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High Temperature Heat Pump Integration using Zeotropic Working Fluids for Spray Drying Facilities

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Abstract

This paper presents an analysis of high temperature heat pumps in the industrial sector and demonstrates the approach of using zeotropic mixtures to enhance the overall efficiency. Many energy intensive processes in industry, such as drying processes, require heat at a temperature above 100 °C and show a large potential to reuse the excess heat from exhaust gases.
This study analyses a heat pump application with an improved integration by choosing the working fluid as a mixture in such a way, that the temperature glide during evaporation and condensation matches the temperature glide of the heat source and sink best possibly. Therefore, a set of six common working fluids is defined and the possible binary mixtures of these fluids are analyzed. The performance of the fluids is evaluated based on the energetic performance (COP) and the economic potential (NPV). The results show that the utilization of mixtures allows a heat pump application to preheat the drying air to 120 °C with a COP of 3.04 and a NPV of 0.997 Mio. €, which could reduce the natural gas consumption by 36 %.

Selection and/or peer-review under responsibility of the organizers of the 12th IEA Heat Pump Conference 2017.

Keywords: Heat Pump Integration; Zeotropic Mixtures; Spray Dryer; Industrial Sector; Industrial Heat Pumps

1. Introduction

Spray drying processes are energy intensive processes with a significant potential for improvements. The technology is utilized for the production of for example milk and coffee powder. This is relevant for the presented case related to Danish industry, but spray driers are in operation many places in Europe and worldwide. While many research activities concentrate on an improved integration of the waste heat with or without the use of a conventional heat pump, the present study aims to take advantage of using a zeotropic refrigerant mixture as working fluid.

1.1. State of the art

This section discusses the present use of spray drying processes and its relevance to industrial sector. Furthermore, a literature review for the integration of heat pumps in drying processes is undertaken, and the potential advantages of zeotropic refrigerant mixtures compared to state of the art technologies.

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1.1.1. Drying Processes in Danish Industry

The industry sector is a major consumer of energy in the Danish economy, accounting for almost 70,000 TJ of fuels and more than 27,000 TJ of electricity in 2012 [1]. Drying processes within the industry have the highest fuel consumption, and use 19 % of the total fuel input to the industry, equivalent to 12.800 TJ [2].

Drying processes are particularly important in the food industry, where water has to be removed from the products in order to conserve them. In Denmark, 33 % of the fuels used for drying processes are used in the food industry, 26 % in the wood and paper industry and 10 % in the building material industry.

Typical drying processes in the food industry consist amongst others of the types: spray, direct contact, drum, fluidized bed, conveyer and freeze drying [3]. Most of these processes remove water by means of hot air, which is often rejected into the environment without further utilization. Excess heat from drying processes thus presents a large part of the industrial waste heat. Approximately 7 % of the industrial waste heat in Denmark originates from drying processes. The food industry alone has excess heat of 375 TJ from these processes [4].

Table 1: Overview of drying technologies, application areas and temperature ranges for industries in Northern Europe. [2], [3], [5], [6]

Drying Technology	**Industry**	T_{in} **[°C]**	T_{out} **[°C]**	**Share [%]**
Spray Dryer	Milk/ Coffee/ Starch	200 - 270	60 - 110	25
Drum Dryer/ Kiln Dryer/ Rotary Dryer	Sugar[1]	500	120	14
	Fruits/ Vegetables	65 – 105	20 – 40	5
	Fishmeal	500 – 600	80 – 120	9
Direct Contact Dryer (steam)	Bone meal	180	133	16
Fluidized Bed- / Fixed Bed- / Conveyer-/ Tray Dryer	Milk/ Coffee/ Starch	60 – 90	40 – 70	5
	Cereals/ Sugar	40 – 70	20 – 30	23
	Fruits/ Vegetables	40 – 60	20 – 30	3

[1] In Denmark largely covered by superheated steam dryers

An overview of drying technologies in different industries applicable to Northern Europe is given in Table 1, also showing temperature characteristics of the drying medium. Drying temperatures in the food industry are generally low (< 250 °C) to retain temperature sensitive components affecting the nutritional value of the product. These low drying temperatures result in low temperatures of the excess heat, which are typically below 90 °C. To efficiently remove water from the product and have short residence times in the dryer, large mass flow rates of the drying medium are often used. The share of drying energy used for individual applications refers to Danish industry. It indicates the magnitude for the different processes and shows that approximately 25 % of the energy used in drying processes is from spray drying and that 36 % of drying energy demand is below 100 °C.

