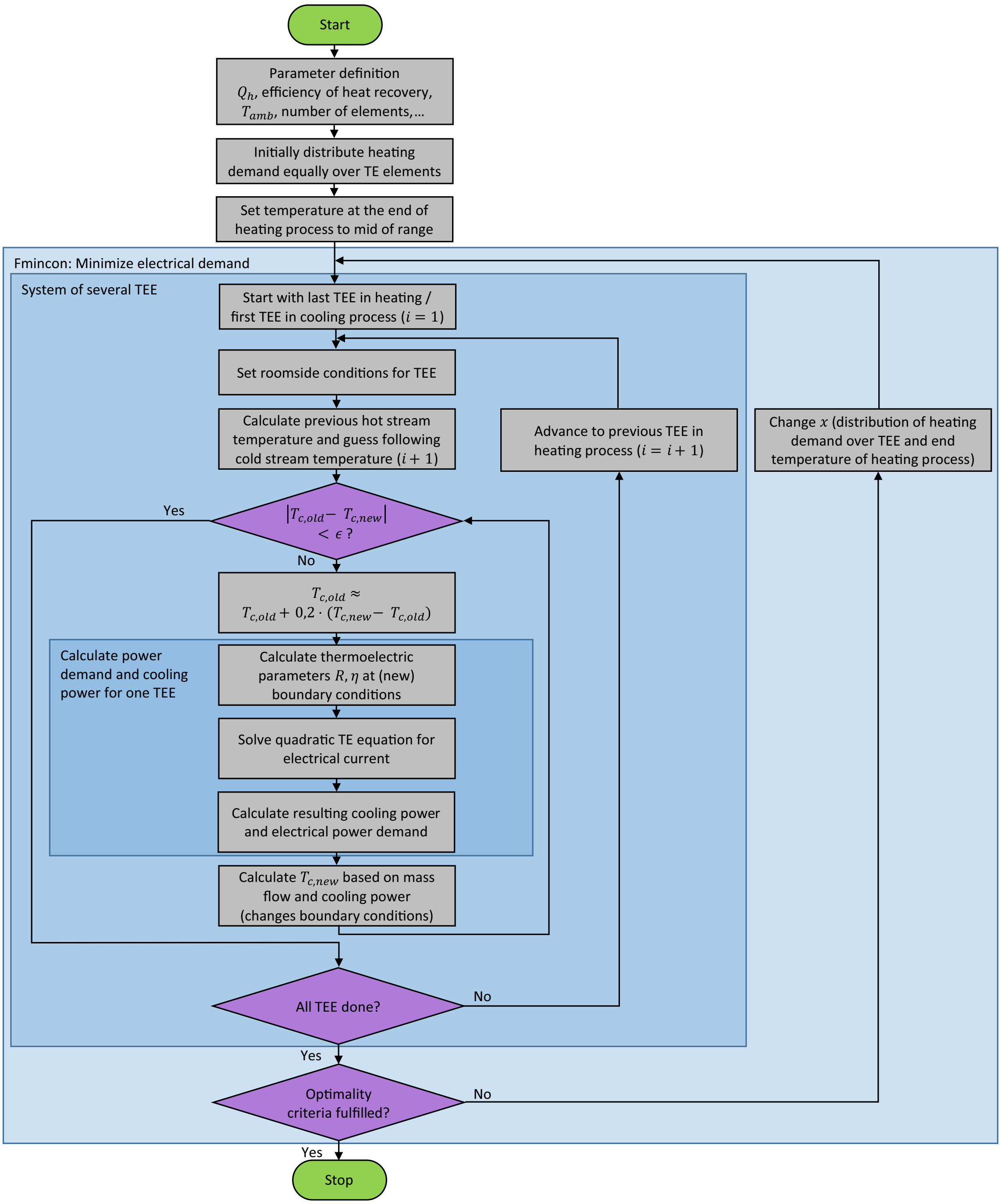


Fig. 2 Flow chart containing the main operations of the optimization process

height h_{HS} and length l_{HS} with a number of fins N_{fin} which all have width w_{fin}, height h_{fin} and length l_{fin} based on [33]. For this heat sink, the width between two fins $w_{no,fin}$ is given by

$$w_{no,fin} = \frac{w_{HS} - N_{fins} \cdot w_{fin}}{N_{fins} - 1} \tag{6}$$

Conclusively, the velocity of the air between fins v is

$$v = \frac{0.8 \cdot \dot{m}}{\rho_{air} \cdot (N_{fins} \cdot w_{nofin} \cdot h_{fin})} \tag{7}$$

