THERMAL ANALYSIS OF ABSORPTION HEAT PUMP IMPLEMENTATION IN AN INDUSTRIAL DRYER

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ABSTRACT

Industrial dryers are very energy-intensive, contributing to a large share of the thermal energy demand in industry. Most of this energy is discharged as moist air to the environment. The calorific content of the exhaust air is high due to the large amount of water vapour in this waste stream. Heat pump systems can be used to recover heat from the exhaust air, hereby raising the energy efficiency of the dryer. Simulations have been performed using data from an existing drying installation with conventional heating. The increase in energy efficiency is analyzed for the implementation of several types of absorption heat pumps in the drying cycles. The simulation results show the influence of different working fluid pairs and different configurations: type I, type II and double lift cycle. The highest amount of energy savings is achieved with type I absorption heat pumps using water–lithium bromide as the working fluid pair. With optimized temperature levels in the different components, the thermal energy use of the complete dryer can be reduced with 20%. The performance of absorption heat pumps in drying systems is however still bounded by the temperature limit of the water-lithium bromide working fluid pair. Searching for alternative working fluid pairs for higher temperature applications is therefore still essential and can increase energy savings even more.

INTRODUCTION

Drying is a thermal process with the aim to remove water from a product to reduce its moisture content. Many industries require such a drying process in their production method. Among these industries are chemical, agricultural, food, polymer, ceramics, pharmaceutical, paper and wood processing industries. Drying is a very energy-intensive process, utilizing up to 25% of the national energy use in developed countries [1].

Most energy in drying processes is lost through heat losses and through the moist outlet air of the dryer. The moist outlet air contains a large amount of water vapour, which has a high latent heat content. By (partially) condensing this water vapour, a lot of energy can be recovered and reused in the drying process. One way to achieve this is by implementing heat pumps in the drying process [2]. Heat can be recovered by condensing water vapour from the dryer outlet air. This heat can be brought to a higher temperature by the heat pump, to be used as heat source for the inlet air of the dryer. Various studies have been performed which demonstrate the advantages of the implementation of heat pumps in drying systems or heat pump dryers (HPD). One of the main advantages is energy use reduction. Minea reports 42%-48% electrical energy use reduction for wood drying with electrically driven heat pumps compared to electrical resistance heating, Gabas et al. report 38%-47% for apple drying [3] [4].

Most of the research is focused on electrically driven heat pumps in the drying system. When electricity is not favourable as energy source, thermally driven heat pumps are an alternative. One of the technologies that is currently available is the absorption heat pump (AHP) [5]. Le Lostec et al. investigated absorption heat pumps for wood chip drying [6]. He concluded that the absorption heat pump performs better than a traditional furnace, but that for this case the cost would be high compared to a traditional system.

In this paper, several simulations have been performed to predict the energy (cost) saving potential of the implementation of an absorption heat pump in a dryer system. Data from a reference dryer is used to obtain the boundary conditions for the operation of the absorption heat pump. These simulations are performed within the frame of the transnational CORNET HP4Drying project, which has the aim to further introduce heat pump drying and its advantages to industry.

NOMENCLATURE

COP	[-]	Coefficient of performance
Q	[W]	Heat flow
T	[°C]	Temperature

Subscripts

abs	Absorber
aHP	Air after absorption heat pump heating
AHP	Absorption heat pump
cond	Condenser
dryer	Original dryer
evap	Evaporator
gen	Generator
in	Absorption heat pump heat input
I	Type I absorption heat pump
II	Type II absorption heat pump
out	Absorption heat pump heat output
2L	Double lift absorption heat pump

INDUSTRIAL REFERENCE CASE

The dryer considered in this paper is a milk dryer. Liquid milk enters at a temperature of 75°C and containing 40% of dry substance. Using convective drying, the water in the milk is evaporated and the percentage of dry substance increases. The end product is milk powder with a moisture content of 2%. A mass flow of 5 ton/h milk enters the dryer, which results in about 3 ton/h of evaporated water.

