CASE STUDY OF HEAT PUMP INTEGRATION IN AN ANIMAL FOOD DRYER

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ABSTRACT

In general, most heat losses in industrial dryers arise due to the discharge of moist air. By using heat pump drying systems, (partial) heat from exhaust moist air can be recovered thus improving the energy efficiency substantially. In this study, the performance of heat pump integration in an animal food dryer is examined. Computer simulation models of the original high temperature dryer and the proposed system with heat pump integration and auxiliary heating are developed. Different heat pump cycles and working fluids are investigated in order to determine the best performing heat pump system. In addition, the economic benefit as well as the optimum operating conditions of the dryer with heat pump integrated were also determined. Finally a newly built test rig is described which will be used to demonstrate the original and optimized dryer with heat pump integration or with other measures such as recirculation or in/out heat exchanger to validate of the computer simulations and to convince the plants owners.

INTRODUCTION

Drying is an essential operation in the chemical, agricultural, biotechnology, food, polymer, ceramics, pharmaceutical, pulp and paper, mineral processing, and wood processing industries. For many industrial sectors, drying is one of the most energy-intensive processes. It consumes, for example, up to 70% of all the energy used in the wood industry and 50% in textile manufacturing (Mujundar 2007). In the developed countries it consumes up to 25% of the national industrial energy (Minea 2011). By integrating a heat pump both sensible and latent heat can be recovered from the dryer exhaust air, improving the overall thermal performance (Adapa et al. 2002). Heat pump drying operates well at medium to low temperatures as well as for pre-heating in high temperature applications (Alves-Filho 2013).

The results of various studies, from numerical simulation studies to experimental tests, have proven these advantages of heat pump drying. For example, in the first study of heat pump drying in 1973 by Hodgett, the author reported that the energy consumption of a heat pump dryer was less than on conventional steam heated drying (Colak et al. 2009). Queiroz et al. 2004 carried out experimental studies on tomato drying and compared the performance of heat pump drying with hot air drying.
using electrical resistances. It was shown that the coefficient of performance (COP) of the heat pump was between 2.58 and 2.68 and the energy consumption reduced by about 40%. Similar results were obtained for apple drying (Gabas et al. 2004) where the heat pump COP varied from 2.48 to 2.58 and energy consumption reduction was about 38% to 47%.

Even though the advantages of heat pump drying are well known today, it is not yet applied as widely as it could be for industrial processes. Hence, future R&D work on heat pump drying should be carried out in close cooperation with industry (Minea 2013).

Within the frame of the transnational CORNET HP4Drying project, several industrial case studies are performed, including one on a dryer in an animal food production line that will be presented here.

**SIMULATION AND FORMULATION**

To study the performance of heat pump integration in an animal food dryer, the original high temperature dryer and the proposed system with heat pump integration and auxiliary heating are simulated in the Engineering Equation Solver (EES) software package (Klein 2009). With these models the COP can be calculated, as well as the optimum operating conditions of the heat pump.

*Simulation of the original dryer*

![Figure 1. Simulation of original dryer.](image)

The simulated process of the original dryer is represented shown in Figure 1. In the first stage, the initial product with moisture content of 24% (based on dry product) and temperature of 100°C entering from the top of the drying chamber is dried by hot air. In the cooling stage, dried product is then cooled by ambient air and exits the drying chamber at a temperature and final moisture content of around 47°C and 10.3% respectively (based on dry product).
The drying air used to dry product in the drying stage (point 4) is the mixture of hot air (point 3, $T_{a3}=140^\circ C$) and the air after the cooling stage (point 2). After the drying stage, the air at the exit of the drying chamber is passed through a cyclone to remove dust. The exhaust moist air temperature is around 50°C.

The energy consumption of the original dryer is the heat needed to heat air from ambient temperature to the required temperature (140°C) and defined as:

$$Q_{\text{original dryer}} = m_{\text{air}} \times (h_{140} - h_{\text{ambient}}) \quad (1)$$

**Simulation of a dryer with a simple heat pump integrated**

![Diagram of a dryer with a simple heat pump system integrated.](image)

Figure 2 shows the simulation of a dryer with a simple heat pump system integrated. In this heat pump drying system, the evaporator is employed to recover heat from exhaust moist air while the condenser is used to preheat the ambient air. The preheated air after the condenser is heated again to 140°C by an auxiliary heat exchanger.

