ON THE POTENTIAL OF USING HEAT FROM SOLAR THERMAL COLLECTORS FOR HEAT PUMP EVAPORATORS

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Abstract

Solar thermal collectors can be used in combination with heat pumps to cover the heat demand for space heating and domestic hot water preparation. Different concepts exist for the combination of these two components into a system concept. Some of these concepts offer the ability to switch from using heat from the solar collectors directly to serve the demand to using heat from the solar collectors indirectly as a heat source for the evaporator of the heat pump. In the same system concept, the heat pump may be able to switch from using heat from the solar thermal collector to using an alternative low temperature heat source, such as ambient air. In this paper a general mathematical relationship is derived for determining whether using heat from solar collectors for the evaporator of the heat pump instead of using it directly is beneficial for the energetic performance of these systems. It is shown that there is a limit for the solar irradiation on the collector field above which using collector heat for the evaporator of the heat pump instead of using it directly is not advantageous. This irradiation limit depends on the characteristic performance curves of the solar collector and of the heat pump, as well as on the operating conditions, above all on the temperature levels of the heat sink and the different heat sources. Based on dynamic annual simulations, examples are shown for the maximum amount of heat that can be delivered from the collector to the heat pump evaporator at times where this mode of operation is of advantage for the performance of a solar and air source heat pump heating system. Both the mathematical relationship and the examples show that using solar collector heat for the evaporator of a heat pump is more beneficial for systems that operate with a large temperature difference between the non-solar heat source and the heat demand.

1. Introduction

In central European countries, both the use of solar thermal collectors and the use of heat pumps for space heating and domestic hot water (DHW) preparation have increased in recent years (EHPA 2010; Weiss & Mauther 2011). Along with these developments, also combined solar and heat pump systems are now increasingly demanded and advertised by manufacturers (Trojek & Augsten 2009). An overview of different solutions for the hydraulic connections and the control of such systems used by different manufacturers and presented in different research projects has been given by several authors (Trinkl et al. 2004; Henning & Miara 2008; Haller et al. 2010, Frank et al. 2010). Some of the systems on the market today use heat from solar thermal collectors not only directly to serve the heat load (Fig. 1a), but also indirectly by delivering heat to the evaporator of the heat pump (Fig. 1b). In most of these systems, the heat pump has the possibility of either using heat from the solar collectors or using a different heat source such as ambient air, which is then often the main heat source for the heat pump. These are the systems that are in the focus of this paper.

Using solar heat for the evaporator of the heat pump instead of using e.g. ambient air may increase the coefficient of performance (COP) of the heat pump because the collector delivers heat at a generally higher temperature level than the ambient air. At the same time, the efficiency of solar collectors may be increased in this operation mode since they run at considerably lower temperatures serving the evaporator of the heat pump than they would run at if they were serving the required temperature of the final heat demand directly. However, a higher efficiency of the collector together with a higher COP of the heat pump are not yet a guarantee for a higher system performance. Citherlet et al. (2008) presented annual simulation studies for systems where heat from the solar collector can be used for the evaporator of the heat pump instead of using ambient air. They found only little improvement of the overall system's performance factor (PF_{sys}) in terms of useful heat delivered ($Q_{u,sys}$) over electric energy used ($W_{el,sys}$) as defined by eq. 1.

$$PF_{sys} = \frac{Q_{u,sys}}{W_{el,sys}} \quad \text{(eq. 1)}$$

Karagiorgas et al. (2010) found that using heat from a solar thermal collector for the heat pump increases the system performance only in the morning and in the evening hours of an example day and presented a curve for switching from indirect to direct use of collector heat depending on irradiation and outdoor temperature, valid for their particular collector and heat pump characteristics only. A general mathematical explanation for the point of switching from direct to indirect heat use or vice-versa and conclusions drawn from such an analysis has not been found in the literature.