The heat transfer area from the heat sink to air consists of the fins area A_{fin} and the area between the base of the fins A_{base}.

$$A_{base} = (N_{fins} - 1) \cdot w_{no\,fin} \cdot l_{HS} \tag{8}$$

$$A_{fin} = 2 \cdot h_{fin} \cdot l_{HS} \tag{9}$$

In order to calculate the total heat transfer coefficient α of one heat sink, the total thermal resistance R_{tot} is calculated using the convective heat transfer coefficient α_{conv} as well as fin efficiency η_{fin} and heat conductivity of the heat sink λ_{HS} according to [33].

$$\alpha = \frac{1}{R_{tot} \cdot A_{PT}} \tag{10}$$

$$R_{tot} = \frac{1}{\alpha_{conv} \cdot (A_{base} + N_{fins} \cdot \eta_{fin} \cdot A_{fin})} + \frac{h_{HS} - h_{fin}}{\lambda_{HS} \cdot w_{HS} \cdot l_{HS}} \tag{11}$$

The convective heat transfer coefficient α_{conv} is calculated based on Nusselt number Nu which contains a modified Reynolds number Re^*.

$$\alpha_{conv} = Nu \cdot \frac{\lambda_{air}}{b_{fins}} \tag{12}$$

$$Nu = \left(\frac{1}{Nu_1^3} + \frac{1}{Nu_2^3} \right)^{-1/3} \tag{13}$$

with

$$Nu_1 = Re^* \cdot Pr_{air} \cdot 0.5 \tag{14}$$

$$Nu_2 = 2/3 \cdot \sqrt{Re^*} \cdot Pr_{air}^{(1/3)} \cdot \sqrt{1 + \frac{3.65}{\sqrt{Re^*}}} \tag{15}$$

$$Re^* = \frac{\rho_{air} \cdot v \cdot b_{fins}^2}{\eta_{air} \cdot l_{HS}} = Re \cdot \frac{b_{fins}}{l_{HS}} \tag{16}$$

Fin efficiency is calculated with Eq. 17.

$$\eta_{fin} = \frac{tanh(m \cdot h_{fin})}{m \cdot h_{fin}} \tag{17}$$

with

$$m = \sqrt{2 \cdot \frac{\alpha_{conv}}{\lambda_{HS} \cdot t_{fin}}} \tag{18}$$

The resulting total heat transfer coefficient α is used to calculate the temperature differences necessary to ensure the defined heat flux ($\dot{Q} = A\alpha\Delta T$).

A mean thermal conductivity λ_m of the area separating the supply and exhaust air flow A is needed to model the thermal short circuit heat flux. The area A minus the area of a TEE A_{PT} gives the area which is taken up by the separator A_{noPT}. Based on these areas and thermal properties of separator as well as TEE, the mean thermal conductivity λ_m can be determined according to equation 19.

$$\lambda_m = \frac{h_{PT}}{A_{PT}} \cdot \left(\frac{A_{PT} \cdot \lambda_{PT}}{h_{PT}} + \frac{A_{noPT} \cdot \lambda_{plastic}}{h_{noPT}} \right) \tag{19}$$

Once these parameters are known, the required electrical demand P_{el} in this configuration can be calculated by the following algorithm: For every TEE $i \in [1 : n]$, its warm side temperature $T_{PT,h}$ is calculated by adding the temperature difference caused by the heat transfer to the mean air temperature at the supply side

$$T_{PT,h} = \frac{T_{s,out} + T_{s,in}}{2} + \frac{Q_h(i)}{\alpha \cdot A_{PT}} \tag{20}$$

Since cold side TEE temperature depends on absorbed heat Q_c which again depends on the TEE cold side temperature $T_{PT,c}$, iteration is necessary and therefore an initial temperature estimation. For the first iteration, it is assumed that the temperature difference between the air flow and TEE is equal in amount to the warm side temperature difference, but reversed in sign.

$$T_{PT,h} - \left(T_{s,out} - \frac{\Delta T_s(i)}{2} \right) = -\left[T_{PT,c} - \left(T_{e,in} - \frac{\Delta T_s(i)}{2} \right) \right] \tag{21}$$

$$\Leftrightarrow T_{PT,c} = T_{e,in} + T_{s,out} - \Delta T_s(i) - T_{PT,h} \tag{22}$$