To calculate the performance of a heat pump drying system, a set of inputs such as: working fluid, isentropic efficiency of the compressor, pinch point of evaporator and condenser, volumetric mass flow rate of drying air must first be entered by the user. Once calculated, the chart at the bottom right corner of the Figure 2 shows the T-s diagram of the calculated heat pump system while the one at the top right corner shows the dry bulb temperature and humidity ratio of the drying air at different points of the drying process. The performance of a heat pump is represented by the COP and is defined as:

$$\text{COP} = \frac{Q_{\text{cond}}}{W_{\text{comp}}} \quad (2)$$

In this study, several heat pump systems as followed are also investigated and compared in order to find the most suitable heat pump system for this drying case.
Simulation of a dryer with a heat pump and a suction line heat exchanger (SLHX)

The heat pump using a SLHX is presented in Figure 3. In this configuration, the heat is transferred from high temperature liquid leaving the condenser to low temperature refrigerant vapor leaving the evaporator. In other words, liquid refrigerant is subcooled before entering the expansion valve while vapor refrigerant is superheated before entering the compressor. Thus, using a SLHX helps to prevent flash gas formation at the expansion valve inlet and the risk of liquid refrigerant at the compressor inlet (Klein et al 2000).

The transcritical CO₂ cycle investigated in this study also employs a SLHX. The T-s diagram of a transcritical CO₂ cycle is presented in Figure 4. The lines 5-6 and 2-3 present the subcooling process of hot liquid and superheating of cold vapor, respectively.

The COP of a heat pump employing a SLHX is also defined by (2).

Simulation of a dryer with a heat pump employing a subcooler

Another system investigated is the system using a subcooling heat exchanger. As shown in Figure 5a, ambient air is preheated by a subcooler and condenser before it enters an auxiliary heat exchanger and is heated to the required temperature. Since the subcooler is used to heat the drying air as well, the COP is calculated as:

\[
\text{COP} = \frac{Q_{\text{cond}} + Q_{\text{subcooler}}}{W_{\text{comp}}} \tag{3}
\]

Simulation of a dryer with heat pump with an external heat exchanger

Since the exhaust air temperature is around 50°C, much higher than the ambient air temperature, an external air/air heat exchanger can be used to recover heat from the exhaust moist air to preheat ambient air (Figure 5b). The preheated drying air leaving the external heat exchanger is then passed through the condenser and an auxiliary heat exchanger like on previous systems.

For an electrically driven heat pump, the total energy consumption of the dryer with integrated heat pump is:

\[
Q_{\text{HPD}} = W_{\text{comp}} + m_{\text{air}} \times (h_{140} - h_{\text{elec HP}}) \tag{4}
\]
Simulation of a dryer with a gas engine driven compressor heat pump

For an electrical heat pump, the economic benefit depends on the ratio of electricity to gas price. It is more favourable to implement an electrical heat pump when this ratio is small and vice versa. Using a gas engine driven compressor heat pump instead of an electrical heat pump is a solution to avoid this dependence. The simulation of this drying system is presented in Figure 6. The compressor of the heat pump is driven by a gas engine. The available heat from the coolant fluid and exhaust gases of the gas engine are also utilised to heat drying air before it enters the auxiliary heat exchanger. Therefore, the performance of a gas engine driven compressor heat pump is represented by combined heat and power efficiency ($\eta_{CHP}$) and calculated as the ratio of useful heat used to heat drying air to energy consumption of the gas engine:

$$\eta_{CHP} = \frac{Q_{useful}}{Q_{fuel}} = \frac{Q_{cond} + Q_{coolant} + Q_{exhaust}}{Q_{fuel}} = \frac{\text{COP} \times W_{\text{comp}} + (Q_{\text{coolant}} + Q_{\text{exhaust}})}{Q_{fuel}} = \text{COP} \times \eta_{\text{mech}} + \eta_{\text{thermal, coolant fluid}} + \eta_{\text{thermal, exhaust gases}}$$ (5)
where $\eta_{\text{mech}}$, $\eta_{\text{thermal, coolant fluid}}$ and $\eta_{\text{thermal, exhaust gases}}$ are:

$$
\eta_{\text{mech}} = \frac{W_{\text{comp}}}{Q_{\text{fuel}}}
$$

$$
\eta_{\text{thermal, coolant fluid}} = \frac{Q_{\text{coolant}}}{Q_{\text{fuel}}}
$$

$$
\eta_{\text{thermal, exhaust gases}} = \frac{Q_{\text{exhaust}}}{Q_{\text{fuel}}}
$$

For a gas engine driven heat pump, total energy consumption of a dryer with a heat pump integrated is:

$$
Q_{\text{HPD}} = Q_{\text{fuel}} + m_{\text{air}} \times (h_{140} - h_{\text{gas HP}})
$$

(6)

**Economic benefit calculation**

In this study, the economic benefit of the dryer with a heat pump integrated is expressed by the relative energy cost, which is defined as the ratio of the energy cost of the heat pump dryer to the energy cost of the original dryer.

$$
\text{Relative energy cost} = \frac{\text{Total energy cost of dryer with integrated heat pump}}{\text{Energy cost of original dryer}} \times 100 \% \quad (7)
$$

The primary goal of this paper is to determine the heat pump configuration and operating condition in which the (relative) energy cost is minimal.