The milk is dried by a convective process. A schematic overview of the dryer is shown in Figure 1. The dryer consists of two stages. In the first stage, a flow of 42311 m³/hr of ambient air with relative humidity of 60% is heated from 20°C to 210°C through thermal oil which is heated by a gas burner. This heated air passes to the drying chamber where water evaporates, so increasing the moisture content of the air and decreasing its temperature. In the second stage, a fresh air flow of 19568 m³/h is heated from 30°C to 50°C and added to the drying chamber, also removing moisture from the milk product. The moist air flows mix and exit the dryer at a temperature of 60°C. The moisture content can be calculated from the mass balance of the dryer when air leakages are neglected.

The energy use of the dryer is equal to the energy needed to heat up the first air flow from 20°C to 210°C and the second from 30°C to 50°C and is equal to 2,861 MW. The energy use per mass of evaporated water is equal to 967 kWh_{th}/ton. This value is in the range that is expected for a dryer which uses no heat recuperation.

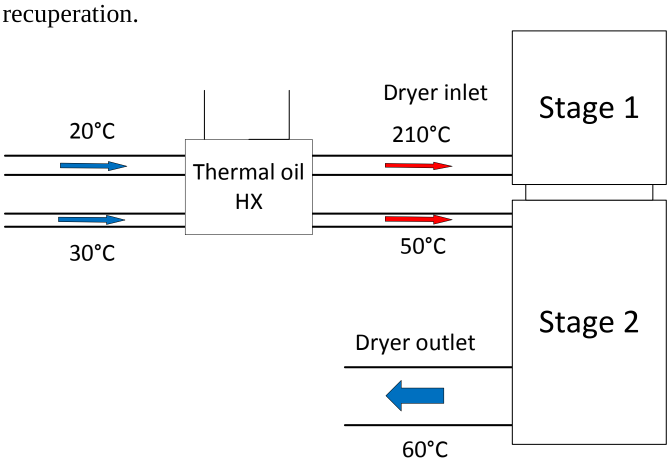


Figure 1 Schematic representation of the milk dryer

ABSORPTION HEAT PUMP MODEL

This paper focuses on the analysis and the energy savings that are related to the implementation of absorption heat pumps in the described industrial dryer. The total heat demand of the dryer cannot be fully delivered by the absorption heat pump, so additional heating will still be necessary in each case to achieve the needed temperatures for the dryer.

The absorption heat pump cycles are simulated using Matlab. Two types of working pairs are investigated: lithium bromide and water (LiBr-H_2O) and water and ammonia (H_2O-NH_3). For LiBr-H_2O, the solution and refrigerant vapour properties were gathered from the CoolProp database [7]. For H_2O-NH_3, the properties were determined using the functions described by Pátek and Klomfar [8].

An absorption heat pump cycle consists of at least four main components: evaporator, condenser, absorber and generator. A solvent heat exchanger is usually also included to improve the system performance. The several components can be linked by pumps or valves. The evaporator, condenser and solvent heat exchanger are modelled using the heat balance and pinch point equations. The pressures in the evaporator and condenser are equal to the pressures in the absorber and generator respectively. The pressures are equal to the equilibrium pressure at the temperature and the concentration of the solvent in the respective component. The heat transfer in the absorber and generator is also modelled through the heat balance and pinch point equations.

Several assumptions and approximations are made to develop the absorption heat pump models:

- The refrigerant exits the evaporator as saturated vapour and it exits the condenser as saturated liquid.
- The solvent and the refrigerant are in equilibrium at the absorber and generator.
- Pressure losses in piping and heat exchangers is assumed to be negligible for the simulation results.
- Expansion valves are adiabatic.
- Heat losses to the environment are neglected.
- Only steady state is simulated, transient effects are not taken into account.

All the parameters used in the simulations are summarized in Table 1.