If, on the other hand, calculations of previous elements have been carried out already, the TEE cold side temperature is estimated as follows:

$$T_{PT,c} = T_m - \frac{\dot{m} \cdot c_{p,air} \cdot \Delta T_e}{\alpha \cdot A_{PT}} \tag{23}$$

with

$$T_m = \frac{T_e(i-1) + T_e(i)}{2} \tag{24}$$

$$\Delta T_e = T_e(i-1) - T_e(i) \tag{25}$$

Starting from these initial conditions and with an initial difference in old and new cold side temperature of $1K$, the following iteration is carried out until the difference falls below ϵ:

1. Narrow the gap between newly calculated and old cold side temperature by roughly 20% of their difference: $T_{PT,c,old} \approx T_{PT,c,old} + 0.2 \cdot (T_{PT,c,new} - T_{PT,c,old})$
2. Calculate resulting electric power demand P_{el} and cooling power $\dot{Q}_c$ in separate function (Subsect. 2.6).
3. Calculate necessary temperature difference ΔT_e to ensure cold flux $\dot{Q}_c$.
4. Recalculate cold side TEE temperature according to $T_{PT,c,new} = T_{e,in} - \frac{\dot{Q}_c}{\alpha \cdot A_{PT}} - \frac{\Delta T_e}{2}$.
5. Check the termination condition and restart if not fulfilled

If the criterion is fulfilled, the results are saved and P_{el} is returned as electrical demand for TEE i. The individual demands of all TEE sum up to the total systems demand $P_{el,total}$.

Model of heat flux in thermoelectric element

This section describes the method of the function to calculate the thermal and electrical behaviour of TEE i based on [5, 12]. The following equation can model the emitted heat of a TEE:

$$Q_h = \eta \cdot I \cdot T_{PT,h} + \frac{R \cdot I^2}{2} - \lambda_m \cdot A_{PT} \cdot \frac{\Delta T}{h_{PT}} \tag{26}$$

Heat extraction can be modelled analogously:

$$Q_c = \eta \cdot I \cdot T_{PT,c} - \frac{R \cdot I^2}{2} - \lambda_m \cdot A_{PT} \cdot \frac{\Delta T}{h_{PT}} \tag{27}$$

The three terms represent different ways of heat flux. $\eta \cdot I \cdot T_{PT,c}$ or $\eta \cdot I \cdot T_{PT,h}$ stands for the heat extracted/ emitted via the Peltier effect. $R \cdot I^2$ is electric power consumption of the system (ohmic losses) and $\lambda_m \cdot A_{PT} \cdot \frac{\Delta T}{h_{PT}}$ is the heat conduction term. The heat extracted due to the Peltier effect at the cold side junctions is proportionally related to Seebecks coefficient η, current I and the junctions' temperature (cold side temperature $T_{PT,c}$). On the hot side, the emitted heat is proportional to the temperature of the hot side junctions (hot side temperature $T_{PT,h}$).

It is assumed that power loss is distributed equally across the TEE and therefore a heat source of $\frac{R \cdot I^2}{2}$ in each case benefits the heat production on the warm side and reduces the cooling power on the cold side by the same amount.

For $\Delta T = T_{PT,h} - T_{PT,c} > 0$ it is visible that heat conductivity decreases the heating power as well as the cooling power. This decrease is because heat always flows against the desired temperature gradient.

Some parameters in these equations can be determined from the datasheets of the TEE or the experimental setup. All used parameters can be found in Table 3.

Linear regression based on three different temperature levels ($T_{PT,h} = 25, 50, 75°C$), six different electrical currents ($2A$, $4A$, ...$12A$) and temperature differences ΔT ranging from $0K$ to $90K$ is carried out for the TEE "European Thermodynamics APH-199-14-08-E" [28]. The result is shown below:

$$R = 0.007872\Omega/K \cdot T_m - 0.6102\Omega \tag{28}$$

$$\eta = 0.00009767V/K^2 \cdot T_m + 0.05294V/K \tag{29}$$

$$\lambda_{PT} = -0.011W/(mK^2) \cdot T_m + 6.58W/(mK) \tag{30}$$

With a given heating demand $\dot{Q}_h$ and hot side temperature $T_{PT,h}$, we can calculate the electric current that is necessary to generate the desired heat by solving the quadratic equation 26 for the positive solution.