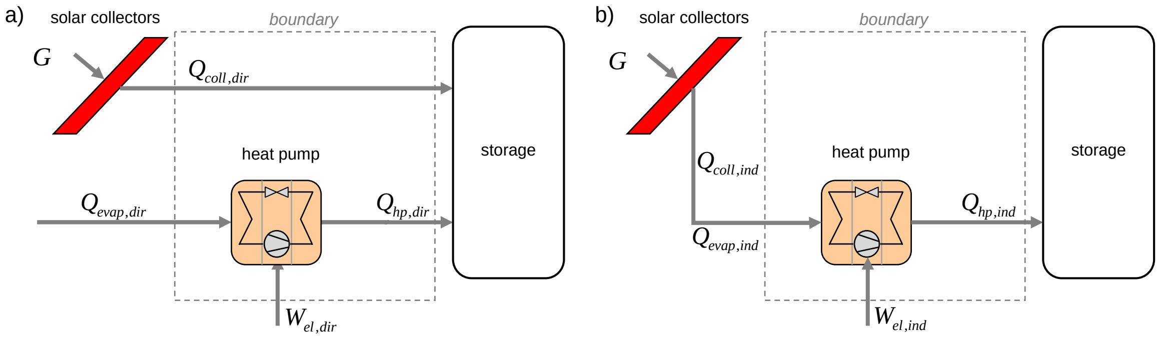

Fig. 1: Energy flows in a combined solar and heat pump system illustrating a) direct collector heat use and b) indirect collector heat use (adapted from Haller & Frank 2010).

In this paper, the requirements that must be met in order to reach a higher system performance by using solar collector heat for the evaporator of the heat pump instead of using it directly to serve the final heat demand is analyzed for systems where the heat pump is a dual source heat pump that can use not only collector heat, but also ambient air as a heat source, for days where there is an average heat demand that is higher than the heat that could be delivered by the solar thermal collectors in direct operation. The research presented is embedded into a joint Task 44 / Annex 38 of the International Energy Agency's Solar Heating and Cooling Programme and the Heat Pump Programme (Hadorn 2010). Part of the research presented in this paper is currently under review for the ISES Solar Energy Journal and/or has been presented in German language at the 21st OTTI Symposium Thermische Solarenergie in Bad Staffelstein, Germany (Haller & Frank 2011).

2. Methods

2.1 General assumptions

For the investigated system it is assumed that the solar collectors, the heat pump and the storage are hydraulically connected in a way that collector heat may either be used directly (Fig. 1a) or serve the evaporator of the heat pump (indirect use, Fig. 1b). The heat pump has the possibility to use a different heat source instead of the collector heat (e.g. air, ground heat, etc.). The crucial question for the control of such a system is when to switch from direct collector heat use to indirect heat use and vice-versa. In order to answer this question, it is assumed that indirect collector heat use is better if the system's performance factor for this operation mode is better than the system's performance factor in direct heat use mode (eq. 2).

$$PF_{sys,ind} > PF_{sys,dir} \qquad \text{(eq. 2)}$$

The two modes of collector operation (direct and indirect) can be compared if the amount and temperature of heat delivered to the storage is identical for both cases. This requires that enough energy is demanded from the system in order to use the heat within a short time, such that the storage temperatures of both operation modes are also identical.

2.2 Simulation studies

All simulations have been performed with the software TRNSYS (Klein et al. 2004) and with simulation time steps of 2 minutes. A simplified hydraulic schematic for the simulated combined solar and heat pump system is shown in Fig. 2. In this system, the solar collectors deliver heat to the lower part of a heat store with an immersed heat exchanger. The heat pump charges the upper part of the storage for domestic hot water preparation, and the middle section of the storage for space heating. Domestic hot water preparation is done with an external heat exchanger, and space heating is prepared with a low temperature heating system with controlled supply temperature.

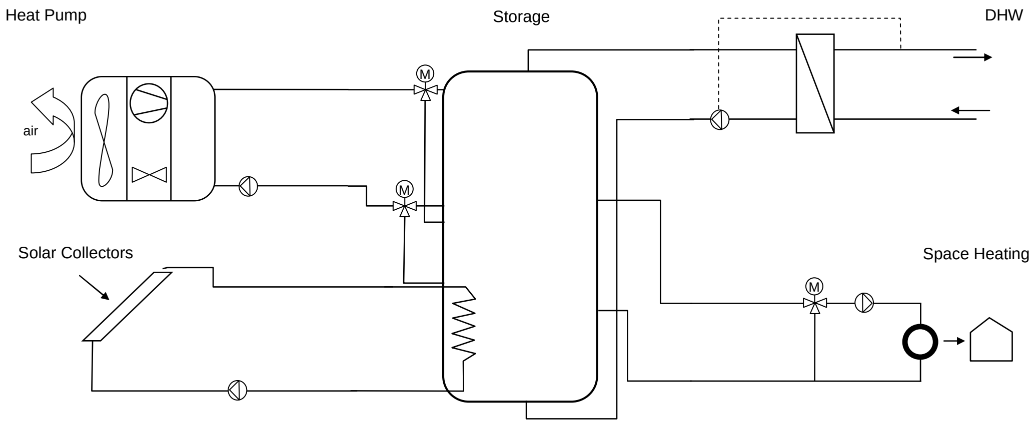

Fig. 2: Simplified schematic of the combined solar & heat pump heating system.