Table 1 Parameters used in simulation

Parameter	Unit	Value
Stage 1		
Ambient air temperature	°C	20
Ambient air relative humidity	%	60
Air volume flow (at 20°C)	m³/h	42311
Milk inlet mass flow	ton/h	5
Milk inlet moisture content	%	60
Stage 2		
Ambient air temperature	°C	30
Ambient air relative humidity (at 20°C)	%	60
Air volume flow (at 30°C)	m³/h	19568
Air outlet temperature	°C	60
Milk outlet moisture content	%	2
Absorption heat pump		
Pinch point of the absorber, generator	K	8
Pinch point of the evaporator, condenser	K	8
Pinch point of the solvent heat exchanger	K	8

The analysis of the different absorption heat pump cycles and configurations is made in function of the temperature to which the air is heated by the absorption and heat pump. At this point, the cycle is optimized to achieve the lowest total energy use. This calculation neglects the pumping power needed for the absorption heat pump, however the simulations have shown that this power is negligible compared to the heat flows.

ABSORPTION HEAT PUMP: TYPE I

An absorption heat pump of type I is similar to a basic compression heat pump (Figure 2). The condenser, expansion valve and evaporator are also present, but the compressor is replaced by an absorption cycle. This absorption cycle consists of an absorber, a pump, a generator, a valve and usually also an internal heat exchanger. The refrigerant vapour exiting the evaporator is absorbed in a liquid absorbent in the absorber, which releases heat. The strong refrigerant solution is then pumped to a higher pressure. Next it flows to the generator, where heat has to be added to boil out the refrigerant which then goes to the condenser. The weak refrigerant solution from the generator returns over a valve to the absorber. The advantage of the absorption cycle is that it requires little work input for the pump compared to the work input of a compressor. However, it does require a high temperature heat input in the generator. When implemented in a drying system, the inlet air is heated by both the absorber and the condenser. The exhaust air is used to transfer heat to the evaporator. The high temperature heat required in the generator is delivered by an external heat source. The coefficient of performance (COP) is defined in Eq. (1) as the ratio of the heat delivered by the absorber (Q_{abs}) and condenser (Q_{cond}) to the heat delivered to the generator (Q_{gen}), the work input of the pump is neglected. This COP has to be higher than 1 to have an AHP that has a lower energy consumption than a conventional burner. The relative energy savings are defined by Eq. (2). Q_{dryer} is the heat needed in the original dryer.

$$COP_I = \frac{Q_{abs}+Q_{cond}}{Q_{gen}} \tag{1}$$

$$Relative\ energy\ savings = 100\% \frac{Q_{abs}+Q_{cond}-Q_{gen}}{Q_{dryer}} \tag{2}$$

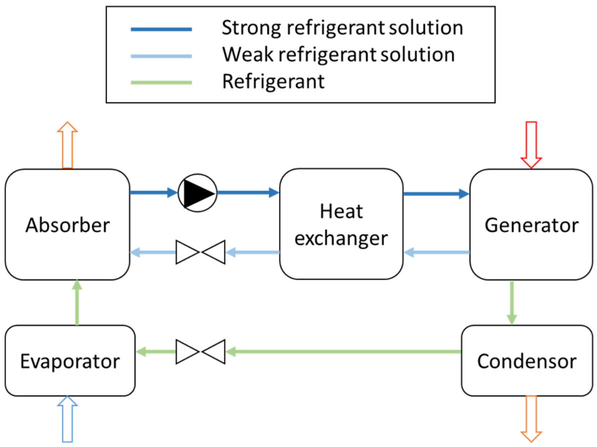
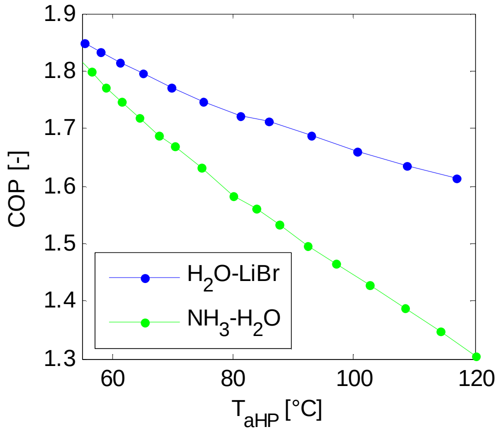


