

CASE STUDIES OF THERMALLY DRIVEN HEAT PUMP ASSISTED DRYING

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ABSTRACT

In general, most heat losses in industrial dryers arise due to the discharge of humid air. By using heat pump drying (HPD) systems, heat from the exhaust humid air can be recovered, thus improving the energy efficiency substantially. In this study, the performance of thermally driven HP integration in an animal food and a blood dryer were examined. Computer simulation models of the original high temperature dryers and the proposed system with HP integration and auxiliary heating were developed. It is found that, when using a gas engine, the maximum energy cost saving is limited by the temperature of the coolant fluid. The maximum energy cost saving when using a gas turbine is a bit higher, however at a much higher operating temperature.

Keywords: heat pump drying; simulation; energy analysis; economic benefit; hp4drying

INTRODUCTION

Drying is one of the most energy intensive processes for many industrial sectors. Generally, heat losses in industrial dryers are mostly due to the discharge of moist air and conduction through the drying chamber walls [Minea, 2015]. By using HPD systems, which have the ability to recover partially this heat loss, up to 50% or more primary energy used can be saved [Minea, 2015]. In order to evaluate the performance of the HPD, various studies, from numerical simulation to experimental tests, have been carried out and published. For example, Minea, 2012, carried out experimental studies on wood drying and compared the performance of HPD with the conventional dryer. It was shown that the HP dryer saves up to 48% of the total equivalent (electrical and fossil) energy use. A batch-type HP dryer for medicinal plants was investigated by Ziegler et al., 2013. It is found that the energy efficiency of the HPD system operated in partially open mode is higher in comparison with a closed cycle system. While comparing with a conventional dryer, the energy efficiency of a partially open HPD can be higher or lower dependent on the ambient temperature. In the study of Prasertsan et al., 1996, 1997^{a, b}, two open and two partially closed HPD systems using R22 as working fluid were investigated. The influences of the ambient air conditions, the recirculation air ratio (RAR) as well as bypass air ratio (BAR) on the performance of these systems were determined. Dryers using complex HP systems such as two-stage evaporator systems [Hawladar et al., 2001; Chua et al., 2005] and two cycle HPs [Lee et al., 2010] were also investigated.

However, when comparing to a conventional dryer using primary energy as the heat source, the operating energy cost of an electrical driven HP dryer significantly depends on the ratio of electricity to gas price. To avoid this dependence, employing a thermally driven HP such as a gas engine/turbine driven compressor HP could be a good solution. The performance of these HPD systems integrated in two drying processes is investigated and presented in this study.

Models of the original high temperature dryers and of the proposed system with HP integration and auxiliary heating are implemented in Engineering Equation Solver (EES).

Simulation of the original dryers

The layouts of the two original dryers are represented in figure 1. In these drying processes, the wet product enters into the upper part and exits from the bottom part of the drying chamber. However, there are some differences in the air path of the two drying processes. In the animal food drying process, the fresh air is directly heated up to 140°C by a heater before it comes to the drying chamber and the exhaust air temperature exits the cyclone at around 50°C. Meanwhile, in the blood drying process, a water/air heat

exchanger is used to recover heat from the exhaust air after the cyclone and pre-heats the fresh air to around 50°C before it is heated up to the required temperature of 180°C by the heater. The temperature of the exhaust air after the water/air heat exchanger is around 50°C.

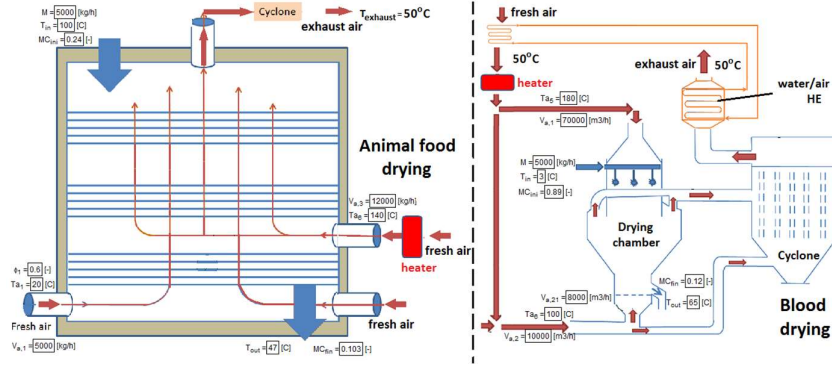


Figure 1: Layouts of the original animal food dryer (left) and blood dryer (right)