Drying processes in the food industry in Denmark are mainly covered by fossil fuels (36 % fuel oil and 27 % natural gas) and biomass or biomass-based fuels (18 %). A reduction in the fuel use of these processes would thus not only reduce operating costs but also CO_2 emissions.

1.1.2. Review of possible Energy Efficiency Improvements

Drying processes can be improved with respect to different criteria, such as energy demand, emissions or cost. Atkins et al. [7] summarizes the different efficiency improvement measures based on Kemp [8] in three categories: reduction of heat required for the drying process, reduction of energy consumption by improved efficiency (heat recovery), and lastly replacement of fuel.

Energy efficiency measures such as optimizing the dryer performance by increasing the surface areas or more efficient drying techniques, such as superheated steam or microwave drying, have been implemented in many companies and replace some of the technologies shown in Table 1. However, a complete replacement of the dryer is associated with changes in the production process and high investment costs. Most processes using air as drying medium cannot recover heat from the drying air after the drying process. The air cannot be recirculated, as growth of bacteria could spoil the product and the temperatures of the excess drying air are in a range, which makes direct heat recovery difficult.

The present study focuses on improving the process efficiency and aims to decrease the fuel consumption by utilizing the excess heat from spray dryers. A lot of research has been done to develop efficient solutions to integrate the waste heat below 100 °C. This can be done by direct heat exchange or by using a heat pump to lift the temperature level. For example, Ai et al. [9] propose to use the waste heat from a soy powder spray dryer at

70 °C for covering the space heating demand and preheating the incoming drying air. Wang et al. [10] analyze the potential of preheating the drying air by direct heat exchange with the excess air and additional heating with a transcritical R134a heat pump, which matches the temperature profile of the drying air well.

Atkins et al. [7] point out the limitations of using the waste heat at 65 °C to 70 °C in direct heat exchange for preheating the drying air, which is in many cases already preheated to around 40 °C and therefore propose to include additional heat demands on site in that temperature range.

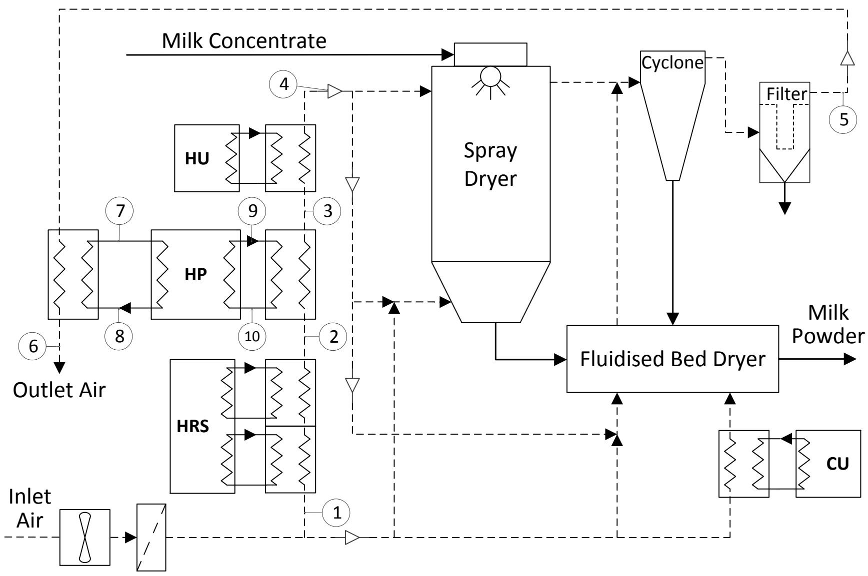
Based on the mature heat integration in the temperature range below 100 °C, and the availability of state of the art technologies in this range, Jensen et al. [11] propose the use of ammonia-water hybrid absorption-compression heat pump technology. The authors present an economically feasible solution for using the waste heat from excess air at 80 °C to preheat the drying air up to 106 °C. This is enabled by a sophisticated integration of the heat pump into the sink and source profile and a match of their temperature profiles. This behavior can be observed as well for heat pumps utilizing zeotropic working fluid mixtures [12] and motivates the present study.

1.2. Scope of work

The present study investigates the use of zeotropic mixtures of six natural working fluids, specifically hydrocarbons, for optimal utilization of excess heat from a spray drying facility. The study is based on a numerical model of the selection of the mixture and the configuration of the heat pump.