Once the current is known, the extracted heat $\dot{Q}_c$ can be calculated by equation 27 and the electrical power demand can be calculated by:

$$P_{el} = \eta \cdot I \cdot \Delta T + R \cdot I^2 \tag{31}$$

Therefore, extracted heat and electrical power demand can be calculated, when the higher level of the algorithm provides the following inputs:

Individual heating demand $\dot{Q}_{h,i}$
TEE temperatures $T_{PT,h}$ and $T_{PT,c}$
Heat conductivity λ_m
Area A_{PT} and height h_{PT} of the TEE
Coefficients for linear parameter approximation

Results and discussion

Steady state assumption

The relative deviation in annual energy demand resulting from the steady state assumption is displayed in Fig. 5 for different, set temperature differences via the TEE in steady state. It can be seen that the transient simulation results in significantly lower energy demand at small temperature differences in steady state operation. High temperature differences in steady state operation (that are not desirable) lead to even higher energy demand in transient operation. This underlines the importance of reaching small temperature differences. A subsequent transient optimization is recommended in order to design a control strategy and system size for given use cases.

Heating solely with thermoelectrics

Figure 3 shows the calculated coefficients of performance (COP) for various heating loads and numbers of TEE in the system at ambient temperatures ranging from −10 to 19 °C. The results in the top graphs are obtained solely by thermoelectric operation. The bottom results include the usage of an additional upstream heat recovery unit. The secondary y-axis shows the corresponding surface area of the implemented TEEs without heat sinks. The yellow and green areas in the diagram show operating conditions in which the system supplies more thermal power than its electrical power demand. The system reaches COP greater than six in the best case at almost ambient temperature and very small heating demand. In the worst case, COP drops below 0.75 for low ambient temperatures and few TEE. For a heating demand of 100 W, nine TEEs are sufficient to supply more heat than direct electric heating. It is clearly visible that COP decreases with increasing thermal load and increases with ambient temperature: The average COP drops from 1.81 at a thermal load of 100W to 1.29 at 1000W to 1.14 at 2000W. At the same time, the average COP increases from 0.94 at an ambient temperature of -10°C to 1.06 at 0°C to 1.34 at 10°C

Furthermore, a higher number of TEE in the system leads to an increase in COP in all cases. These observations are as expected since they either lead to smaller individual heating demand for a single TEE or reduce the overall temperature difference. The required number of TEE to reach a COP of one or above does increase less than linearly with heating demand. Therefore, the necessary number of elements per provided heating power decreases for increased system size, suggesting that the technology is subject to economies of scale. This is true for most ambient temperatures except temperatures very close to room temperature.

Benefit of a heat recovery unit

A heat recovery unit proves beneficial in most operating conditions. Figure 4 shows the difference in COP for a system with additional heat recovery compared to the purely thermoelectric system. Especially for high heating demands, the heat recovery contributes to the systems overall efficiency in almost every configuration. In general, the positive effect of