The power use of the original dryer is the heat rate needed to raise the temperature of the incoming air to the required temperature and is defined as:

$$Q_{\text{original dryer}} = m_{\text{air}} (h_{\text{heated air}} - h_{\text{incoming air}}) \quad [1]$$

where $h_{\text{heated air}}$ is the specific enthalpy of the drying air after the heater.

$h_{\text{incoming air}}$ is the specific enthalpy of the fresh air for the animal food case and of the preheated air for the blood drying case.

Simulation of a dryer with a gas engine/turbine driven HP integrated

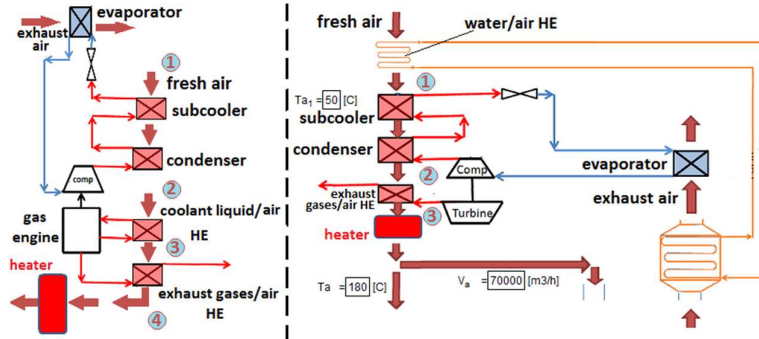


Figure 2: Layouts of a dryer with a gas engine driven compressor HP integrated in the animal food dryer (left) and with a gas turbine driven compressor HP integrated in the blood dryer (right)

The HP system investigated in this research is a R245fa HP with a subcooler which is proven as the best performing system for these cases [Tran, 2015^{a,b}]. In the HPD systems as shown in Figure 2, the evaporator is employed to recover heat from the exhaust air while the subcooler and the condenser are used to preheat the drying air. For the HP system using a gas engine to drive the compressor (figure 2 – left), the available heat from the coolant fluid and exhaust gases of the gas engine are also used to further heat drying air before it enters the auxiliary heater. Therefore, the performance of a gas engine driven compressor HP is represented by a combined heat and power efficiency (η_{CHP}) and calculated as the ratio of the useful heat used to heat drying air to the energy consumption of the gas engine:

$$\begin{aligned} \eta_{\text{CHP}} &= \frac{Q_{\text{useful}}}{Q_{\text{fuel}}} \\ &= \frac{Q_{\text{subcooler}} + Q_{\text{cond}} + Q_{\text{coolant}} + Q_{\text{exhaust}}}{Q_{\text{fuel}}} \\ &= \text{COP} * \eta_{\text{mech}} + \eta_{\text{thermal, coolant fluid}} + \eta_{\text{thermal, exhaust gases}} \end{aligned} \quad [2]$$

where COP, η_{mech} , $\eta_{\text{thermal,coolant fluid}}$ and $\eta_{\text{thermal,exhaust gases}}$ are:

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

$$\eta_{\text{mech}} = \frac{W_{\text{comp}}}{Q_{\text{fuel}}} \quad [4]$$

$$\eta_{\text{thermal, coolant fluid}} = \frac{Q_{\text{coolant}}}{Q_{\text{fuel}}} \quad [5]$$

$$\eta_{\text{thermal, exhaust gases}} = \frac{Q_{\text{exhaust}}}{Q_{\text{fuel}}} \quad [6]$$