1.2.1. Case Study of a Spray Drying Processes in the Dairy Industry

The production of milk powder occurs in several steps, which are similar for different powder types, i.e., whole milk, skim milk and whey powder. The incoming raw milk is first thermally treated for pasteurization and the milk properties are standardized for fat content. In an evaporator section, consisting of multi-stage falling film evaporators, the solid content of the milk is increased from 13 % to 55 %. The concentrate is then passed on to the spray dryer, where the remaining water is removed by hot air. The finished product has a dry matter content of above 95 %. Figure 1 shows the suggested configuration with a heat pump (HP) for heat recovery. Data for the overall flows in the drying section are given on the right hand side. The incoming ambient air (1) is currently pre-heated with an existing Heat Recovery System (HRS) to 70 °C at (2), using condensate from the evaporators, of which some is used in a hybrid water-ammonia heat pump to increase the temperature. The preheated air is then currently heated to its final temperature of 210 °C at (4) using steam generated in a natural gas boiler.



Abbrev.: CU/HU – Cold/Hot Utility; HRS – Heat Recovery System; HP – Heat Pump

	T	$\dot{m}$
	[°C]	[kg/s]
Drying Air		
1	15	43.7
2	70	43.7
3	120-125	43.7
4	210	43.7
Excess Air		
5	70	61.5
6	45	61.5
Secondary Cycle Source		
7	65	14.8
8	40	14.8
Secondary Cycle Sink		
9	75	10.6
10	125-130	10.6

Figure 1: Flow sheet of a drying process for milk powder production with possibility for heat pump integration with key numbers from a reference plant in Denmark

The heat pump installation would use the heat from the excess air (5) to replace part of the heat load from the heating utility (HU). The temperature of the excess air (6) after exploitation by the heat pump is a design parameter, which influences the heat load. The HU load from (2) to (4) is partly moved to the heat pump heating from (2) to (3). The excess air is cooled to 45 °C, utilizing 1.55 MW waste heat, to reach a temperature above 120 °C at (3).

The implementation of a heat pump faces several challenges. In order to avoid contamination and allow freedom when locating the heat pump on site, secondary cycles are introduced on both sides. The secondary cycles are operated with water, which is pressurized on the sink side. For the gas-to-water heat exchangers a minimum temperature difference of 5 K is assumed. Since the investment costs for the secondary cycle heat exchangers varies for each application it is excluded from economic calculations, but separately reported.

2. Methods

The analysis is based on a lumped parameter approach to model the components of the heat pump, including first principles to largest possible extent. The component dimensions are determined based on common correlations for performance, and are applied to determine economic investment costs of the individual units.

2.1. Heat Pump Model

The high temperature glides in heat sink and source result in an increased possible improvement, which can be gained by employing mixtures as working fluids. In order to analyze which working fluid mixture shows the best performance for a specific case, the model is developed to conduct heat pump simulations for several pure and mixed working fluids and provide the required parameters for a sufficient evaluation and comparison of the different solutions. The diverse medium properties of the mixtures influence the thermodynamic cycle significantly, which requires a flexible and adaptable model.

2.1.1. Structure of Numerical Model

Figure 2 shows the flow sheet of the model (left) and the according T-$\dot{Q}$-diagram (right). The entire model is formulated in the object oriented programming language Modelica [13] and is implemented in Dymola [14]. It uses the TILMedia [15] interface for accessing fluid properties from REFPROP [16].

The thermodynamic cycle is a steady state model and consists of heat exchangers, a compressor and a throttling valve. The heat is transferred from the heat source to the working fluid, which is evaporated and superheated at a low pressure, before it is compressed to a higher pressure. At this pressure, the working fluid rejects the heat to the heat sink while being cooled, condensed and subcooled, before it is expanded to the evaporation pressure again.

The heat transfer processes in heat source and sink are divided into single- or two-phase processes and are modelled in separate objects, irrespective of if they might be manufactured as one component.

The heat exchangers consist of energy balances and are equidistantly discretized in one dimension along the flow direction with respect to the transferred heat. The medium properties are defined by a medium state, which is calculated as the arithmetic average between inlet and outlet for each control volume. The single-phase parts of the heat exchangers are simulated as one control volume, whereas the two-phase parts are divided into seven volumes. According to the utilized heat transfer correlations and for numerical reasons, transcritical conditions are defined for pressures above 0.9 p_{crit}.

The condensation p_{cond} and evaporation pressure p_{evap} are indirectly defined by the definition of the pinch point temperature difference, $\Delta T_{\mathrm{pinch}}$=10 K given for all heat exchangers. The condensation pressure is fixed because the minimum temperature difference between working fluid and heat sink in the condenser and desuperheater is equal to $\Delta T_{\mathrm{pinch}}$. Additionally, the temperature difference between working fluid and heat sink at the outlet of the subcooler is set to $\Delta T_{\mathrm{pinch}}$. The evaporation pressure is defined by the minimum temperature difference between working fluid and heat source in evaporator and superheater.