Figure 2 Type I absorption heat pump - hollow arrows indicate heat flows

There are two pairs of working fluids for absorption heat pumps that are currently being used in industry: H_2O-LiBr (H_2O acts as refrigerant) and NH_3-H_2O (NH_3 acts as refrigerant). There are advantages and disadvantages for both pairs [9]. A simulation is made to compare both pairs by the energetic and economic profits of the absorption heat pump.

Figure 3 shows the COP and the relative energy savings of the absorption heat pump in function of the air temperature of the first air flow after heating by the AHP. The second air flow is only heated to a maximum of 50°C. Two counteracting effects are observed. When increasing the air temperature, less additional heating is needed after the absorption heat pump, so less primary energy is needed. However by increasing the air temperature, the COP of the absorption cycle lowers which results in a higher primary energy consumption. This results in an optimum temperature for energy savings. For NH_3-H_2O, this optimum is reached at a heated air temperature of around 110°C. For the H_2O-LiBr working pair this optimum cannot be perceived, because the maximum operating temperature is limited. To increase the temperature in the absorber, the LiBr concentration has to be increased. This LiBr concentration is bounded by the solubility of LiBr in water, which limits the maximum achievable temperature.



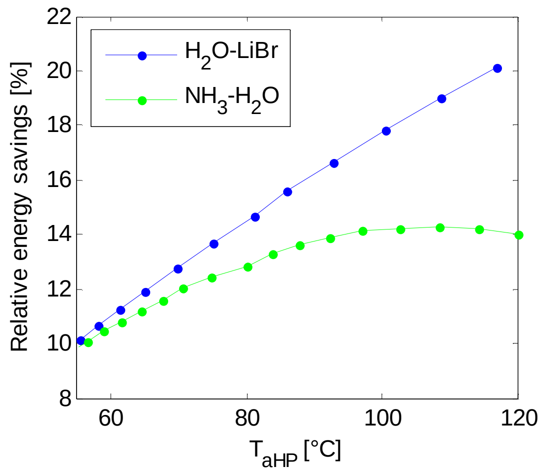


Figure 3 COP (top) and relative energy savings (bottom) of AHP for different working fluid pairs

It is clear that the working fluid pair NH_3-H_2O does not perform better than H_2O-LiBr at the temperature range and temperature lifts that are used for this industrial case. A NH_3-H_2O absorption heat pump could perform better at lower temperatures. This is the effect of the variation of the fluid properties, in particular the heat of vaporization, with temperature and solvent concentration. Also at higher temperatures, the NH_3-H_2O absorption heat pump requires high operating pressures, which makes it more expensive. The temperatures needed for the drying process are in the higher range where H_2O-LiBr performs better, so for the next simulations, only this working pair will be discussed.

An absorption heat pump type I supplies heat at both the condenser and the absorber. It can be chosen which one operates at a higher temperature and as such in which order their respective heat is added to the air. Figure 4 shows the COP (top) and the relative energy savings (bottom) for both configurations. The system with the absorber at the highest temperature has a lower COP, and slightly lower energy savings. However, the most important difference is that due to the solubility of LiBr, the highest temperature that can be reached is much lower for this system (around 78°C) than for the system with the condenser at a higher temperature (around 117°C). As a result, this causes the latter configuration to be the more promising solution.