The space heat load and the domestic hot water profile were taken from the IEA-SHC Task 32 reference system's SFH 100 building (Heimrath & Haller 2007). The supply and return temperature of the heating system have been lowered to 35/30 °C in order to match the heat pump application. A general overview of the key values for the climates of Zurich and Madrid is given in Tab. 1.

Tab. 1: Key parameters of the reference heating system SFH100 for the climates of Zurich and Madrid.

Parameter	**Unit**	**Zurich**	**Madrid**
Space heating load	GJ/a	50.7	22.0
DHW demand	GJ/a	10.9	10.7
Design ambient temperature	°C	-10	-7
Heat load at design ambient temperature	kW	6.5	5.9
Supply/return temperature for space heating at design ambient temperature	°C	35/30	35/30
Radiator exponent	-	1.3	1.3
Global horizontal irradiation	GJ/m^2a	3.913	5.983
Average ambient temperature	°C	9.07	13.91

The heat pump model simulates the working fluid cycle based on a performance map of the compressor, the UA-values of the heat exchangers, and the properties of the working fluid (R410A). The nominal power of the simulated heat pump at A2W35 is 16 kW, and the mass flow rate of the heated water in the condenser was assumed to be 3000 kg/h. Defrosting losses are taken into account by introducing a reduction of the efficiency that is calculated when the temperature in the evaporator drops below -3 °C. The simulated steady state performance of the heat pump operated with the air source evaporator and alternatively operated with the brine source evaporator is shown in Fig. 3.

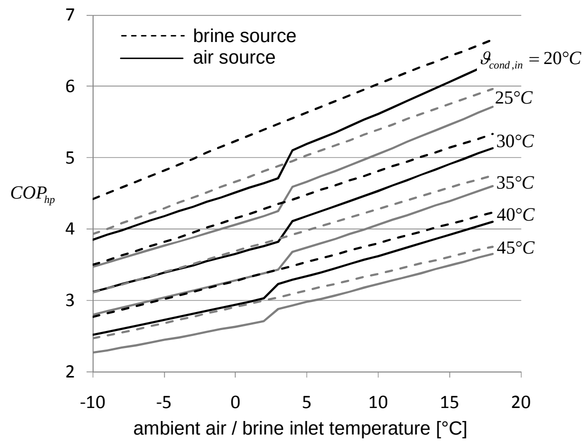

Fig. 3: Coefficient of performance (COP) of the dual source heat pump at different inlet temperatures of the evaporator and the condenser of the heat pump.

The curves show that the performance of the heat pump is in general lower when using air as a heat source for three reasons:

- the air fan is using electricity in addition to the compressor
- the heat transfer from air to the working fluid is worse than from brine to the working fluid
- defrosting losses that have to be taken into account (sharp drop of the performance below 5 °C).

Cycling losses are taken into account by a time constant for reaching 62% of the full heating power after starting the compressor that drives the refrigerant cycle.

The solar collectors were simulated with a slightly modified version of the collector model of Perers & Bales (2002). This collector model is based on the following equation:

$$\begin{aligned}\dot{Q}_{out}/A_{coll} &= F'(\tau\alpha)\cdot K_b \cdot G_b + F'(\tau\alpha)\cdot K_d \cdot G_d - c_{w,F'} \cdot u_w \cdot (G_b + G_d) \\ &- c_{IR} \cdot (L - \sigma \cdot T_{amb}{}^4) - (a_1 + c_{w,hl} \cdot u_w) \cdot \Delta T_{amb} - a_2 \cdot |\Delta T_{amb}| \cdot \Delta T_{amb} - C_{eff} \cdot d\vartheta_m/dt\end{aligned} \quad \text{(eq. 3)}$$

With $\vartheta_m = (\vartheta_{coll,out} + \vartheta_{coll,in})/2$ and $\Delta T_{amb} = \vartheta_m - \vartheta_{amb}$. The performance parameters of the solar collectors correspond to the standard covered selective flat plate collector defined in the IEA-SHC Task 32 and to an uncovered selective flat plate collector tested at the SPF in Rapperswil (Tab. 2). The collector slope was 45° and the orientation towards south for all simulations. Both collectors are equipped with selective coating and therefore $c_{IR} = 0$ was assumed. The incident angle modifier K_b was calculated with $K_b = 1 - b_0 \cdot (1/\cos(\gamma) - 1)$. A collector area of 16 m^2 was used unless other values are explicitly stated.