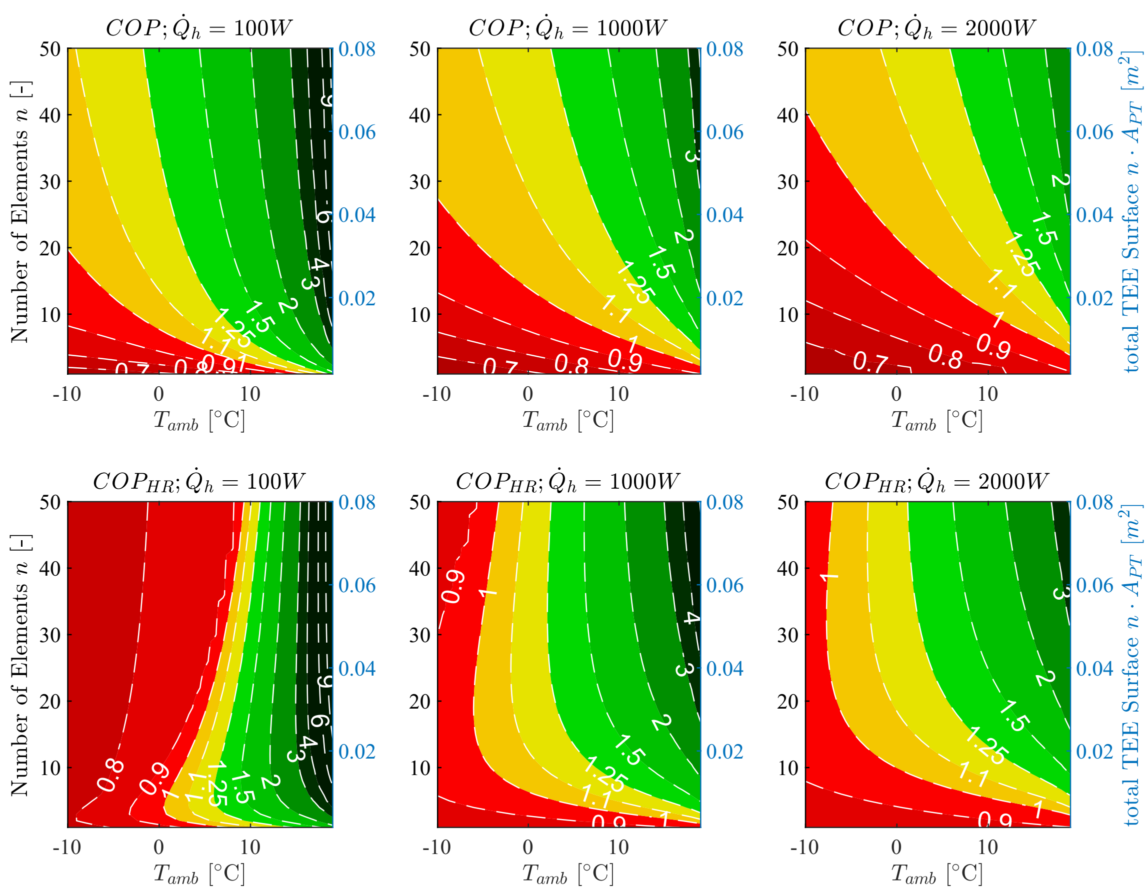


Fig. 3 Coefficient of performance for various numbers of TEEs, various ambient temperatures T_{amb} at a total heating demand of $\dot{Q}_h = [100W, 1000W, 2000W]$ Top: no heat recovery unit. Bottom: with heat recovery unit

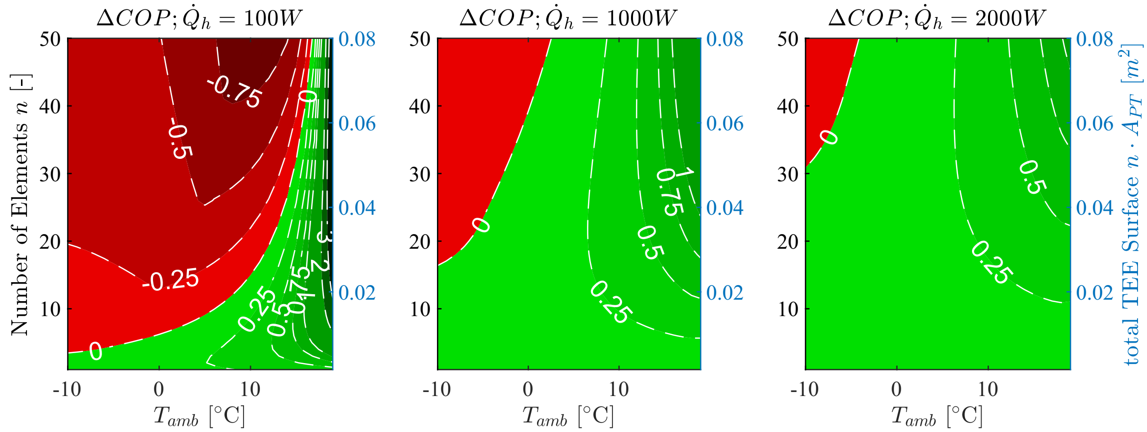


Fig. 4 Change in the coefficient of performance due to the use of an upstream heat recovery unit for various numbers of TEEs, various ambient temperatures T_{amb} at a total heating demand of $\dot{Q}_h = [100W, 1000W, 2000W]$

a heat recovery unit decreases with decreasing ambient temperature. This decrease is since an upstream heat recovery unit reduces the exhaust side inlet temperature and increases the supply side inlet temperature for the TEE-part of the system. Hence, the medium temperature difference for the TEE increases along with the temperature difference over the heat recovery unit. The most significant benefit can therefore be harnessed for small temperature differences in the heat recovery. The efficiency loss due to higher temperature differences is greater for small thermal outputs. The cause of this efficiency loss is that the proportion of the heat flux lost in the thermal short circuit in relation to the total power is greater. For big numbers of TEE in the system, the efficiency loss due to the higher temperature difference outweighs the benefit of the heat recovery unit. Comparing the different heating loads shows that the number of TEEs at which a heat recovery is no more reasonable increases less than linearly with increased heating load or system size. The COP of a system consisting of 10 TEEs is improved by 0.2 on average,