The calculations in this paper are based on the assumptions that the refrigerant flow rate is constant and heat and pressure losses are neglected. The other inputs used to calculate and compare the performance of the original dryer and the proposed systems are presented in Table 1.

**Table 1. The inputs for simulating calculation**

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Unit</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Isentropic efficiency of the compressor</td>
<td>-</td>
<td>0.7</td>
</tr>
<tr>
<td>Pinch point of the evaporator</td>
<td>K</td>
<td>8</td>
</tr>
<tr>
<td>Pinch point of the condenser</td>
<td>K</td>
<td>8</td>
</tr>
<tr>
<td>Pinch point of the subcooler</td>
<td>K</td>
<td>15</td>
</tr>
<tr>
<td>Pinch point of the gas cooler (the CO$_2$ system)</td>
<td>K</td>
<td>15</td>
</tr>
<tr>
<td>Subcooling temperature of the simple system</td>
<td>K</td>
<td>1</td>
</tr>
<tr>
<td>Superheating temperature</td>
<td>K</td>
<td>5</td>
</tr>
<tr>
<td>Effectiveness of the SLHX</td>
<td>-</td>
<td>0.75</td>
</tr>
<tr>
<td>Effectiveness of the external heat exchanger</td>
<td>-</td>
<td>0.5</td>
</tr>
<tr>
<td>Percentage of heat recovered by the coolant heat exchanger ($\eta_{\text{thermal, coolant fluid}}$)</td>
<td>-</td>
<td>0.302</td>
</tr>
<tr>
<td>Percentage of heat recovered by the exhaust gases heat exchanger ($\eta_{\text{thermal, exhaust gases}}$)</td>
<td>-</td>
<td>0.186</td>
</tr>
<tr>
<td>Mechanical efficiency of gas engine ($\eta_{\text{mech}}$)</td>
<td>-</td>
<td>0.364</td>
</tr>
<tr>
<td>Ambient air temperature</td>
<td>C</td>
<td>20</td>
</tr>
<tr>
<td>Ambient air relative humidity</td>
<td>%</td>
<td>60</td>
</tr>
<tr>
<td>Volumetric flow rate of the hot air</td>
<td>m$^3$/h</td>
<td>12000</td>
</tr>
<tr>
<td>Volumetric flow rate of the air used for cooling stage</td>
<td>m$^3$/h</td>
<td>5000</td>
</tr>
</tbody>
</table>
RESULTS AND DISCUSSION

Generally, we would like to recover as much heat as possible from the exhaust air to reduce heat losses. However, more recovered heat means the heat pump will operate at a lower evaporation temperature and thus pressure. Moreover, as mentioned above, the heat recovered from the exhaust air is then used to preheat the ambient air. Thus, the more heat recovered from the exhaust air, the higher is the temperature of the preheated air. Consequently, an increasing temperature difference between the condensation and evaporation decreases the heat pump COP. Hence, there is a trade-off between the amount of heat recovered and the heat pump COP.

Comparison among different system configurations

According the results of the simulation represented in Figures 7a, 7b and 7c, for all three working fluids, the COP of the system using a subcooling heat exchanger is the highest one. This is because more useful heat is utilised by adding a subcooler to the system thus improving the COP. The benefit of the subcooler rises with the air temperature after the condenser ($T_{a_{HP}}$) since the proportion of the heat release at the subcooler to the heat release at the condenser increases with increasing $T_{a_{HP}}$. For example, for the R245fa heat pump, the COP of a subcooler heat pump system is about 36% higher in comparison with a simple system at $T_{a_{HP}}$ of 60°C while this increase is about 54% at $T_{a_{HP}}$ of 80°C. Among three fluids, employing a subcooler is most effective for the R134a system, for which the improvement is around 44% and 76% at $T_{a_{HP}}$ of 60°C and 80°C, respectively. The corresponding values for an ammonia system are only 23% and 28%, respectively.