In order to ensure a dry compression for both dry and wet working fluids, the inlet of the compressor is defined so both the superheat before and after the compressor [17] are at least 5 K.

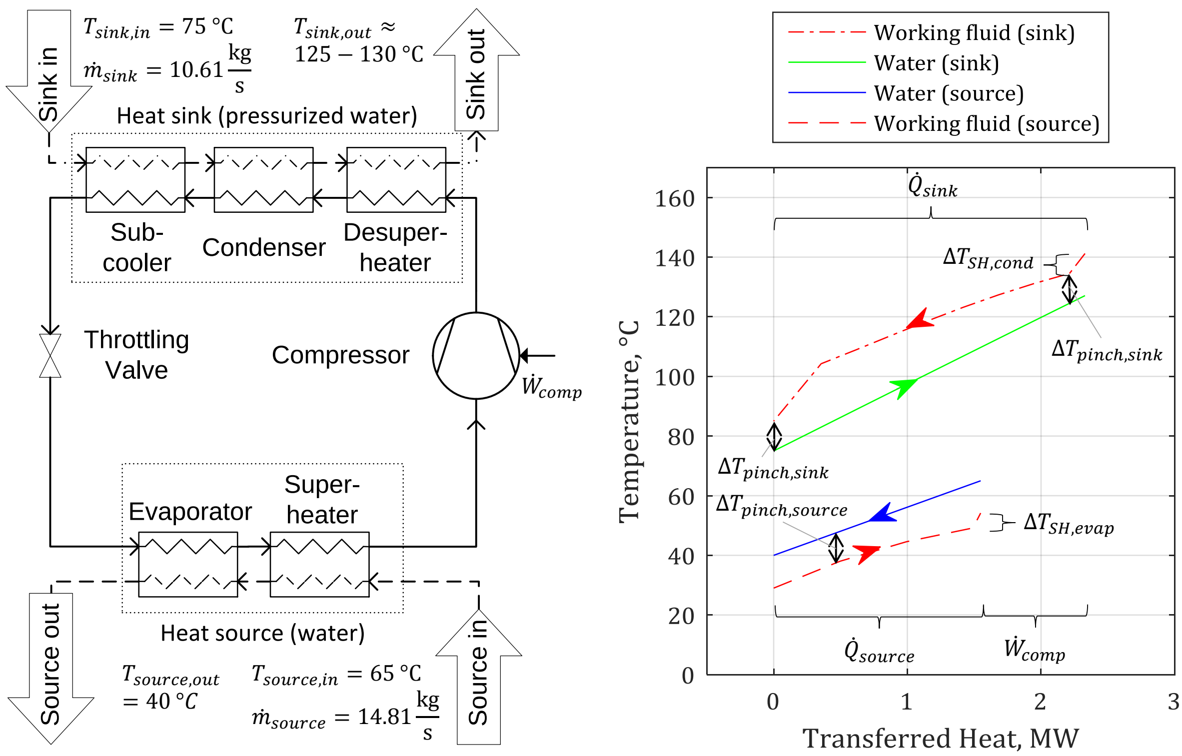


Figure 2: Flow sheet (left) and T-$\dot{Q}$-Diagram of heat pump model with an illustrative mixture profile (right)

The compressor is modelled with a constant isentropic efficiency $\eta_{comp,is}$=0.8 for the compression process and an additional motor efficiency of η_{motor}=0.95 [18], whereas the throttling valve is modelled as an isenthalpic expansion process. The losses in the motor are considered as heat losses to the environment.

Based on the described thermodynamic model the coefficient of performance COP is defined by the supplied heat $\dot{Q}_{sink}$ and the compressor power $\dot{W}_{compr}$ to be used a measure of the performance.

$$COP = \frac{\dot{Q}_{sink}}{\dot{W}_{compr}} \tag{1}$$

2.1.2. Component Sizing

The previous sections describe the calculation of thermodynamic efficiency measures, but optimization with respect to these does not necessarily coincide with the economic optimum. For enabling the possibility of an economic evaluation of the heat pump solution, it is crucial to size the components and calculate the costs.

The heat exchangers constitute a significant part of the heat pump application and the investment cost, which raises the influence of the sizing procedure. For this application plate heat exchangers (PHE) with chevron-type corrugation are chosen, since they have been increasingly used as evaporators and condensers for this temperature range [11], and allow a compact and modular design.