The storage tank was simulated using the model described by Drück (2006). The storage volume was 1 m^3.

Tab. 2: Flat plate collector parameters with aperture area as the reference area..

Parameter	Unit	covered	Uncovered
$F'(\tau\alpha)$	-	0.8	0.954
a_1	W/m2K	3.5	9
a_2	W/m2K2	0.015	0
$c_{w,F'}$	s/m	0	0.01
$c_{w,hl}$	J/m3K	0	3.77
b_0	-	0.18	0.018
K_d	-	0.87	0.99

3. Results

3.1 General findings

The two modes of collector operation (direct and indirect) can be compared if the amount and temperature of heat delivered to the storage is identical for both cases. This requires that enough energy is demanded from the system in order to use the heat within a short time, such that the storage temperatures of both operation modes are identical too. With the assumption that the electric work for running the collector loop pump is identical for direct and for indirect collector heat use operation and can therefore be neglected in the analysis ($PF_{sys,ind} = COP_{hp,ind}$), it follows from eq. 2 that:

$$\frac{Q_{hp,ind}}{W_{el,ind}} > \frac{Q_{coll,dir} + Q_{hp,dir}}{W_{el,dir}} \qquad \text{(eq. 4)}$$

With $W_{el,dir} = Q_{hp,dir} / COP_{hp,dir}$ and assuming that the total heat delivered is equal in both cases, i.e. $Q_{coll,dir} + Q_{hp,dir} = Q_{hp,ind}$, it follows that:

$$\frac{COP_{hp,ind}}{COP_{hp,dir}} > \frac{Q_{hp,ind}}{Q_{hp,ind} - Q_{coll,dir}} \qquad \text{(eq. 5)}$$

With the definitions of $\Delta COP_{hp} = COP_{hp,ind} - COP_{hp,dir}$ and $\Delta\eta_{coll} = \eta_{coll,ind} - \eta_{coll,dir}$, and with some substitutions and rearrangements, the general criterion that has to be met in order to get an advantage in terms of increased system performance from indirect heat use of the collector instead of direct heat use is found to be:

$$\frac{\Delta COP_{hp}}{\left(COP_{hp,dir} - 1\right)} \cdot \frac{\Delta\eta_{coll}}{\eta_{coll,dir}} > 1 \qquad \text{(eq. 6)}$$

The implication of this general condition can be demonstrated assuming e.g. a COP of the heat pump using a different source than the solar collector ($COP_{hp,dir}$) of 2.5. In this case, an advantage from using collector heat for the evaporator of the heat pump is possible if the COP of the heat pump increases by 1, and at the same time the collector efficiency increases by +150% relative to the direct collector operation mode where the heat pump uses a different heat source to deliver heat to the storage (Fig. 4). The curves of Fig. 4 show for different values of $COP_{hp,dir}$ the minimum increase of collector efficiency and of the COP of the heat pump in order to be able to increase the systems performance factor by using collector heat indirectly.

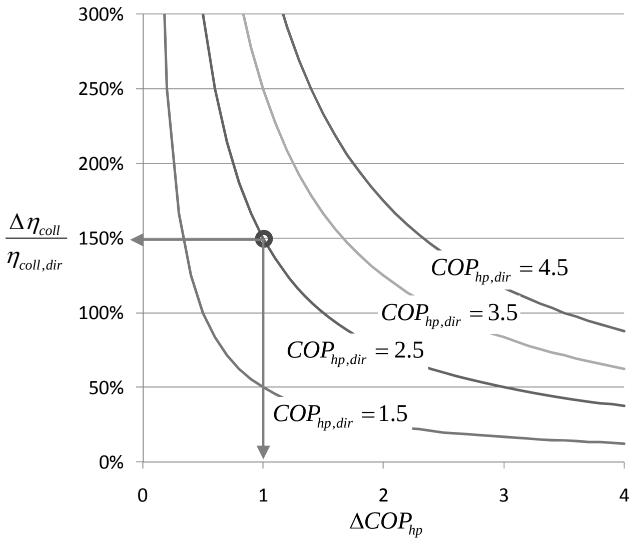

Fig. 4: Curves for switching to indirect collector heat use for different values of $COP_{hp,dir}$ (Haller & Frank 2011).