The operating mechanism of a gas turbine driven compressor HP is similar to a gas engine driven compressor HP (figure 2 - right). However, in this case, the only waste heat stream is exhaust gas of the gas turbine. Thus, the η_{CHP} of a gas turbine driven compressor HP is defined as:

$$\eta_{\text{CHP}} = \text{COP} * \eta_{\text{mech}} + \eta_{\text{thermal, exhaust gases}} \quad [7]$$

The total power consumption of a HPD system is:

$$Q_{\text{HPD}} = Q_{\text{fuel}} + m_{\text{air}} (h_{\text{heated air}} - h_{\text{pre-heated air}}) \quad [8]$$

where $h_{\text{pre-heated air}}$ is the enthalpy of the drying air preheated by the exhaust gas of the gas engine or gas turbine.

Economic benefit calculation

In this study, the economic benefit of a HPD is investigated based only on the operating energy cost of the drying system and is expressed by the relative energy cost (REC), which is defined as the ratio of the energy cost of the HPD system to the energy cost of the original dryer:

$$\text{relative energy cost} = \frac{C_{\text{HP dryer}}}{C_{\text{original dryer}}} 100 \quad [\%] \quad [9]$$

The fixed costs (investment, installation) of the HPD are not considered. For the original dryer, the fuel to heat conversion ratio of the air heater is assumed 100%.

Assumptions and inputs

The calculations in this paper are based on the assumptions that the refrigerant flow rate is constant and heat and pressure losses are neglected. For the gas engine, the mechanical and thermal efficiencies at part load operation are the same as at full load operation. These values are the efficiencies of the MAN E0834-E312 (for the animal food drying case) and MAN E0836-E302 (for the blood drying case) gas engines operating at speed of 1500rpm. It is also assumed that the temperature of the coolant fluid is 90°C and the minimum temperature difference of the coolant fluid/air heat exchanger is 10°C. This means that the maximum temperature of the drying air (heated by the coolant fluid/air heat exchanger) that can be reached is 80°C (T_{a3} , figure 3). For the gas turbine, the mechanical efficiency is also assumed constant while the mass flow rate of the exhaust gas is proportional to the turbine load. The parameters of the gas turbine are obtained from the Capstone C65 and Capstone C200LP gas turbines for the animal food drying and the blood drying systems, respectively. The input parameters of the calculations are presented in Table 1.

PERFORMANCE ANALYSIS BY SIMULATION

Generally, we would like to recover as much heat as possible from the exhaust air to reduce heat losses. However, more recovered heat means that the HP will operate at a lower evaporation temperature and thus pressure. Moreover, as mentioned above, the heat recovered from the exhaust air is used to preheat the ambient air. Thus, the more heat recovered from the exhaust air, the higher the temperature of the

Table 1. The inputs for simulating calculation

Parameter	Unit	Value	
		Animal food drying	Blood drying
Ambient air temperature	°C	20	
Ambient air relative humidity	%	60	
Isentropic efficiency of the compressor	-	0.7	
Pinch point of the evaporator, condenser	°C	8	
Pinch point of the subcooler	°C	15	
Superheating temperature	°C	5	
Volumetric flow rate of the hot air	m ³ /h	12000	70000
Volumetric flow rate of the drying air flows to the bottom of the drying chamber.	m ³ /h	5000	8000
Volumetric flow rate of the drying air flows to the bottom of the cylinder.	m ³ /h	-	2000
Parameters of the gas engine			
Gas engine power output (full load)	kW	47	75
$\eta_{\text{thermal, coolant fluid}}$	-	0.302	0.308
$\eta_{\text{thermal, exhaust gases}}$	-	0.186	0.225
η_{mech}	-	0.364	0.367
Parameters of the gas turbine			
Gas turbine power output (full load)	kW	65	190
η_{mech}	-	0.29	0.31
Exhaust gas temperature	°C	309	280
Exhaust gas flow (full load)	kg/s	0.49	1.3

preheated air. Consequently, an increase in temperature difference between the condensation and evaporation is created. This causes a decrease of the HP system efficiency. Hence, there is a trade-off between the amount of heat recovered and the HP system efficiency.