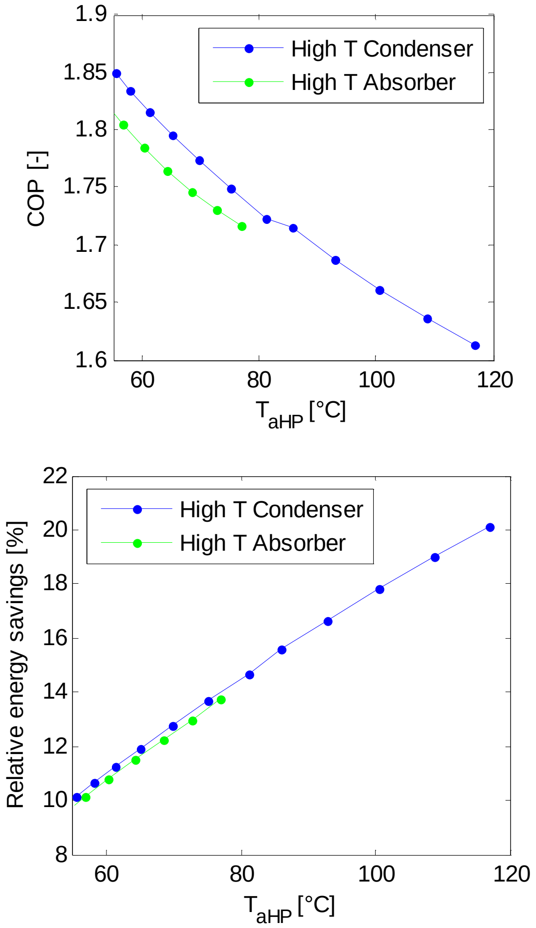


Figure 4 COP (top) and relative energy savings (bottom) of type I AHP for different configurations

ABSORPTION HEAT PUMP: TYPE II

An absorption heat pump of type II or heat transformer consists of the same components as absorption heat pumps of type I, but the pressure levels in these components are different (Figure 5). Because of this, the heat transformer does not require heat input at high temperature, but at medium temperature. In a drying system, this medium temperature heat is transferred from the exhaust air to the generator and to the evaporator. The high temperature heat from the absorber is used to heat the inlet air. The low temperature heat generated in the condenser has to be released to the environment. The COP is defined in Eq. (3) as the ratio of the heat delivered by the absorber to the sum of the heat delivered to the evaporator and generator, the work input of the pumps is neglected. The COP for this cycle will always be smaller than 1. The relative energy savings are defined by Eq. (4).

$$COP_{II} = \frac{Q_{abs}}{Q_{evap}+Q_{gen}} \quad (3)$$

$$Relative\ energy\ savings = 100\% \frac{Q_{abs}}{Q_{dryer}} \quad (4)$$

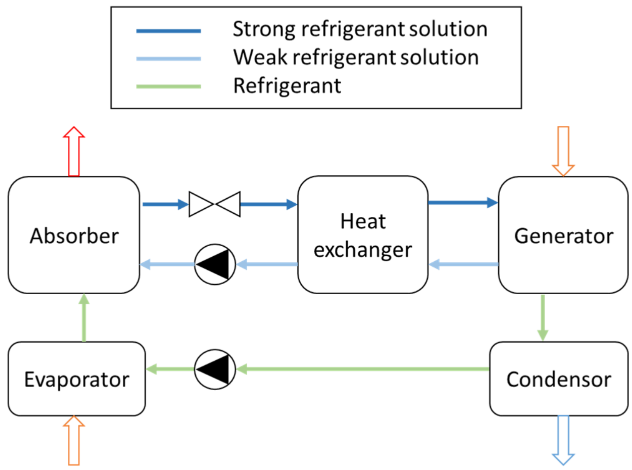


Figure 5 Type II absorption heat pump - hollow arrows indicate heat flows

Similar to the condenser and absorber in a type I absorption heat pump, either the evaporator or the generator can be chosen as the component which works at the highest temperature. For both configurations, the results are shown in Figure 6. The COP of the configuration with the evaporator at a higher temperature is slightly higher. The relative energy savings however will only be related to the temperature because the AHP requires (almost) no extra energy input. Only very low temperatures can be achieved in this case. To go to higher temperatures in the absorber, the temperature in the condenser has to go down. This is limited because the heat still has to be at a higher temperature than the ambient air temperature, to be able to remove this heat from the condenser. It can be concluded that for this case, the absorption type II is not a good solution.