The sizing procedure of each heat exchanger accounts for both pressure drop and heat transfer area by determining number of plates and their length according to the medium properties and the thermodynamic cycle. Geometry of the corrugations and the plate width are chosen according to commercially available plates [11]. The free flow area and thereby the number of plates is determined by a maximum allowed velocity of the fluid on the working fluid side, for liquid of $v_{max,liq}$=0.4 m/s and for gas $v_{max,gas}$=4 m/s [19]. The pressure drops are not included in the thermodynamic cycle calculation, but analyses have shown reasonable values both on working fluid and water side.

The known medium properties, mass flow rates and geometrical parameters determine the heat transfer coefficients in each volume, used to calculate the required heat transfer area, yielding the length of the plates.

Based on different reviews [17], [20], [21], the correlations for the heat transfer from the fluid to the plate were chosen, according to the fluid state and the heat exchanger type:

- Subcritical single phase flow: Martin (1996) [22]
- Supercritical single phase flow: Petuhkov and Kirrillov (1958), simplified by Kays (1966), with parameters from [21]
- Condensation (of mixtures): Tandon (1986) [23]
- Evaporation (of mixtures): Gungor & Winterton (1987) using Thome & Shakir (1987) correction factor for the nucleate boiling contribution [24]

According to Ommen et al. [18], reciprocating piston compressors are chosen for this capacity, since they promise to show the best economic performance. The motor and other electrical equipment is considered to be suitable for flammable working fluids. The component cost for the compressor are determined by the suction volume flow rate, which takes a constant volumetric efficiency $\eta_{\mathrm{comp,vol}}$=0.8 into account [18]. Further parameters which limit the usability of compressors are the compression ratio $p_{\mathrm{cond}}/p_{\mathrm{evap}}$, the absolute pressure at the condenser side p_{cond} and the compressor discharge temperature $T_{\mathrm{comp,out}}$. Since limitations of these parameters are not reported precisely and consistent, they were not used as excluding criteria, but the results of promising cycles were evaluated towards being in a reasonable order with respect to these aspects. This procedure enables to point out, which limitations might need to be pushed by further developments.

2.1.3. Economic Calculations

In this study the net present value, NPV, is used as an indicator of the economic performance. It considers the costs related to the investment as well as annular costs for power operation and maintenance and revenues from saved natural gas consumption [25].

The NPV describes the time value of the investment at the time of the purchase and hence considers the total capital investment costs TCI as well as the annular cash flows (Eq. 2).

$$NPV = -TCI - OMC - \frac{FC_{HP}}{CRF} + \frac{FC_{NG}}{CRF} \tag{2}$$

The total capital investment is the sum of the investment of the components. The costs for installation, startup, working capital, etc. are summarized as a fixed factor of the components purchased equipment costs $\mathrm{PEC_j}$, which is estimated for an expansion of an existing plant as 4.16 [25]. The purchased equipment costs $\mathrm{PEC_j}$ of a component with the capacity X_{j} can be calculated by scaling of the cost $\mathrm{PEC_{ref}}$ for a reference component with the capacity of X_{ref} with the scaling factor β [25].

The inputs for the cost functions are summarized in Table 2 and taken from Ommen et al. [18], who analyzed the costs for industrial heat pumps in Denmark from suppliers and correlated those to the cost functions.

Table 2: Parameter for cost functions [18]

Component	Comment	$\mathrm{PEC_{ref}}$	X_{ref}	β
Compressor incl. elec. Motor	$p_{\mathrm{max}} = 28$ bar, $T_{\mathrm{max}} = 180$ °C, $5 - 280\ \mathrm{m^3/h}$	19,850 €	$279.8\ \mathrm{m^3/h}$	0.73
Plate Heat Exchanger		15,526 €	$42\ \mathrm{m^2}$	0.8

The continuous operation and maintenance costs (OMC) can be assumed as a onetime cost at the time of the investment and are 20 % of the TCI [25].

The annual fuel power consumption of the heat pump results from the power consumption of the compressor $\dot{W}_{\mathrm{compr}}$ and an estimation of 7,400 h/a for the annual operation hours. The saved annual natural gas consumption is determined by the heat load covered by the heat pump $\dot{Q}_{\mathrm{sink}}$ and an estimated value for the efficiency of the boiler η_{boiler}=0.9 [18]. The resulting cash flows consider fuel costs of c_{el}=0.0783 €/kWh for electricity and c_{ng}=0.0303 €/kWh for natural gas [26]. Subsidies and taxes are omitted from the calculations.

The annual cash flows are discounted to present day value using the capital recovery factor CRF. The value is determined by an effective interest rate i_{eff} = 4.9 %, assuming an interest rate of 7 %, an inflation rate 2 %, and the economic plant life of 20 years [18], [25].