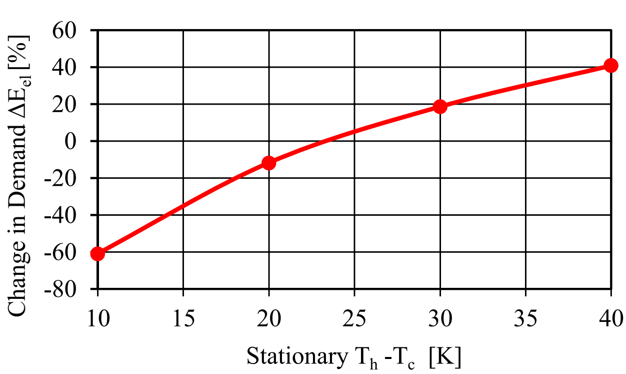


Fig. 5 Estimation of change in calculated electrical energy demand ΔE_{el} for transient simulation instead of the steady state simplification for one TEE

while it improves only by 0.16 on average for 20 TEEs and by 0.1 on average for 30 TEEs. Therefore, the usage of a heat recovery unit is most useful for small systems (i.e. few TEE) and subject to diseconomies of scale regarding the number of TEEs in the system.

Impact of evenly distributed Heating load

When the optimizer can choose the distribution of the heating load, the TEE closest to the room ($i = 1$) is attributed with the highest portion of the heating demand. This reduces the air temperature for the previous TEE in the supply stream and increases their efficiency. Figure 6 shows in the upper part how much of the total heating demand is attributed to the first TEE in relation to the average demand per element for the different configurations. In the lower part of Fig. 6, the resulting gain in system COP compared to the COP with equally distributed heating load is shown.

Low ambient temperatures lead to a stronger gradient in heat distribution. An unevenly distributed demand becomes reasonable once the system consists of more than one TEE. The gain in COP is not significant compared to the benefits of a heat recovery system: The best case achieves an improvement in COP of 0.0715 by distributing the heating load in an optimal uneven way. Adding a heat recovery unit increases the COP by up to 4.5509 in the best case and on average by 0.3404 for all cases with $\Delta COP > 0$. Therefore, it is questionable if the effort of an individual heat distribution is worthwhile or if, for example, an equal voltage supply is sufficient. However, an equal voltage supply would not result in equal distribution of the heating load the since boundary conditions and therefore efficiency varies between the individual TEE. Hence, the appropriate control is the subject of further studies.