Gas engine driven compressor HPD systems

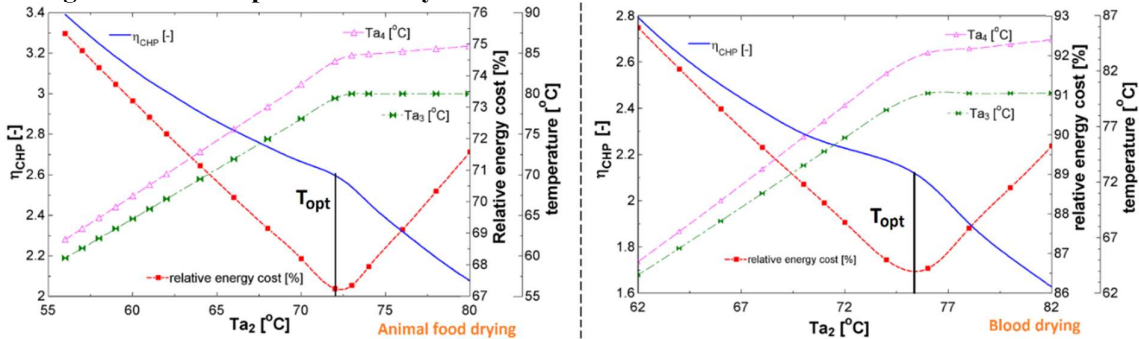


Figure 3: REC, T_{a3} , T_{a4} and η_{CHP} as a function of T_{a2} for the system using a gas engine to drive the compressor in the animal food drying case (left) and the blood drying case (right).

Figure 3 presents the REC and η_{CHP} of the drying system using a gas engine to drive the compressor as a function of the air temperature after the condenser (T_{a2}) for both drying cases. The temperatures of the air after the coolant liquid/air heat exchanger (T_{a3}) and after the exhaust gases/air heat exchanger (T_{a4}) are indicated as well. It can be seen that, for both cases, the energy cost of the systems decreases almost linearly by increasing T_{a2} until it reaches the optimum value T_{opt} which depends on the maximum temperature ($T_{a3\text{max}}$) of the drying air at the exit of the coolant heat exchanger or, in other words, the coolant fluid temperature and the minimum temperature difference of the coolant heat exchanger. Indeed, when T_{a2} is lower than T_{opt} and the corresponding T_{a3} is lower than this $T_{a3\text{max}}$, all the heat available from the coolant fluid can be used to heat the drying air, therefore the energy cost decreases when T_{a2} increases. Once T_{a2} reaches its optimum value and T_{a3} reaches $T_{a3\text{max}}$, the energy used by the gas engine increases with increasing T_{a2} 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, η_{CHP} significantly

decreases and the energy cost rises. In the animal food drying case, the minimum relative cost is about 67.2% at the T_{a2} and η_{CHP} of 72°C and 2.6, respectively. Meanwhile, these values in the blood drying case are 86.5%, 76°C and 2.1, respectively.

Generally, the exhaust gas temperature is much higher than the coolant fluid temperature and therefore the optimum operating temperature does not depend on the exhaust gases temperature. However, the percentage of heat recovered from exhaust gases affects the performance of the gas engine driven HP according to equation (2).