2.2. Working Fluid Search

The consideration of mixtures as working fluids increases the degree of freedom of the working fluid search by the possible combinations and at a broad range of compositions. The higher degree of freedom causes a hardly manageable optimization problem. Thus, the list of pure fluids considered for designing the mixture is narrowed down by choosing the working fluids based on expectations about how they will behave in a mixture.

For some working fluid properties, it is reasonable to consider the properties of the pure fluids. So for e.g. the ozone depletion potential (ODP), the global warming potential (GWP) and toxicity it is required to have low values, which are accepted by current and future legislation [27].

Another basic aspect of the motivation to use mixtures as working fluids is to match the temperature profiles in the heat exchangers for decreasing the exergy destruction due to heat transfer. It can be observed, that fluids with an increasing difference between their normal boiling points tend to show a bigger temperature glide during phase change. Thus, in order to enable a high range of possible temperature glides by mixing the pure fluids, the boiling points of the pure fluids are chosen well distributed over a certain range. Nevertheless, the range of normal boiling points in which the fluids are considered is limited. A too high boiling point, with a respectively high critical temperature, can result in an evaporation pressure below atmospheric pressure and thus in the possibility of leakage into the system, whereas too low boiling points tend to result in high pressures.

The pure fluids should all be miscible with each other at a high range of temperatures and pressures without chemical reactions. This study uses the Hansen solubility parameter [28] to ensure miscibility.

Considering all these requirements, the following set of fluids is defined in Table 3 as a basis for creating binary mixtures. The chosen fluids are all hydrocarbons with a good distribution of the normal boiling points. Hydrocarbons are environmentally sound but mostly flammable [27]. Here, the flammability is accepted, since appropriate standards and regulations exist with suggestions for sufficient safety precautions [29]. Furthermore, flammable hydrocarbons such R290 and R601 are listed as suitable refrigerants for industrial high temperature heat pump applications by the IEA Heat Pump Centre Annex 35 for “Application of Industrial Heat Pumps” [30].

Table 3: List of pure fluids for systematic binary mixture search [16], [19], [31]

Acr.	Name of Fluid	Refrigerant No.	ODP (–)	GWP (–)	Normal Boiling Point (°C)	Critical Temperature (°C)	Critical Pressure (bar)
Pro1	Propane	R290	0	3.3	-42.0	96.7	42.5
IBu2	Iso-Butane	R600a	0	3.0	-11.7	134.7	36.4
nBu3	n-Butane	R600	0	4.0	-0.5	152.0	38.0
IPe4	Iso-Pentane	R601a	0	4.0	27.8	187.3	33.8
nPe5	n-Pentane	R601	0	4.0	36.1	196.6	33.7
nHe6	n-Hexane				68.7	234.5	30.3

The overall goal of the approach is to analyze the influence of choosing a mixture as the working fluid based on different evaluation criteria, such as energetic and exergetic efficiencies, as well as the NPV. The procedure consists of a parameter study, in which the medium composition is varied. The model is simulated for all possible binary mixtures at varying compositions between 0 % and 100 % in 10 equidistant steps. This results in 141 heat pump simulations, in which the cycle as well as the component costs are calculated and subsequently analyzed.

Based on the results of the study, it will be evaluated if these fluids are sufficient or if it seems promising to add further fluids with specific characteristics and if it is sufficient to consider only binary mixtures or if more component mixtures could bring additional benefit.

3. Results

Figure 3 presents an overview of COP (left) and NPV (right) for all possible binary mixtures among the fluids listed in Table 3. The combinations in the legend correspond to the acronyms in the first column of Table 3. The abscissas of the diagrams show the mass fraction of component 2, which is also the less volatile component. This means, that on the left hand side, the diagrams show values for the pure fluids of the more volatile component and on the right hand side vice versa for the less volatile component.