A direct consequence of eq. 6 is that there is a limit for the irradiation on the collector field (G_{lim}) below which the indirect use of collector heat has a positive effect on the system's performance factor. This limit depends on the characteristics of the heat pump and of the collector, as well as on the temperatures of the heat sink (heat use) of the different heat sources, and on the heat loss term of the collector:

$$G_{lim} = \frac{\Delta COP_{hp}}{\left(COP_{hp,dir} - 1\right)} \cdot \frac{x_{dir} - x_{ind}}{\eta_0} + \frac{x_{dir}}{\eta_0} \qquad \text{(eq. 7)}$$

Where x are the losses of the collector in [W/m^2] which depend on the temperature difference between the collector and the ambient:

$$x = a_1 \cdot \Delta T_m + a_2 \cdot \Delta T_m \cdot \left|\Delta T_m\right| \qquad \text{(eq. 8)}$$

With $\Delta T_m = \left(\vartheta_{coll,in} + \vartheta_{coll,out}\right) / 2 - \vartheta_{amb}$.

3.2 Irradiation limits for specific boundary conditions

For the heat pump and covered collector parameters presented in Section 2.2, Fig. 5 shows the irradiation limit (G_{lim}) below which an indirect use of the collector heat leads to an increase in the system's performance factor, depending on the ambient temperature (ϑ_{amb}) and the heat load temperature ($\vartheta_{u,\,flow}$). These curves are based on eq. 7, combined with steady state simulations of the heat pump and collector performance for the different boundary conditions. It can be concluded that the value of G_{lim} is increasing, with increasing temperature levels of the heat load and with decreasing temperatures of the ambient air.

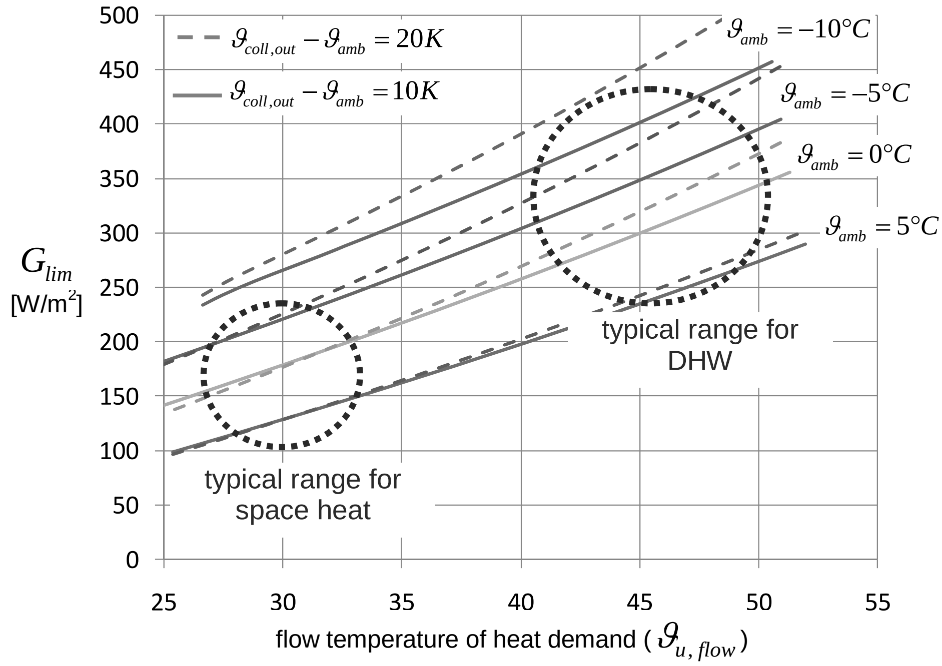

Fig. 5: Irradiation limit (G_{lim}) below which an indirect collector heat use leads to an increase in the system's performance factor of the investigated system, depending on the temperature level (flow temperature) of the heat load, the ambient temperature (which is also the alternative heat source temperature of the heat pump) and the temperature difference between the collector and the ambient for the indirect heat use case (Haller & Frank 2011).

3.3 Potential for improving the system's energetic performance for different climates

Based on transient yearly system simulations for the climates of Zurich and of Madrid with the Task 32 reference conditions, the maximum amount of heat (potential) that can be delivered from the solar collectors to the heat pump with a positive effect on the system's seasonal performance factor has been determined. These potentials have been divided into a potential of using heat from the solar collector for the heat pump at times where the solar collector cannot be operated in direct mode because it would not reach the temperatures required for this operation mode (additional running times), and the potential for switching from a less efficientdirect collector heat use to a more efficient indirect collector heat use (switching).