Gas turbine driven compressor HPD systems

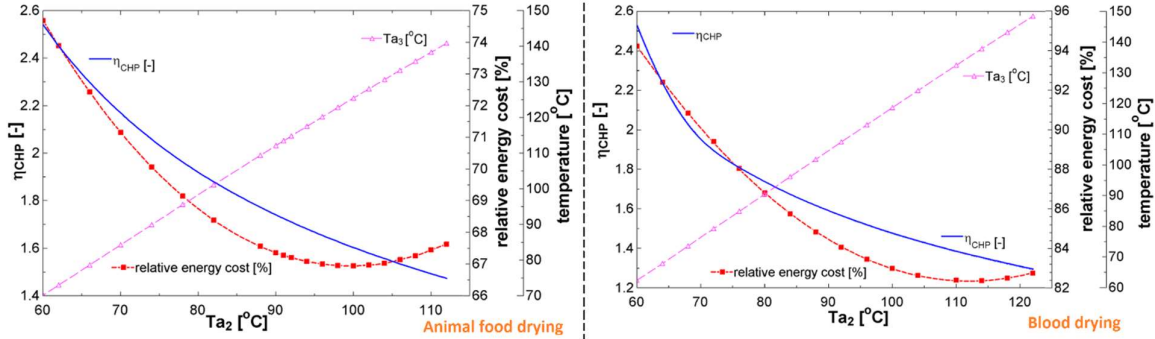


Figure 4: REC, T_{a3} and η_{CHP} as a function of T_{a2} for the system using a gas turbine to drive the compressor in the animal food drying case (left) and the blood drying case (right).

The dependence on T_{a2} of the REC, temperature of the air after the exhaust gases/air heat exchanger (T_{a3}) and η_{CHP} of the HP system using a gas turbine to drive the compressor is shown in figure 4. Since this HP system only has one waste heat stream, being exhaust gases, and its temperature is much higher than the T_{a2} , the minimum REC is not limited by the waste heat temperature as was the case in the gas engine driven compressor HPD system. In this case, there still is an optimum operating condition at which the REC reaches the minimum value, however this optimum operating condition depends on all of the parameters (Table 1) used for the calculation. Thus, it differs for each drying system. The minimum RECs are 66.2% and 82.2% for animal food drying and blood drying, respectively, which are little smaller in comparison with the gas engine case. However, the optimums T_{a2} are much higher being 101°C and 111°C, respectively.

One should also notice that for the animal food drying case, the auxiliary heater could be eliminated if the gas turbine driven HP operates at T_{a2} and T_{a3} of 112°C and 140°C, respectively.

Comparison of the HPD system performance between two drying cases

In both drying systems, the exhaust air is the heat source for the HP system and its temperature is 50°C. However, the exhaust air relative humidity of the animal food drying case (64%) is higher than that of the blood drying system (50%). In addition, the incoming air temperature of the latter system is already preheated to 50°C, thus it decreases the subcooling. Therefore, the HP COP of the animal food drying system is much higher than that of the blood drying system. As a result, the η_{CHP} of the animal food drying system is higher than the blood drying system (Figure 3 and 4) even the mechanical and thermal efficiencies of the gas engine/turbine of the former system are a bit lower.

About the economic benefits, we can see that the REC of the blood drying case is much higher than that of the animal food drying case. Besides having a lower η_{CHP} , it is also because the drying temperature of the former (180°C) is higher, or in other words, the proportion of the heat supplied by the HP system to the heat supplied by the auxiliary heater is smaller.

CONCLUSION

This paper presents the results of two case studies about the performance of thermally driven HP integration in an animal food dryer and a blood dryer. In order to study the performance of HP dryers,

computer simulation models of the original high temperature dryer and the proposed system with HP integration and auxiliary heating are developed. The results showed that, when using a gas engine driven compressor HP, 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. This limitation can be eliminated by using a gas turbine driven compressor HP since a gas turbine only has one waste heat stream being exhaust gases. In addition, the air temperature heated by this system is also higher in comparison with the gas engine driven compressor HP.

NOMENCLATURE

COP	coefficient of performance
$C_{\text{HP dryer}}$	total energy cost of the dryer with a HP integrated [Euro/year]
$C_{\text{original dryer}}$	energy cost of the original dryer [Euro/year]
m_{air}	mass flow rate of the drying air [kg s^{-1}]
Q_{cond}	heat release rate at the condenser [kW]
Q_{coolant}	heat release rate at the coolant heat exchanger [kW]
Q_{evap}	heat delivery rate to the evaporator [kW]
Q_{exhaust}	heat release rate at the gas engine exhaust air heat exchanger [kW]
Q_{fuel}	power consumption of the gas engine [kW]
Q_{HPD}	power consumption of the dryer integrated HP [kW]
$Q_{\text{original dryer}}$	power consumption of the original dryer [kW]
$Q_{\text{subcooler}}$	heat release rate at the subcooler [kW]
W_{comp}	power consumption of the compressor [kW]

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