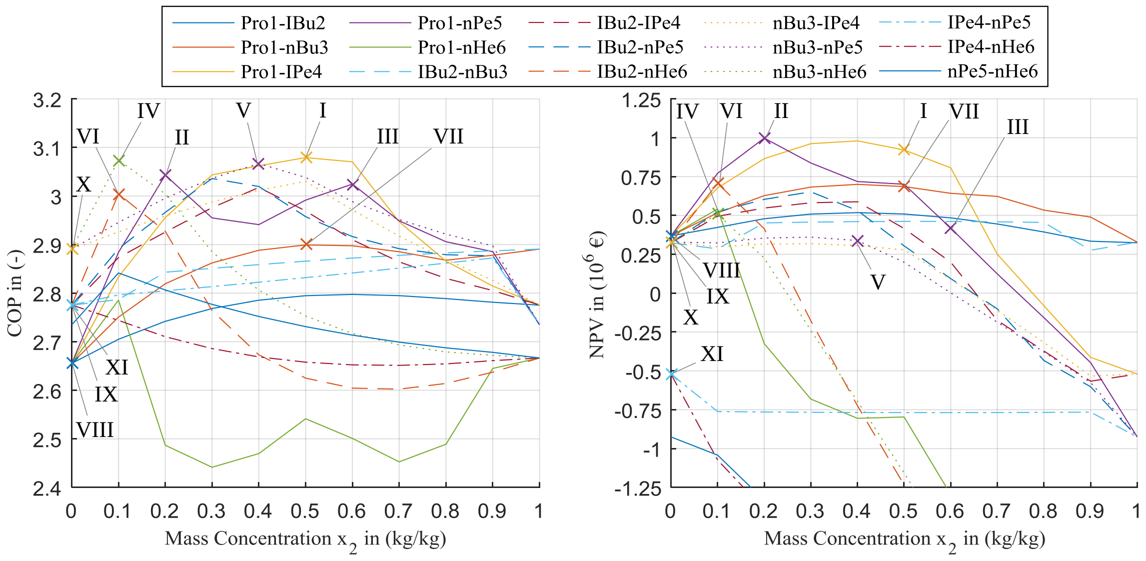


Figure 3: COP (left) and NPV (right) for all simulations. Interesting cases are marked with roman numbers (I-XI). The legend corresponds to the combination of fluids in Table 4.

All cases show a COP between 2.4 and 3.1, and significant variations are observed for each mixture for varying compositions. Some mixtures show a general trend of a positive NPV, while others are hardly feasible. The trend is that higher concentration of the more volatile component is beneficial. The figure shows a nonlinear behavior of COP along the composition, which is caused by opposing effects. It can be seen that COP depends on the match of the temperature profiles, required superheating for very dry fluids, required mass flow rates and further other effects. The most promising cases based on both criteria are numbered by roman numerals and summarized in Table 4.

According to COP, the 50/50 Propane–Iso-Butane mixture (COP_I=3.08), the 90/10 n-Butane–n-Hexane mixture (COP_{IV}=3.08) and the 60/40 n-Butane–n-Pentane mixture (COP_V=3.08) show the best performance, while further solutions with similar values exist. The best pure components are n-Butane (COP_X=2.89), Iso-Butane (COP_{IX}=2.78) and Iso-Pentane (COP_{XI}=2.78). This corresponds to an increase of 6.6 % of the COP for the utilization of mixtures compared to the best pure fluids considered here.

Table 4: Summary of key data and results for the most relevant cases. The best solutions are in bold letters.

No.	Fluid	COP	NPV	TCI	$\frac{p_{cond}}{p_{evap}}$	p_{cond}	p_{crit}	p_{evap}	$T_{comp,out}$	$\dot{V}_{comp,in}$
		(−)	(10^3 €)	(10^3 €)	(−)	(bar)	(bar)	(bar)	(°C)	(m^3/h)
I	50/50 Propane – Iso-Pentane	**3.08**	921	661	6.7	32.9	47.3	4.9	139	2220
II	80/20 Propane – n-Pentane	3.04	**998**	553	5.5	45.7	47.3	8.4	142	1487
III	40/60 Propane – n-Pentane	3.02	419	1011	8.1	24.4	47.1	3.0	139	3288
IV	90/10 n-Butane – n-Hexane	3.07	509	999	9.1	22.9	39.0	2.5	138	3668
V	60/40 n-Butane – n-Pentane	3.07	339	1132	10.2	19.4	37.7	1.9	138	4597
VI	90/10 Iso-Butane – n-Hexane	3.00	707	745	8.1	29.1	38.3	3.6	138	2914
VII	50/50 Propane – n-Butane	2.90	686	617	6.8	41.7	43.4	6.1	139	1966
VIII	Propane	2.66	369	473	5.8	62.7	42.5	10.8	139	1408
IX	Iso-Butane	2.78	324	724	8.9	36.1	36.3	4.0	138	2922
X	n-Butane	2.89	328	902	9.9	28.1	38.0	2.8	139	3519
XI	Iso-Pentane	2.78	-521	1429	14.8	14.6	33.8	1.0	140	8272

The economic analysis shows that according to the NPV several economically profitable solutions exist. Whereas case I and IV have shown the best energy performances, an 80/20 Propane-n-Pentane mixture promises to show the best economic performance with a NPV of 998,000 €. Also, case I shows a comparable economic potential, whereas other thermodynamically promising solutions are eliminated by too high investment costs and thus, a poor economic performance. Among the pure refrigerants, Propane, Iso-Butane and n-Butane indicate a profitable performance with NPVs between 328,000 € and 369,000 €.