The results also show that using a SLHX slightly influences the performance of the heat pump but this influence is not the same for all working fluids. As can be seen from Figures 7a and 7b, for systems using R245fa or R134a as working fluids, the COPs of SLHX systems are up to 10%
higher in comparison with simple systems. For an ammonia heat pump on the other hand, using a SLHX decreases system performance by around 4% (Figure 7c). These results correspond with previous study (Domanski et al. 1994).

**Comparison among working fluids**

The results given in Figure 8a show that the COP of a transcritical CO₂ system is higher than for those of simple systems using ammonia, R134a or 245fa. Among three simple systems, the system using ammonia is the best performing one, followed by R245fa system with a slightly lower COP of about 5%. The R134a system has the worst COP being up to 20% lower than that of an ammonia system.

However, the results are opposite for systems employing a subcooler. As can be seen from Figure 8b, the COP of a transcritical CO₂ system now is lower than those of systems employing a subcooler. Meanwhile, as a result of significant improvement by using a subcooler, the COP of a R134a system is up to 12% higher than an ammonia system and only around 2% lower than the best performing one being R245fa.

Since the subcooler heat pump using the R245fa as working fluid has the best COP, from now on only the R245fa heat pump will be used for further discussions.

**Effectiveness by using an external heat exchanger**

The comparison of performance and relative energy cost between the systems using and not using an external heat exchanger is shown in Figures 9a and 9b. By employing an external heat exchanger the air gets precooled before it goes to the evaporator and preheated before it goes to the condenser, in other words the temperature lift of the heat pump increases. Therefore, employing an external heat exchanger leads to a small decrease of the COP for the dryer with a simple heat pump system (Figure 9a). This effect is more significant for the systems employing a subcooler since it also decreases the subcooling. The reduction of the COP is from 9% to 27% when T_{a HP} varies from 80°C to 60°C for the heat pump using a subcooler (Figure 9b).

However, because a proportion of the heat is already supplied by the external heat exchanger, if the ambient air is preheated by the heat pump to a same temperature for the two systems, then the heat needed from the heat pump of the system using an external heat exchanger is from 62% to 75% of the heat required from the heat pump of the system without an external heat exchanger. As a result, even with a lower COP, the compressor energy consumption and the total energy cost of a system using an external heat exchanger are lower in comparison with the system that does not use an external heat exchanger. For the systems with a heat pump employing a subcooler and an...
electricity/gas price ratio of 3.33 as an example, the energy cost saving is up to 18.4% for a system without external heat exchanger while this value rises up to 23.7% if an external heat exchanger is employed (Figure 9b).

It should also be noted that the optimum operating conditions are different for the two systems due to the different COPs when employing an external heat exchanger. For instance, according to Figure 9b, the optimum Ta HP moves from 69°C to 73°C when an external heat exchanger is employed.

**Influence of electricity to gas price ratio on optimum operating conditions and economic benefit**

Figure 10 illustrates the influence of the electricity to gas price ratio on the performance of the dryer with a subcooler heat pump integrated as well as an external heat exchanger. As expected, the lower the price ratio, the lower the total energy cost is. In addition, it is worth noting that the optimum Ta HP significantly depends on price ratio. For a price ratio of 3.52, the minimum energy cost is 77.4% of the original one at Ta HP of around 69°C. However, when the price ratio drops to 3.19, the optimum relative energy cost and corresponding Ta HP are 75.4% and 76°C, respectively.

**Effectiveness of using an external heat exchanger on dryer integrated with a gas driven compressor heat pump**

As can be seen from the results represented in Figure 11, energy cost of the systems decreases almost linearly by increasing Ta HP until it reaches optimum value Topt. This optimum value depends on the maximum temperature of the drying air at the exit of the coolant heat exchanger (Ta4, Figure 6) or, in other words, the coolant fluid temperature and the minimum temperature difference of the coolant heat exchanger. Indeed, when Ta HP is lower than Topt and corresponding Ta4 is lower than this Ta4max all the heat available from the coolant fluid can be used to heat the drying air, therefore the energy cost decreases when Ta HP increases. Once Ta HP reaches its optimum value and Ta4 reaches Ta4max, if we continue to increase Ta HP the energy consumed by the gas engine increases in
order to supply enough mechanical work to the compressor, whereas the useful heat from the coolant fluid used to heat the drying air decreases. As a result, $\eta_{CHP}$ significantly decreases and the energy cost rises. Generally, the exhaust gas temperature is much higher than the coolant fluid temperature therefore the optimum operating temperature does not depend on the exhaust gas temperature. However, the percentage of heat recovered by exhaust gases affects the performance of the gas engine driven heat pump according to (5).