Fig. 6 shows the potential for additional running times and switching from direct to indirect heat use for the location of Zurich for a system with covered flat plate collectors and for a system with uncovered flat plate collectors, and for space heating systems with different flow and return temperatures. It can be seen that the benefit of using solar collector heat for the evaporator of the heat pump is substantially larger for uncovered collectors than it is for covered collectors, and it increases with increasing temperatures of the space heating system.

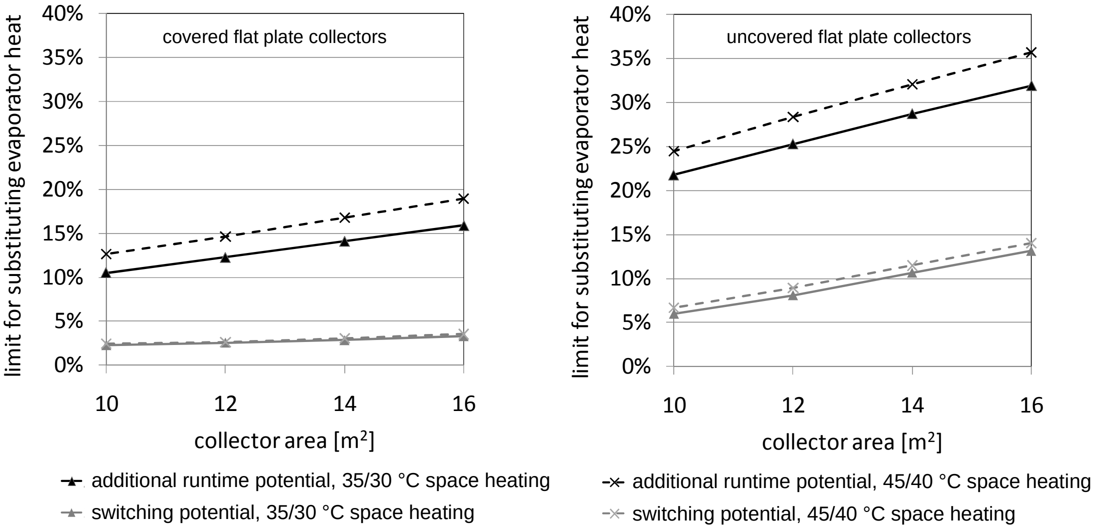

Fig. 6: Theoretical limit of using heat from the solar collectors for the evaporator of the heat pump with benefit for the system's performance factor, given in % of the heat needed for the evaporator of the heat pump, for the climate of Zurich.

Fig. 7 shows the range of values for the potential of advantageous indirect collector heat use for different climatic conditions and collector field areas of 10-16 m^2. For the climate of Madrid, a larger fraction of the heat needed for the evaporator of the heat pump can be covered by heat from the solar collector than for the climate of Zürich (Fig. 7a+b). However, due to the significantly lower space heat demand for the climate of Madrid, in absolute values less heat can be delivered from the collector to the heat pump than for the climate of Zurich (Fig. 7c+d).