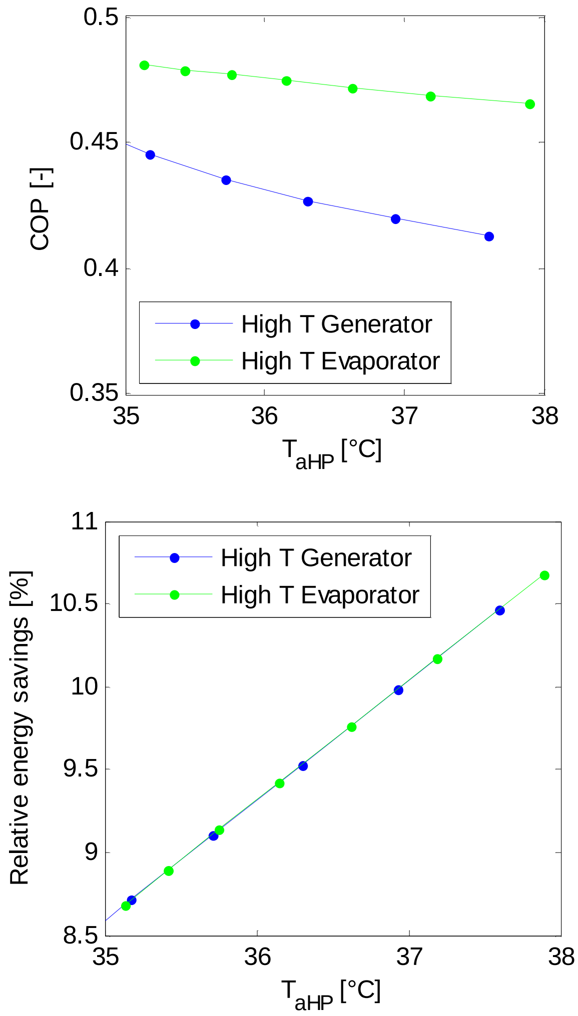


Figure 6 COP (top) and relative energy savings (bottom) of type II AHP for different configurations

DOUBLE LIFT CYCLE

The highest air temperature that can be reached with the absorption heat pumps that have been considered is around 120°C, which is not near the 210°C needed for the dryer installation. A double lift cycle can be used to reach higher temperatures, but at lower COP's [10]. The double lift cycle consists of an extra absorber and evaporator, where the high temperature evaporator is heated by the heat from the low temperature absorber. The other components act in similar ways as for the single lift type I absorption heat pump. The COP of a double lift cycle is defined in Eq. (5) and the relative energy savings are defined in Eq. (6). The cycle configuration is shown on Figure 7. As for the type I absorption heat pump, it can be chosen to operate either the condenser or the (second) absorber at the highest temperature.

$$COP_{2L} = \frac{Q_{abs,2}+Q_{cond}}{Q_{gen}} \quad (5)$$

$$Relative\ energy\ savings = 100\% \frac{Q_{abs,2}+Q_{cond}-Q_{gen}}{Q_{dryer}} \quad (6)$$