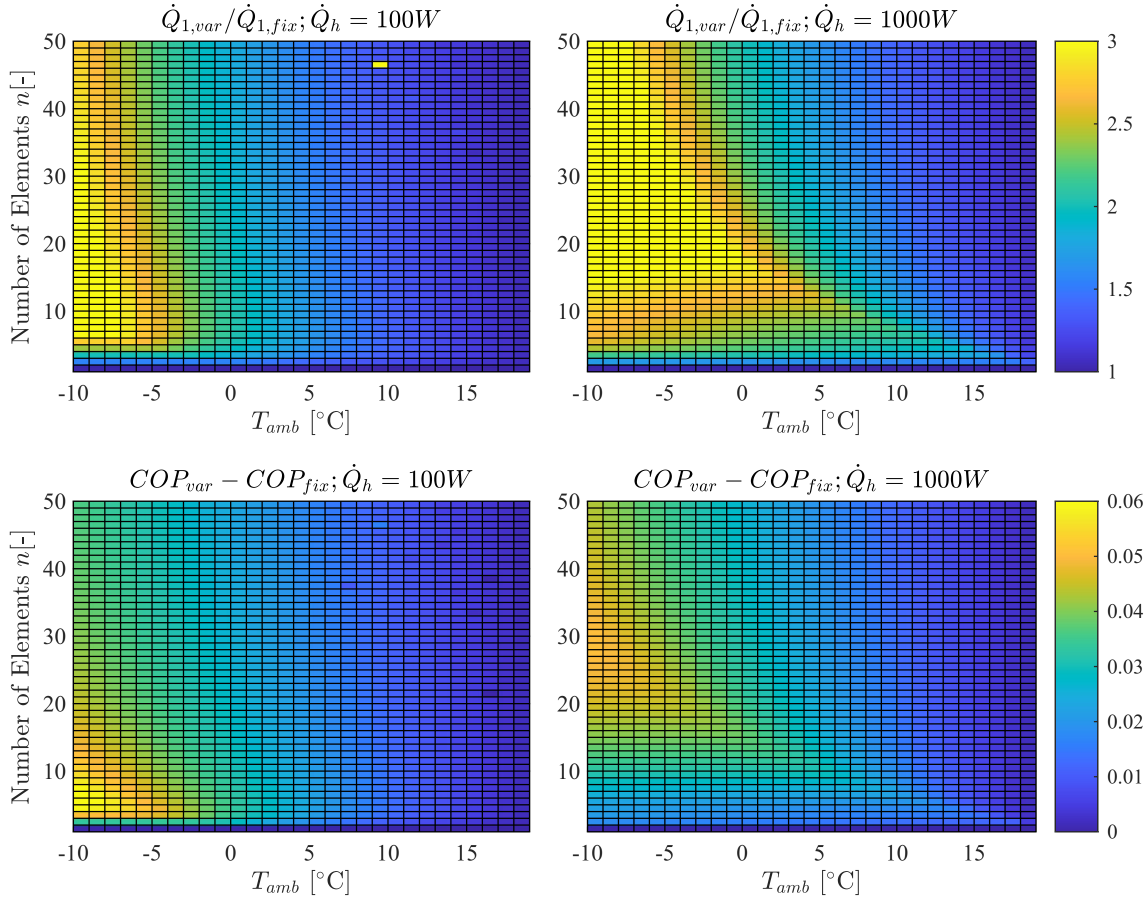


Fig. 6 Top: Heating load at roomside TEE of the supply stream for uneven heat distribution $\dot{Q}_{h,1,var}$ related to heating demand at even heat distribution $\dot{Q}_{h,1,fix}$ for various numbers of TEE, various ambient temperatures T_{amb} at a total heating demand of $\dot{Q}_h = [100W, 1000W]$ Bottom: Change in the coefficient of performance due to the uneven heat distribution compared to even distribution

Suggestions for system design

Figure 7 shows the resulting COP for 10, 20 and 40 TEEs in the system with and without upstream heat recovery unit. It is visible that for smaller systems with e.g. 20 TEEs, the heat recovery unit enables the efficient provision of heat at high heat fluxes and ambient temperatures as low as -6 °C. Furthermore, it increases efficiency significantly for high heat fluxes and at higher ambient temperatures. Without heat recovery unit, efficient heat supply is only possible at lower powers, but even for ambient temperatures as low as -10 °C. For larger systems of 40 TEEs and more, efficiency at high heat fluxes using a heat recovery unit increases even further. At lower heat fluxes, performance is no longer increased by more TEEs and even decreased in low temperature, medium heat flux conditions. On the other hand, the system without heat recovery unit is feasible to supply heat for almost all configurations and outperforms the heat recovery system at low heat fluxes. Without a heat recovery unit, the feasible temperature band decreases with heating demand, while it increases using a heat recovery unit. Therefore, it is absolutely reasonable to install a heat recovery unit if few TEEs generate high heat fluxes. Nevertheless, suppose one can afford to integrate more TEEs in the system. In that case, it might not be necessary to additionally add a heat recovery system since the benefits vanish for larger systems due to economies of scale for the TEE technology and diseconomies of scale for the combination with a heat recovery unit (Table 3).

Conclusions

This study has shown that thermoelectricity poses a great potential for efficient heat generation for small applications and can be further improved by combining it with a heat recovery unit. It proves that even small systems with e.g. ten TEEs can provide heat with COP > 1 at temperatures below 0 °C, when operating at their design point. This makes it superior to direct current heating for many operating conditions. Considering the additional functionality of providing cooling and ventilation as well as