The investment cost for the compressor is determined by the volume flow rate at the inlet $\dot{V}_{\mathrm{comp,in}}$, which causes a high investment cost for fluids with a low density at compressor inlet. This can be noticed, when comparing Iso-Butane and Iso-Pentane with equal COP but large differences in NPV. The volume flow rate at the compressor inlet is 2.5 times as much for Iso-Pentane than for Iso-Butane, which yields doubled investment costs for Iso-Pentane. For the same reason the NPV decreases with an increasing share of n-Pentane and n-Hexane.

The NPVs of n-Butane as the economically best subcritical pure fluid, Propane as the best supercritical pure fluid and 80/20 Propane-n-Pentane as the best mixed refrigerant are equal to a payback time of 9.2 years, 7.1 years and 4.5 years.

The proposed heat pump solution II utilizes 1.55 MW of waste heat from the exhaust gas by cooling it from 70 °C to 45 °C for heating the drying air from 70 °C to 120 °C, which is equal to a heat load of 2.25 MW, while consuming 740 kW electrical power. The supplied heat load replaces a yearly natural gas consumption of $2.0\ \mathrm{mio.\,m^3}$, which means a reduction of 36 %.

The total capital investment cost of the heat exchangers in the secondary cycles for the considered case are in total approximately 85,000 € and can be taken into account if applicable.

4. Discussion

The work is based on numerical models of the component performance, the fluid properties and the economics. All these involve uncertainties, which may have significant impact on the results. Acknowledged models of each involved item were used, but exact conclusions for individual industries will depend on suppliers and negotiations with the customer. It is found that the compressor has a significant impact on the NPV, however the model uses primarily the suction volume flow rate for cost estimation. Some results, e.g., solution II, have a condensation pressure close to the critical pressure. This means that heat transfer is considered to be in the supercritical region and thus implies an increased uncertainty in the applied correlation. Furthermore, all calculated heat exchanger sizes depend on the procedure of sizing the heat exchanger, which is not optimized for each solution. Values for pressure drop and heat transfer coefficients were in an expected range. The supercritical processes typically imply increased investment cost due to higher pressures, which were not accounted in this study. Nevertheless, the compressor gives the most relevant contribution to the investment costs, which decreases the influence of uncertainties from heat exchangers. Possible additional application specific investment costs for e.g. the heat exchangers of the secondary cycles can also influence the result, but for the shown case these cost were coverable.

Additionally the solution is dependent on the boundary conditions. The fixed source outlet temperature defines the temperature span between sink and source and thus, the maximum achievable efficiency. This means, the source outlet temperature can be increased to accomplish stricter economic requirements. Furthermore, site-specific energy prices, subsidies and taxes or emission costs (e.g. CO_2-tax in Denmark or EU-emission certificates) may influence the profitability significantly, which requires a detailed analyses for each case.

The heat exchanger model depends on the level of discretization. For the scope of this work, it is found that the chosen number of control volumes ensures sufficient accuracy with respect to the presented results.

The economic boundary conditions are assumed for industrial plants in Denmark in 2015. Since the NPV is strongly dependent on the assumptions, the results here should rather serve as a measure for comparing the different investments for these specific assumptions than giving absolute values for this kind of technology.

5. Conclusion

The analysis of available waste heat has shown the potential for efficiency improvements in drying processes and the demand for appropriate technologies. It indicates that heat pumps with zeotropic mixtures are promising technologies for an efficient exploitation of heat sources. The paper presents an approach for improved heat pump integration by considering working fluid mixtures for an enhanced exploitation of the heat sources with temperature glide. The results have shown different solutions, which seem promising under consideration of energy-based and economic performance indicators. The best economically and technically feasible performance was reached by 50/50 Propane–Iso-Pentane (COP=3.08, NPV=921,000 €) and 80/20 Propane–n-Pentane

(COP=3.04, NPV=997,000 €), which outperforms n-Butane (subcritical, COP=2.89, NPV=328,000 €) and Propane (supercritical, COP=2.66, NPV=369,000 €) as the best pure refrigerants, considered in this study.

Mixtures can constitute a technology with an increased energy efficiency and economic potential while being in a technically feasible range. The proposed solution could save 36 % of the natural gas consumption by use of electric power instead.

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