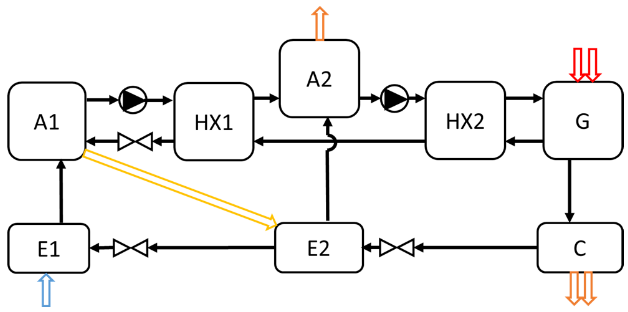


Figure 7 Double lift type I AHP

A = absorber; G = generator; E = evaporator; C = condenser; HX = heat exchanger

The results of the simulations are summarized in Figure 8. The performance of both configurations is nearly equal. However, higher temperatures can be reached when the condenser operates at a higher temperature than the absorber. The highest temperature that can be reached is around 150°C. However, the COP has dropped significantly when compared to the type I AHP, which results in lower energy savings.

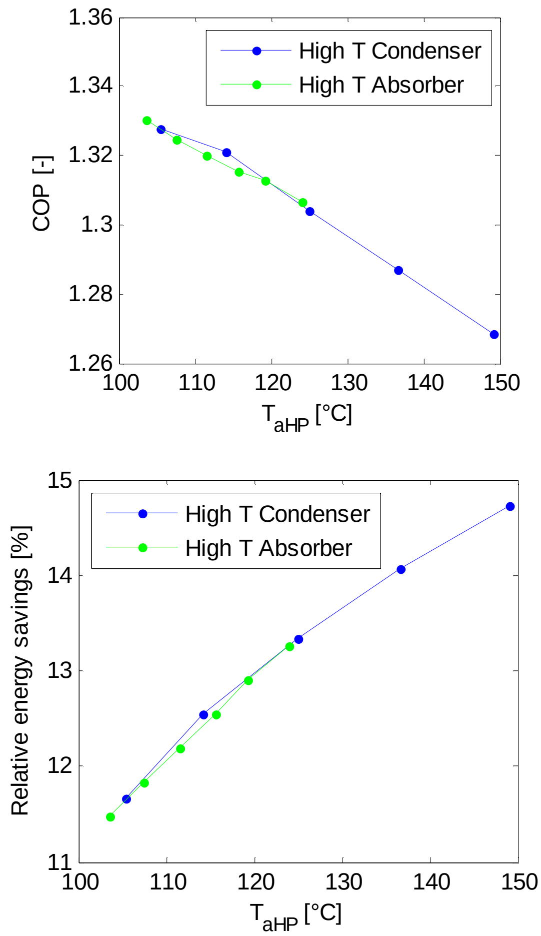


Figure 8 COP (top) and relative energy savings (bottom) of double lift AHP for different configurations

Operating LiBr-H_2O absorption heat pumps at higher temperatures has another drawback. The LiBr-solution becomes very corrosive at higher temperatures. Air temperatures of 100°C and more are therefore hard to reach in practice.

CONCLUSION

Simulations were performed for the implementation of absorption heat pumps in an industrial dryer. Working fluid pairs H_2O-LiBr and NH_3-H_2O were analysed and compared. Also type I, type II and double lift absorption heat pump cycles were analysed and compared. All simulations showed reduction in energy use. For this case, the absorption heat pump type I with H_2O-LiBr as working fluid pair performed best, with maximum energy use reduction of 20% compared to the original dryer.

In practice however, the performance can be limited by the temperature limitation of the H_2O-LiBr working fluid pair. At higher temperature, the solution becomes highly corrosive. Because of this practical limitation, the heat delivered by absorption heat pumps available on the market is limited to about 100°C. This limits the achievable energy use reductions for this and similar cases.

Several opportunities are possible for further energy use reduction and research for absorption heat pump implementation in dryers. Firstly reducing the temperature needed in the drying process could be advantageous for energy use and AHP implementation. More research for corrosive resistant components for H_2O-LiBr solutions at high temperatures could increase the temperatures reached by these types of AHP. Also investigations for other working fluid pairs for AHP can improve AHP performance for other temperature ranges and temperature lifts.

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