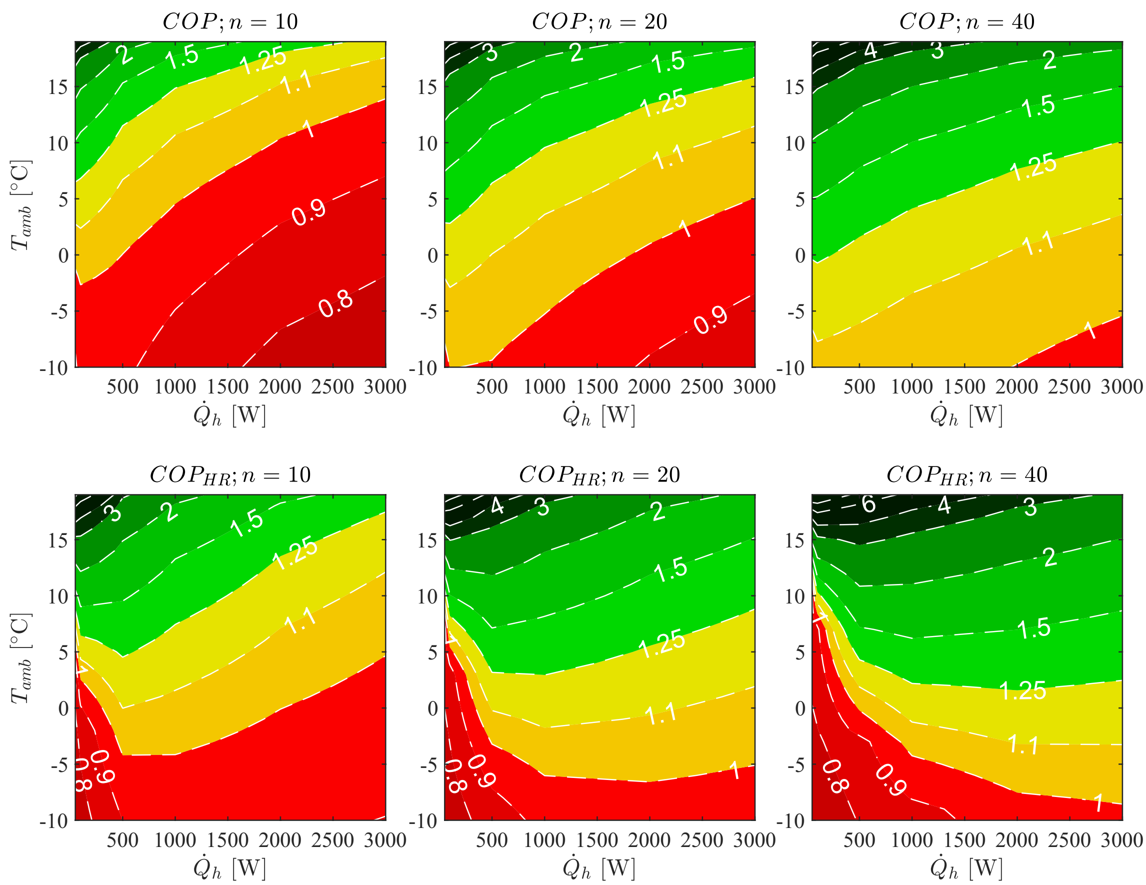


Fig. 7 Coefficient of performance for various ambient temperatures T_{amb} and various heating demands $\dot{Q}_h$ at $n = [10, 20, 40]$. Top: without heat recovery unit. Bottom: with heat recovery unit

Table 3 Parameters

Variable	Value	Unit
A_{PT}	0.0016	m^2
h_{PT}	0.0035	m
$\lambda_{plastic}$	0.24	$W/(mK)$
d_{pipe}	0.02	m
A	0.01	m^2
ϵ	10^{-4}	–
w_{HS}	0.08	m
h_{HS}	0.07	m
l_{HS}	0.075	m
h_{fin}	0.06	m
t_{fin}	0.001	m
N_{fins}	21	pc(s).
Pr_{air}	0.72	–
λ_{air}	0.0262	W/(m K)
ρ_{air}	1.2	kg/m^3
η_{air}	0.0000171	kg/(m s)
$c_{p,air}$	1000	kJ/(kg K)
λ_{HS}	200	W/(m K)

the almost complete avoidance of noise and refrigerants make TEEs a valuable alternative even for heat pumps. The results regarding the COP under different external circumstances can be used to dimension an air temperation system (e.g. a system including 50 TEE provides a COP > 1 in all examined circumstances). Further studies are needed to consider condensation of water from the cold stream air since it possibly further improves the performance. The avoidance of heating losses via air exchange by the TEE ventilation system should also be taken into account to underline the TEE-systems value compared against conventional space heating technologies. A practical and efficient control system needs to be investigated to harness the full potential of the technology. Studies regarding the transient behaviour of selected systems are also needed in order to derive a seasonal energy efficiency ratio. Improved heat transfer techniques like heat pipes and liquid cooling pose another interesting field of research. A test setup to verify the derived conclusions is suggested.

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Declarations

Conflict of interest The authors declare that they have no conflict of interest.

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