As was the case for an electrically driven heat pump system, the gas engine driven compressor heat pump system using an external heat exchanger can save more energy costs even though $\eta_{CHP}$ is lower in comparison with system without external heat exchanger. At optimum operation, the energy cost of the systems with and without external heat exchanger are 64.3% and 67.2% of the energy cost consumed by the original dryer, respectively.

**CONCLUSIONS AND FUTURE WORK**

This paper presents the results of a case study when a heat pump should be integrated in an animal food dryer. In order to study the performance of heat pump dryers, computer simulation models of the original high temperature dryer and the proposed system with heat pump integration and auxiliary heating are developed. The results showed that a R245fa heat pump system with subcooling heat exchanger is the best solution. Furthermore, it is also found that, heat from exhaust air should first be partially recovered by an external heat exchanger then by a heat pump. For the electrically driven heat pump, the energy cost and the optimum operating conditions significantly depend on the price ratio of electricity to gas. When using a gas engine driven compressor heat pump, in which recovered heat from the coolant fluid and exhaust gas of the gas engine is also used to heat drying air, the maximum energy cost saving is limited by the temperature of the coolant fluid.

To overcome the limitation because of cooling fluid temperature, a (micro) gas turbine driven compressor could be a good solution since a gas turbine only has one waste heat stream being exhaust gases. This heat pump system and other heat pump drying systems such as chemical heat pump, absorption heat pump...will be investigated in the next step of the project. Furthermore, within this project an electrically driven heat pump dryer test bench is being developed in order to verify the results from simulations (Figure 12). The test bench aims to demonstrate various optimizations to the drying and heat pump cycle. For the air cycle this includes recirculation and air heat exchangers, for the heat pump cycle this includes the use of a suction line heat exchanger, subcooler, high temperature fluid and innovative control system. The heat pump features an open compressor, which enables precise mechanical power measurement and factors out compressor motor efficiency. In turn, this allows practical simulation of a gas engine powered heat pump dryer. The ‘recovered heat’ from the exhaust gasses is substituted by an additional electric heat, using data and extensive experience from CHP units.

To aid repeatability of test conditions and optimization of a climate dependent control algorithm, the test bench is fed with conditioned air, capable of following a preset summer/winter day/night
cycle. The conditioning unit mainly consists of a capacity controlled R404a refrigeration system, electrical heating and steam humidifier.

Figure 12. Diagram of the test bench.

A main objective is replicating the exhaust air of measured drying processes by means of water evaporation. This way the heat pump dryer can be tested for a wide range of products and different drying schemes, allowing simulation of batch and continuous processes from ongoing case studies. To our knowledge a similar test bench has not yet been developed.

ACKNOWLEDGEMENT

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NOTATION

- \textbf{COP} coefficient of performance
- \textbf{h}_{140} \text{ enthalpy of drying air at } 140^\circ\text{C}
- \textbf{h}_{\text{ambient}} \text{ enthalpy of ambient air}
- \textbf{h}_{\text{elec HP}} \text{ enthalpy of drying air preheated by electrically driven heat pump}
- \textbf{h}_{\text{gas HP}} \text{ enthalpy of drying air after preheated by exhaust air heat exchanger}
- \textbf{m}_{\text{air}} \text{ mass flow rate of drying air}
- \textbf{Q}_{\text{cond}} \text{ heat release rate at condenser}
- \textbf{Q}_{\text{coolant}} \text{ heat release rate at coolant heat exchanger}
- \textbf{Q}_{\text{fuel}} \text{ energy consumption of gas engine kW}
- \textbf{Q}_{\text{exhaust}} \text{ heat release rate at gas engine exhaust air heat exchanger}
\[
\begin{align*}
Q_{\text{HPD}} & \quad \text{energy consumption of dryer integrated heat pump} \quad \text{kW} \\
Q_{\text{original dryer}} & \quad \text{energy consumption of original dryer} \quad \text{kW} \\
Q_{\text{subcooler}} & \quad \text{heat release rate at subcooler} \quad \text{kW} \\
T_{\text{TaHP}} & \quad \text{temperature of drying air preheated by heat pump} \quad ^\circ\text{C} \\
W_{\text{comp}} & \quad \text{energy consumption of compressor} \quad \text{kW} \\
\eta_{\text{CHP}} & \quad \text{combined heat and power efficiency} \quad -
\end{align*}
\]

REFERENCES