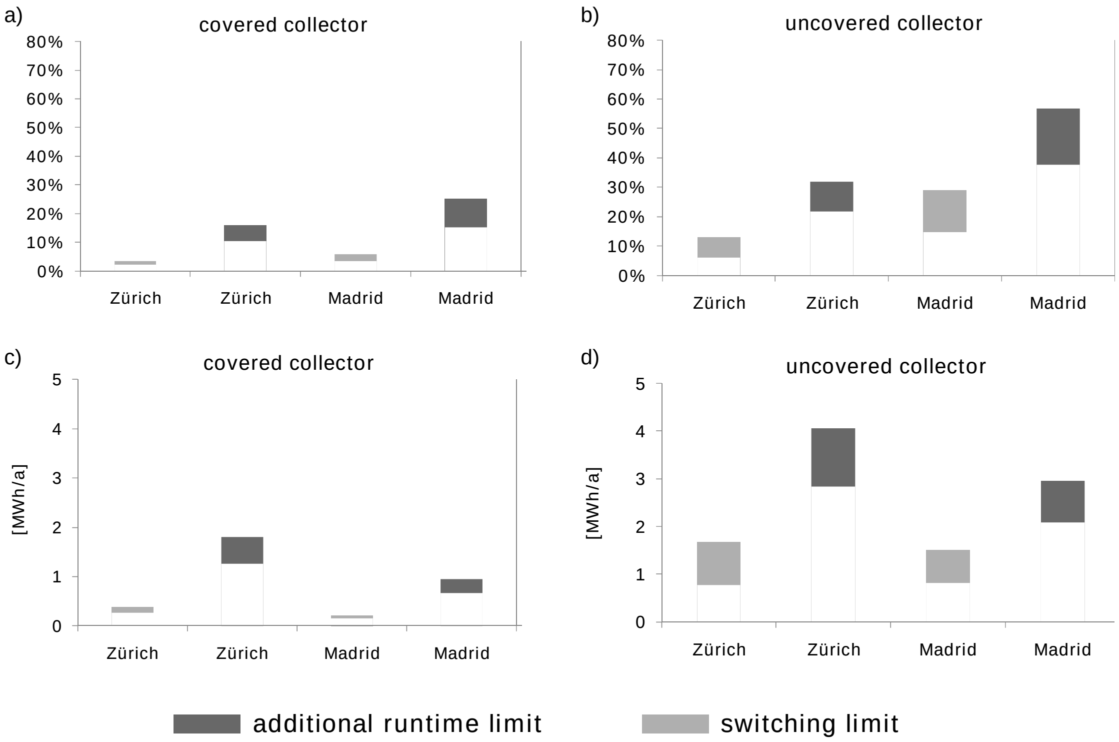

Fig. 7: Range of the annual potential of using heat from the collectors for the evaporator of the heat pump with benefit for the system's performance factor, given in % of the heat needed for the evaporator of the heat pump (a+b) and in absolute MWh of heat delivered (c+d), ranges simulated for system sizes of 1.0 m3 of water storage tank and 10 – 16 m2 of collector area for the climates of Zürich and of Madrid.

4. Discussion

The option to switch from using solar heat directly to using it for the evaporator of a heat pump has been analyzed for systems where the heat pump has the possibility to use a heat source different from the solar collectors (e.g. air). A general mathematical relationship to determine under which circumstances an improvement of the systems performance factor can be expected by using solar heat indirectly instead of using it directly is given with eq. 6. The relationship shows that the conditions for switching are not as easy to meet as one would think of as both the COP of the heat pump and the collector efficiency must increase in order to get a benefit for the system's performance. From this relationship it is concluded that there is a limit for the solar irradiation on the collector field above which using solar collector heat for the evaporator of the heat pump is reducing the system's performance factor rather than increasing it. This irradiation limit depends on the characteristic performance curves of the solar thermal collector and of the heat pump. In particular, the irradiation limit is always higher for collectors with large heat losses to the ambient (e.g. uncovered collectors) than for collectors with low heat losses to the ambient (e.g. covered flat plate collectors). Based on example calculations it has also been shown that the irradiation limit increases with increasing temperatures on the heat demand side, and with decreasing temperatures for the ambient air that was at the same time the alternative heat source for the heat pump in these studies.

5. Conclusion

From the analysis of the option to use solar heat for the evaporator of a dual source heat pump instead of using it directly it can be concluded that this option has a higher potential for improving the overall system's energetic performance for systems with high temperatures on the demand side and low temperatures on the side of the heat source and the ambient air. It has also been shown that the potential of increasing the system's performance by indirect collector operation is mainly based on additional running times of the collector. At times where the solar collectors can be used in direct operation mode, this is most often the preferred mode of collector operation. The effect on systems with the possibility of heat storage on the cold side of the heat pump has not been analyzed in this work.

6. Acknowledgements

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7. Symbols

symbols

A_{coll}	collector area, m^2
a_1	linear heat loss coefficient, W/m^2K
a_2	quadratic heat loss coefficient, W/m^2K^2
b_0	factor for IAM calculation, -
C_{eff}	effective specific heat capacitance of the filled collector, J/m^2K
COP_{hp}	coefficient of performance of a heat pump in terms of heat output divided by electricity input, -
$c_{w,F'}$	influence of wind speed on $F'(\tau\alpha)$, s/m
c_{IR}	long wave radiation exchange factor, -
$c_{w,hl}$	wind speed factor, J/m^3K
$F'(\tau\alpha)$	effective transmission-absorption product multiplied with the collector efficiency factor, -
G	solar irradiation, W/m^2
K	incident angle modifier, -
L	long wave irradiation, W/m^2
Q	Heat or solar energy, J
$\dot{Q}$	Heat transfer, W
PF	performance factor in terms of heat output divided by electricity input, -
T	absolute temperature, K
t	time, s
u_w	wind speed, m/s
W	work, J
x	specific heat loss, W/m^2
γ	incident angle, °
η	collector efficiency, -
ϑ	temperature, °C
ϑ_m	mean collector temperature, °C
σ	Stefan-Boltzman constant, W/m^2K^4

subscripts

amb	ambient
b	beam
coll	solar thermal collector
cond	condenser
d	diffuse
day	restricted by daily limits
dir	collector heat used directly
el	electricity
evap	evaporator of the heat pump
flow	flow line
hp	heat pump
in	inlet
ind	collector heat used indirectly, i.e. for the evaporator of the heat pump
irrad	irradiation on the collector plane
limit	limit
OFF	during times without collector operation
ON	during times with collector operation
out	outlet
pot	potentially useable for the evaporator of the heat pump
sys	system
u	useful energy

8. References

EHPA - European Heat Pump Association, 2010. OUTLOOK 2010 - European Heat Pump Statistics / Summary. Brussels, Belgium.

Citherlet, S., Bony, J. & Nguyen, B., 2008. SOL-PAC - Analyse des performances du couplage d’une pompa à chaleur avec une instnallation solaire thermique pour la rénovation - Rapport final. Swiss Federal Office of Energy (SFOE).

Drück, H., 2006. Multiport Store Model for TRNSYS - Type 340 - V1.99F.

Frank, E., Haller, M., Herkel, S. & Ruschenburg, J., 2010. Systematic Classification of Combined Solar Thermal and Heat Pump Systems. In: Proc. of the EuroSun 2010 Conference, Graz, Austria.

Hadorn, J.C., 2010. Solar and Heat Pump Systems - A New IEA SHC Task 44 & HPP Annex and Analysis of Several Combinations for Low Energy Houses. In: Proc. of the EuroSun 2010 Conference, Graz, Austria.

Haller, M.Y. & Frank, E., 2010. Kombination von Wärmepumpen mit solarthermischen Kollektoren - Konzepte und Fragestellungen. In: CD - Konferenzbeiträge 16. Status-Seminar “Forschung und Bauen im Kontext von Energie und Umwelt”, 2./3. Sep., BRENET - Nationales Kompetenznetzwerk Gebäudetechnik und erneuerbare Energien, ETH Zürich, 1-6.

Haller, M.Y. & Frank, E., 2011. Steigert die Nutzung von Solarkollektoren als Wärmequelle für Wärmepumpen die System-Arbeitszahl? In: 21. Symposium Thermische Solarenergie, 11. - 13. Mai, OTTI Regensburg, Bad Staffelstein, Germany, CD.

Haller, M.Y., Frank, E., Trinkl, C. & Zörner, W., 2010. Systematische Gliederung der Systemkombination von solarthermischen Anlagen mit Wärmepumpen. In: 20. OTTI Symposium Thermische Solarenergie, Bad Staffelstein, Germany.

Heimrath, R. & Haller, M., 2007. Project Report A2 of Subtask A: The Reference Heating System, the Template Solar System - A Report of IEA SHC Task 32: Advanced Storage Concepts for Solar and Low Energy Buildings.

Henning, H.-M. & Miara, M., 2008. Systems using solar thermal energy in combination with heat pumps - 1st concept paper. Fraunhofer ISE.

Karagiorgas, M., Galatis, K., Tsagouri, M., Tsoutsos, T. & Botzios-Valaskakis, A., 2010. Solar assisted heat pump on air collectors: A simulation tool. Solar Energy, 84(1), p.66-78.

Klein, S.A., Beckman, W.A. & Mitchell, J.W., 2004. TRNSYS 16, A Transient System Simulation Program. Solar Energy Laboratory, University of Wisconsin, Madison USA.

Perers, B. & Bales, C., 2002. A Solar Collector Model for TRNSYS Simulation and System Testing - A Technical Report of Subtask B of the IEA-SHC - Task 26.

Trinkl, C., Zörner, W. & Hanby, V., 2004. A Review on Solar-Assisted Heat Pump Systems for Domestic Heating. In: Proc. of the EuroSun 2004 Conference, Freiburg, 1, 734 - 743.

Trojek, S. & Augsten, E., 2009. Solartechnik und Wärmepumpe - Sie finden zusammen. Sonne Wind & Wärme, 06/2009(06), p.62-71.

Weiss, W. & Mauthner, F., 2011. Solar Heat Worldwide - Markets and Contribution to the Energy Supply 2009 - Edition 2011. International Energy Agency Solar Heating and Cooling Programme.